Simulation and Modeling of the CO_2 Refrigeration System

Chi-Chuan Wang (王啟川), PhD, FASME, FASHRAE

Professor, Department of Mechanical Engineering, National Chiao Tung University, Hsinchu, Taiwan

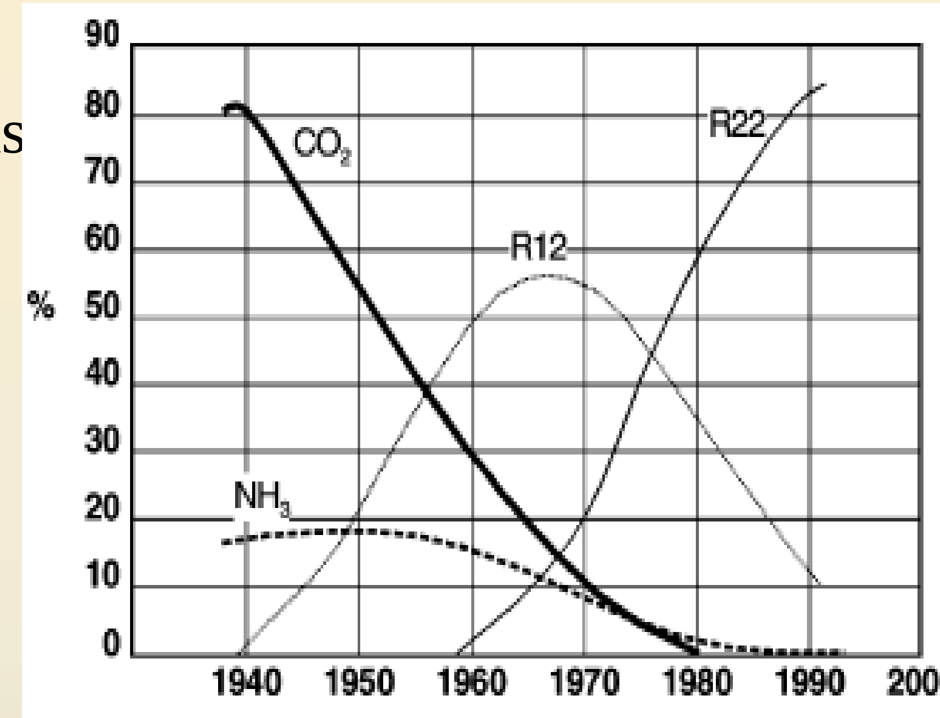
E-mail: ccwang@mail.nctu.edu.tw

Outline

1. General Background
2. Literature Reviews
3. Simulation Method
4. Simulation results
5. Conclusion

General Background for CO_2 Refrigeration

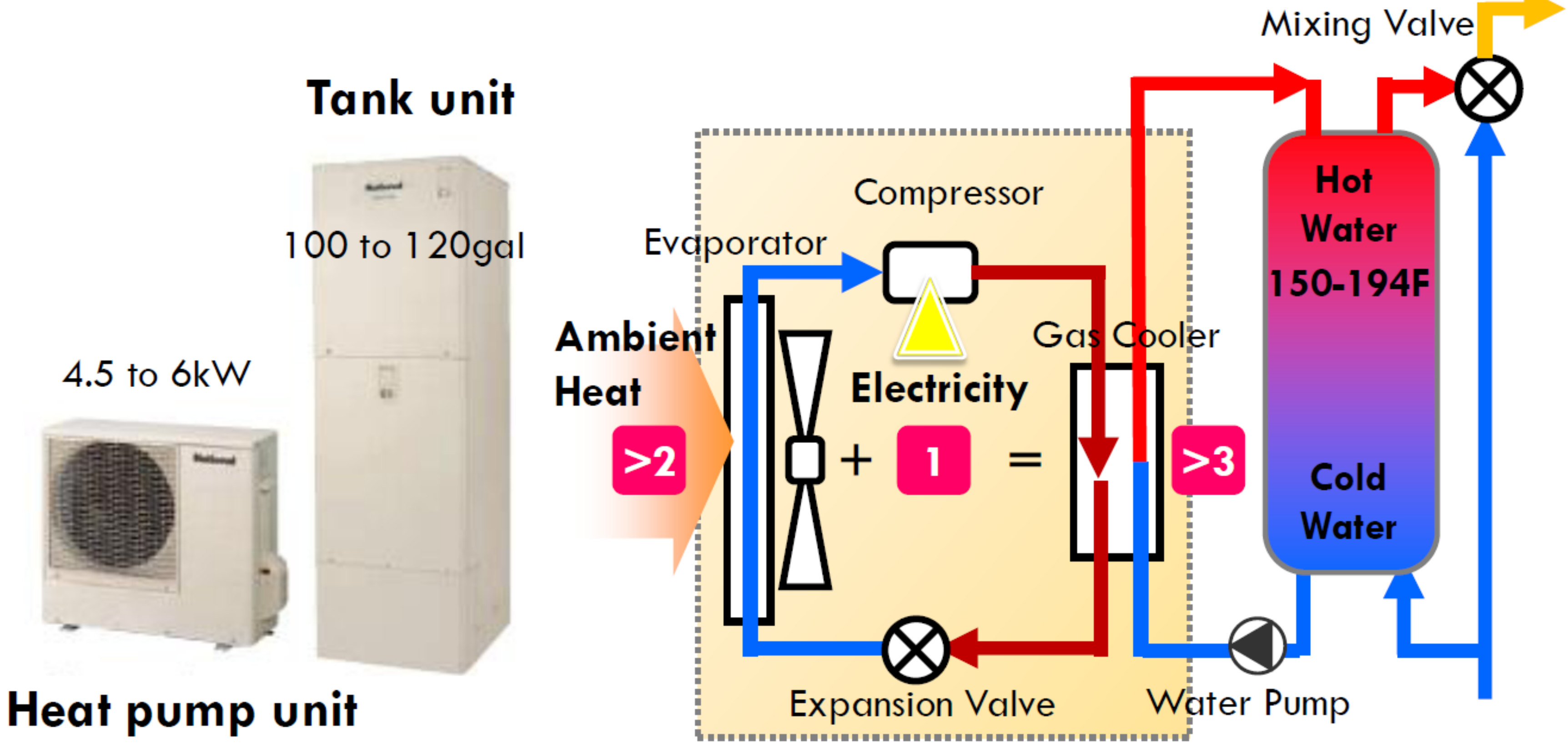
- Carbon dioxide (CO_2) had been used as a refrigerant for compression refrigeration system as early as 1880.
- Non-toxic, non-combustible, low cost and readily available.
- CO_2 Refrigerant is used over 80% in fishing vessel and relevant transportation.
- Carbon dioxide was greatly phased out when synthetic refrigerant was invented since 1920.
- From 1990, the concerns on environmental concerns has revived the comeback of CO_2.



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

Typical CO_2 Heat Pump Water Heater

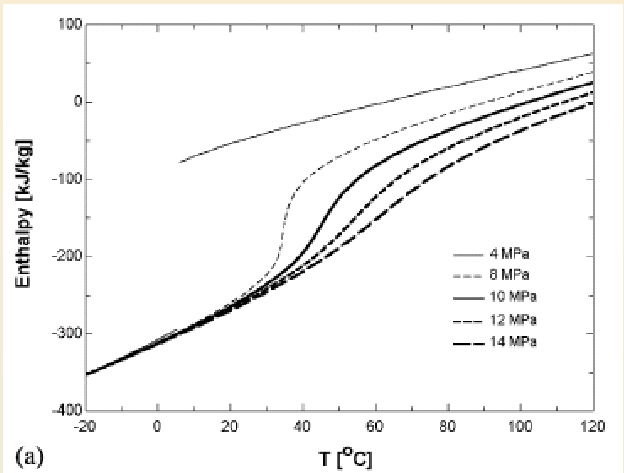
Move ambient heat into Hot water.

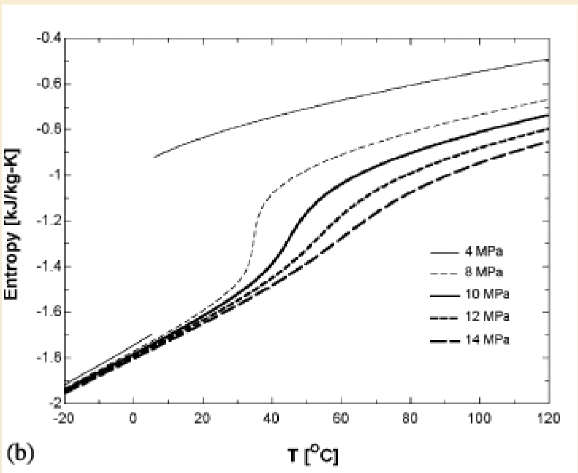


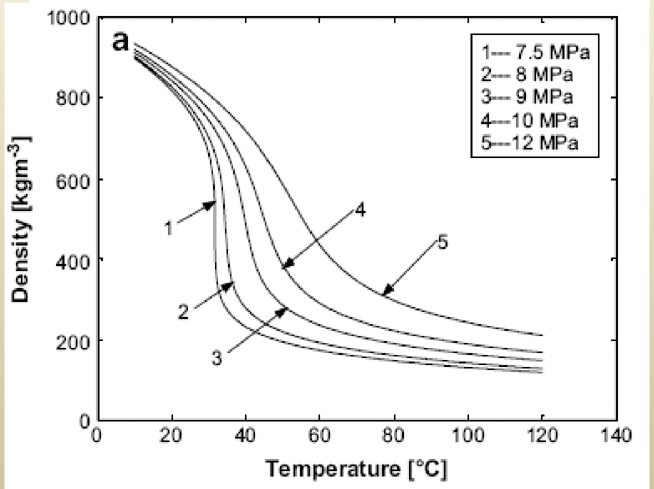
Source of the product picture: model# HE-K37AQS from Panasonic(Matsushita) online catalogue
http://national.jp/sumai/hp/online.html

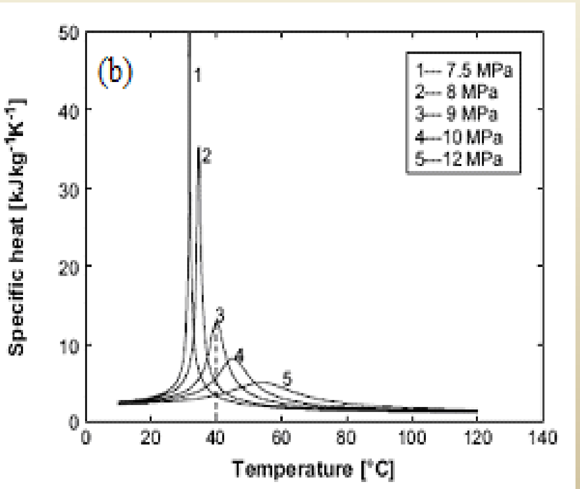
Thermodynamic and transport properties of CO_2 near the critical point

- Near the critical point, density, enthalpy and entropy changes considerably. Specific heat shows a spike phenomenon. The peak is especially pronounced at the “pseudo” critical point.

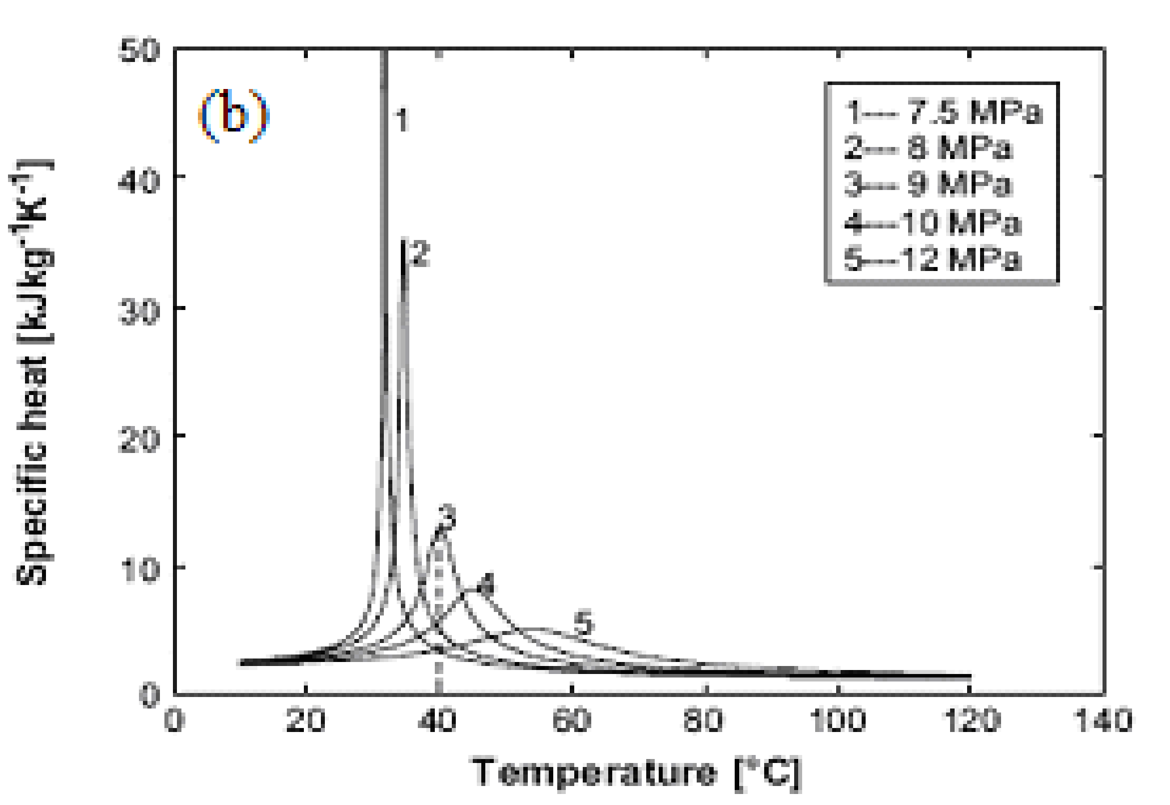






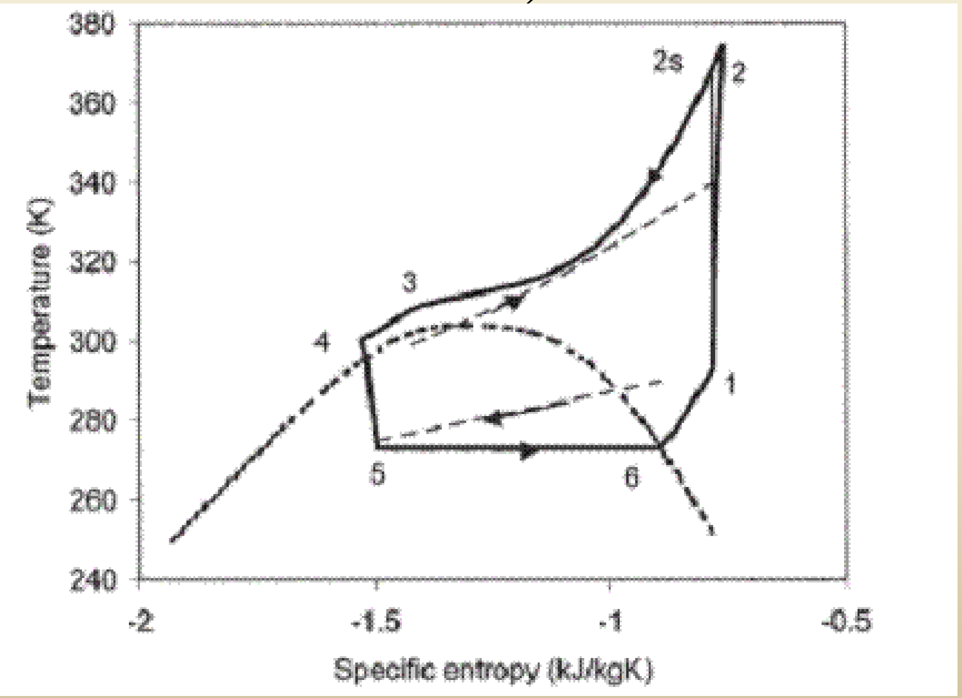


- When the pressure is increased above the critical pressure, each pressure value corresponds to an extreme value. This temperature with extreme specific heat value is called pseudo-critical temperature. This pseudo-critical temperature will increase as the pressure rises, but the spike of the maximum value of the specific heat becomes less evident with increasing pressure.

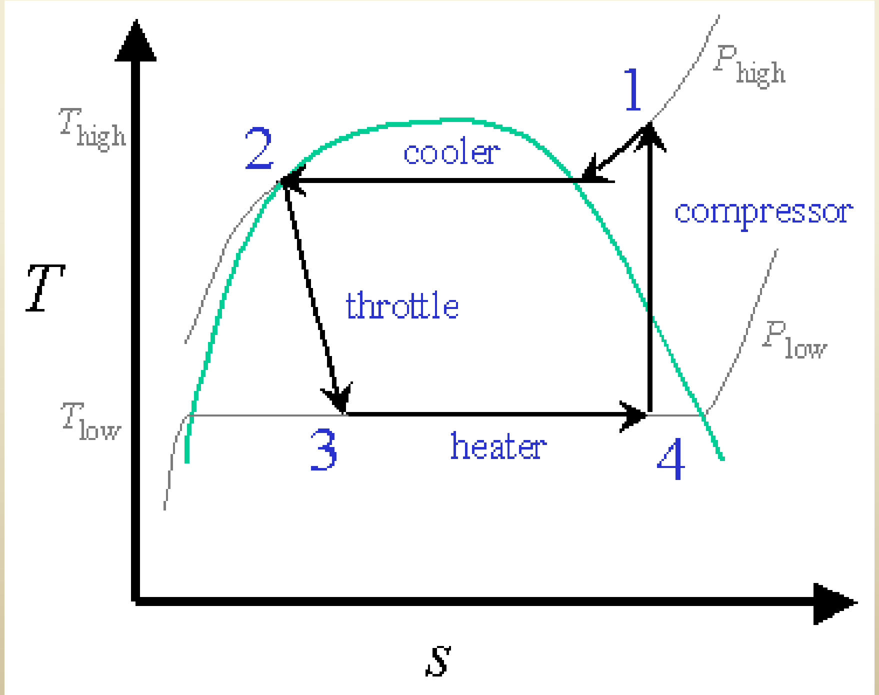


Typical CO_2 refrigerating cycle is above the critical point. The expansion process and the evaporation process are in the subcritical region. Therefore, CO_2 refrigeration cycle operates across supercritical and subcritical point region and they are called the Transcritical Cycle. Because the temperature of the critical point of carbon dioxide is low (about 31.1 °C). Therefore, for the normal ambient temperature, no condensing occurs and single-phase gas cooling is encountered.

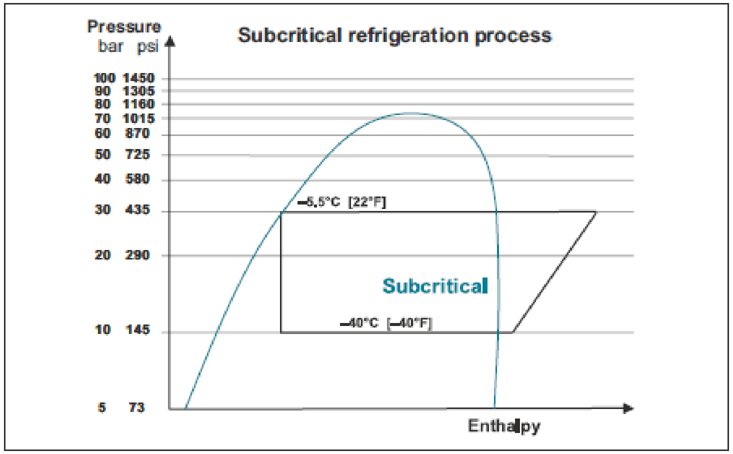
CO_2

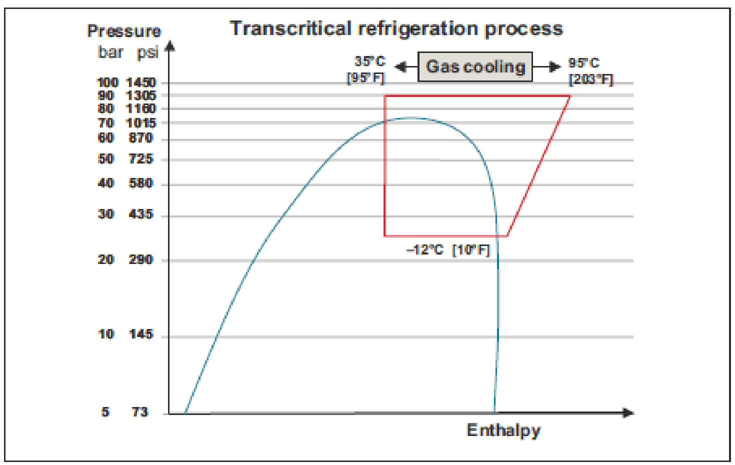


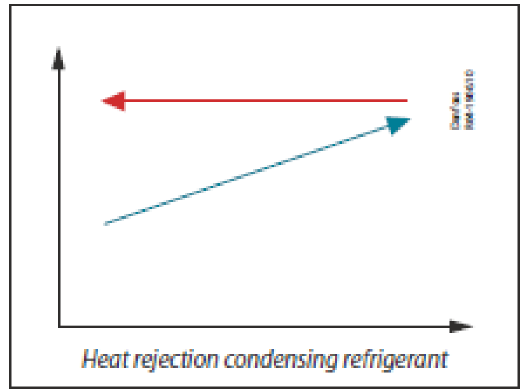
R-134a

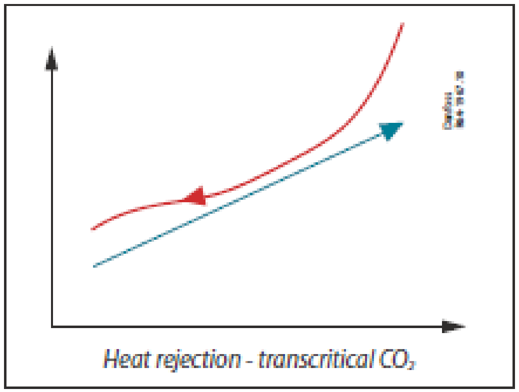


Typical P-h diagram for CO_2 & conventional refrigerant









COP of the Refrigerants

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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.

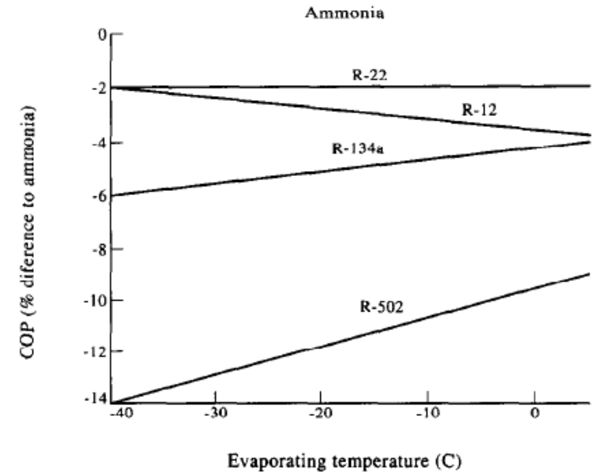


Fig. 2. Relative COP of an isentropic vapour compression cycle for different refrigerants [4].

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.

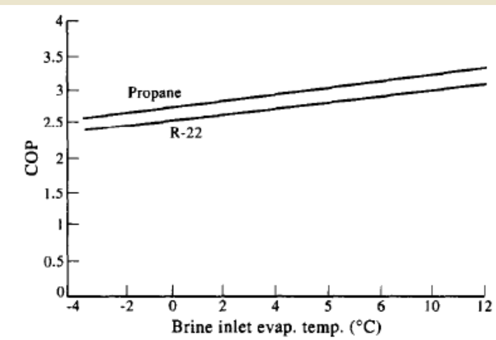


Fig. 3. COP of a heat pump with R-22 and R-290, for different brine inlet evaporator temperatures [12].

INTERNATIONAL JOURNAL OF REFRIGERATION 31 (2008) 545–551

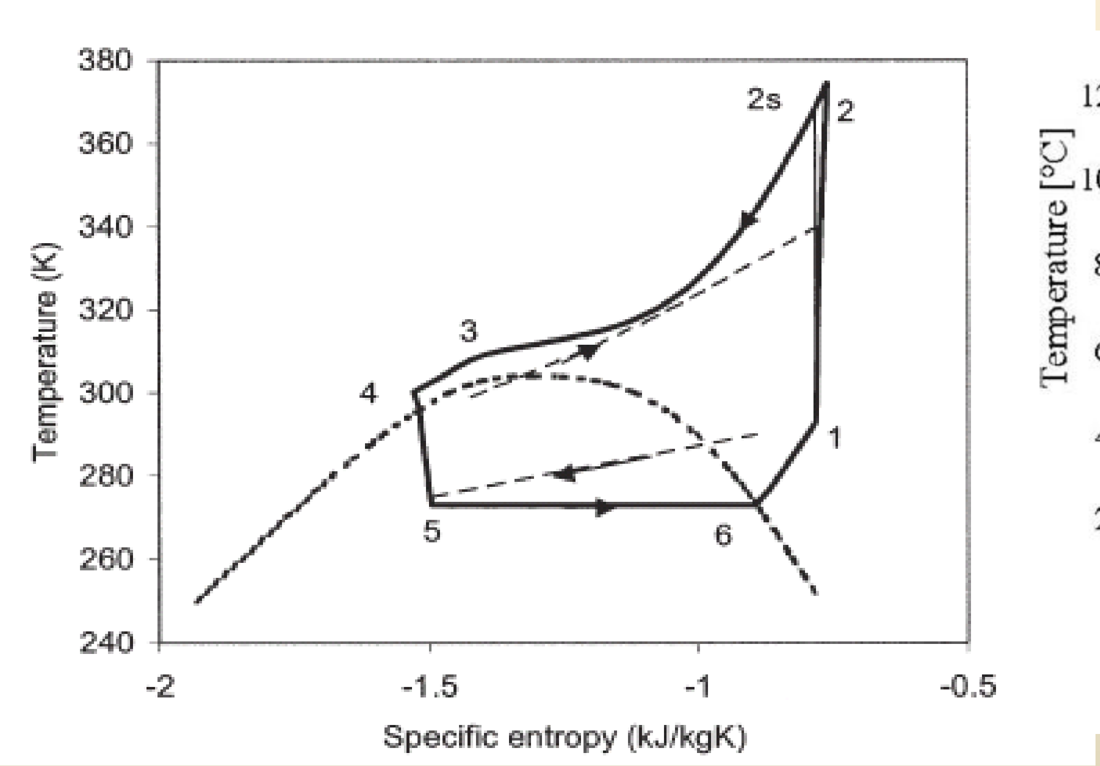
Comparison between CO_2, R410A & R-407C

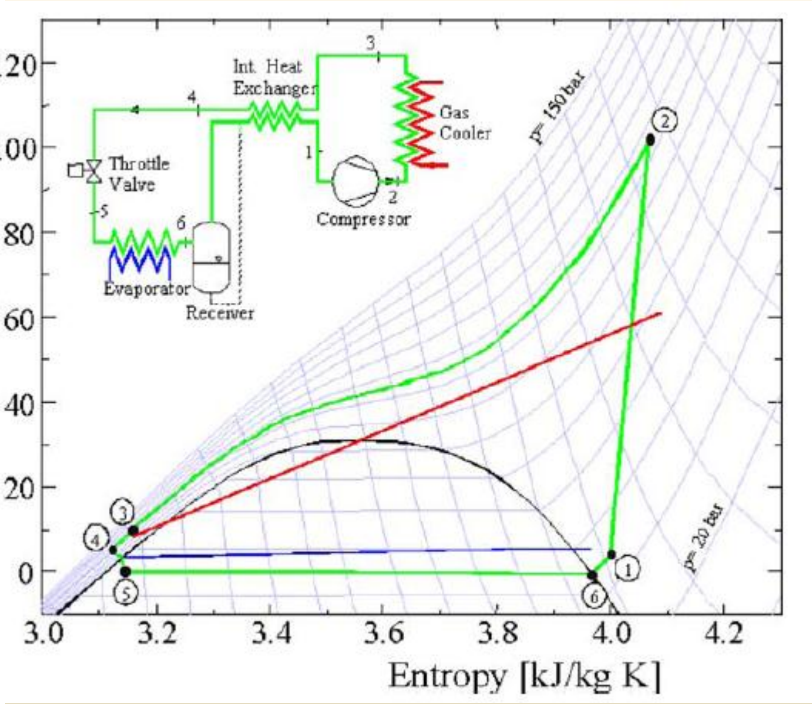
Table 1. Refrigerant characteristics

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.

Aim of this study: develop a simulation program for CO_2 refrigeration cycle.





Literature Reviews

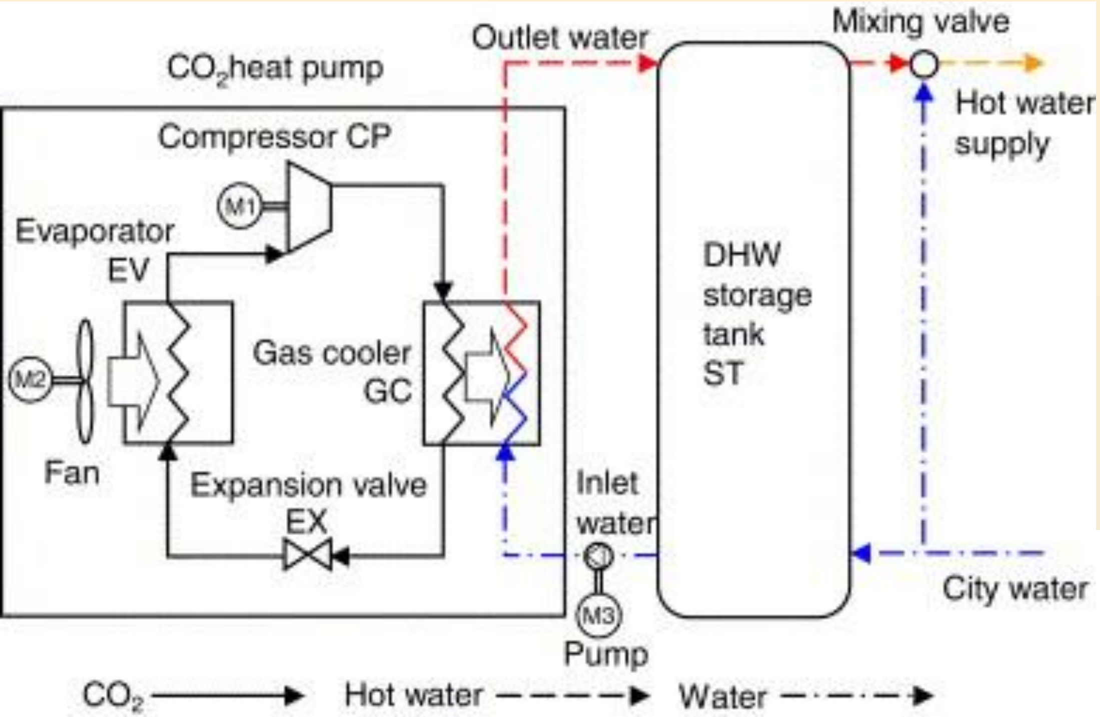
- There is a big difference between carbon dioxide refrigeration cycle and traditional refrigerants. The critical temperature and critical pressure of CO_2 are 31.1°C and 78.8 Bar in room temperature applications (such as outside air at 35 °C), and the refrigerant will operate above the critical point.
- CO_2 refrigerant can't be cooled by ordinary condenser. CO_2 is cooled by near single-phase gas. Normally we call the heat exchanger as **Gas Cooler** rather than condenser.

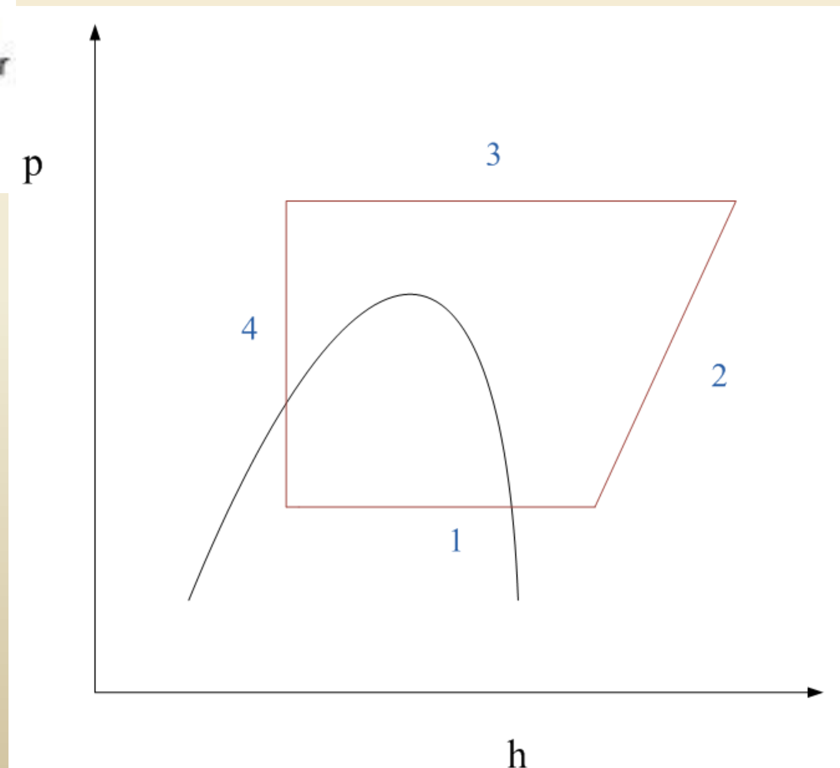
	Gas Cooler	Evaporator	Expansion device Model	Inner HX	Method (conditions outside the HX geometry, the water inlet temperature, air inlet temperature)
Kim et al.	Water Tube-in-tube	Water Tube-in-tube	$\Delta h=0$	yes	Fixed: Pdis、Tsup
Yang et al.	Water Shell-tube	Water Shell-tube	$\dot{W}_{\exp}=G_r(h_3-h_{4,is})\times\eta_{\exp,is}\times\eta_{\exp,m}$	No	Fixed: Pdis
Sarker et al.	Water Tube-in-tube	Water Tube-in-tube	$\Delta h=0$	Yes	Maximum COP
Yokoyama et al.	Water Tube (double-pipe)	Water Tube (double-pipe)	$\Delta h=0$	No	Fixed: $\dot{m}_{co2}$、P_{eva}、$T_{ci,gc}$
Yamaguchi et al.	Water Tube-in-tube	Air Fin-tube	$\Delta h=0$	yes	Fixed：$T_{w,o}$
Wang et al.	Water Tube-in-tube	Water Tube-in-tube	$\Delta h=0$	no	Maximum COP T_{sup}、$T_{w,o}$
Sarker et al.	No description	No description	$\Delta h=0$	yes	Fixed Maximum COP
Sarker et al.	Water Tube-in-tube	Water Tube-in-tube	$\Delta h=0$	No	Fixed :Pdis Fixed :degree of superheat
Sarker et al.	Water Tube-in-tube	Water Tube-in-tube	$\Delta h=0$	No	Fixed: Pdis Fixed :Psuc

Basic outline of the present simulation method

	Gas Cooler	Evaporator	Expansion device Model	Inner HX	Method (conditions outside the HX geometry, the water inlet temperature, air inlet temperature)
Present Study	Water Tube-in-tube	Air Fin-and-Tube (Wet Coil)	$\Delta h = 0$ Capillary pressure drop equation	No	No Fixed conditions

Simulation method





The CO_2 refrigeration Cycle contains four major components, including a gas cooler, an evaporator, a compressor and an expansion valve.

Compressor Model

The compressor calculations assume for the isentropic adiabatic process. First, we can generate equation by the continuity equation and the energy equation.

$$\dot{m}_{r,o} - \dot{m}_{r,i} = 0$$

$$\dot{m}_{r,o} h_{r,o} - \dot{m}_{r,i} h_{r,i} = W_{com}$$

isentropic efficiency and volumetric efficiency for the compressor

$$\eta_{isen} = \frac{h_{r,isen} - h_{r,i}}{h_{r,o} - h_{r,i}} \qquad \eta_v = \frac{\dot{m}_{r,i}}{\rho_{r,i} V_{com} N_{com}}$$

From Sarkar et al.,

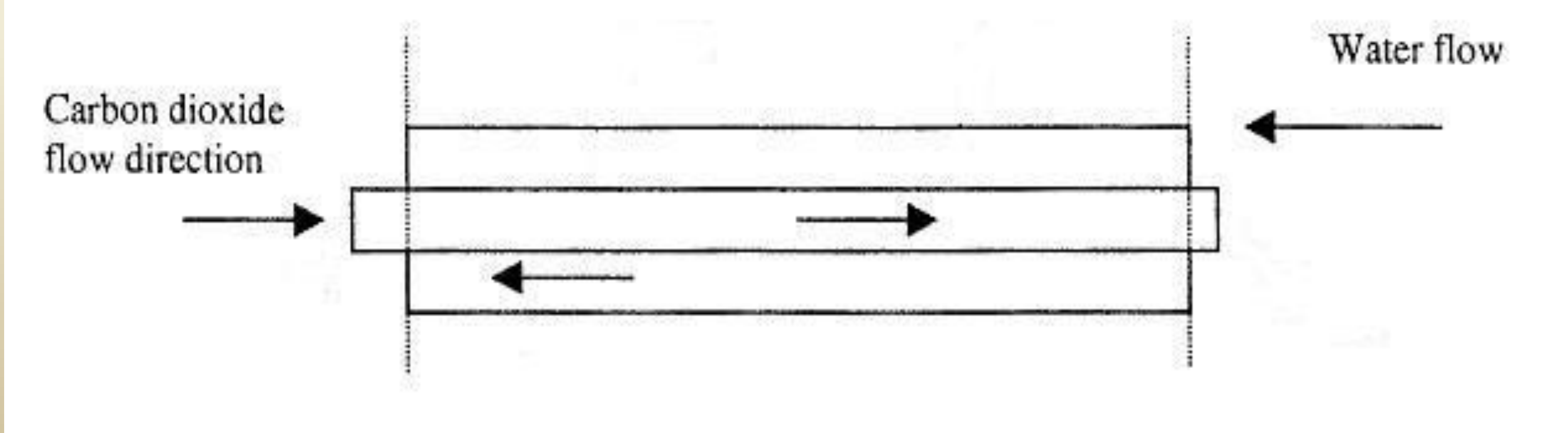
$$\eta_v = 0.9207 - 0.0756\left(\frac{P_{dis}}{P_{suc}}\right) + 0.0018\left(\frac{P_{dis}}{P_{suc}}\right)^2$$

$$\eta_{isen} = -0.26 + 0.7952\left(\frac{P_{dis}}{P_{suc}}\right) - 0.2803\left(\frac{P_{dis}}{P_{suc}}\right)^2 + 0.0414\left(\frac{P_{dis}}{P_{suc}}\right)^3 - 0.0022\left(\frac{P_{dis}}{P_{suc}}\right)^4$$

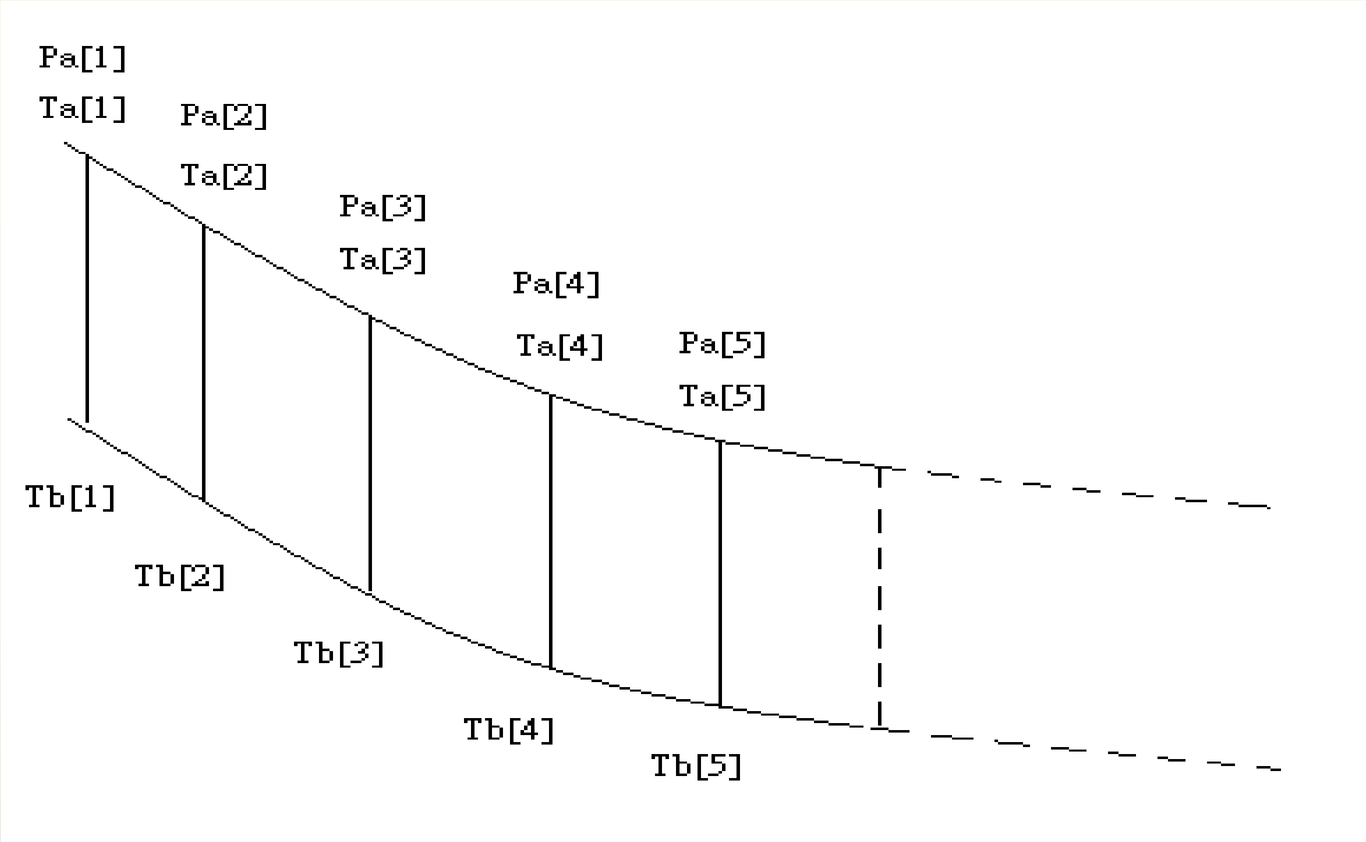
Gas Cooler Model

Objective: Simulation of the Targeted HX – A Tube-in-tube HX (Double Pipe HX)

- The heat exchanger is a tube-in-tube type, and the water flows counter-currently against the coolant (carbon dioxide) during the heat exchanging process.



Discretizing the HX due to significant change of physical properties



Due to considerable change of physical properties, especially near pseudo-critical temperature, the heat exchanger must be subdivided into many small segments.

Basic equations

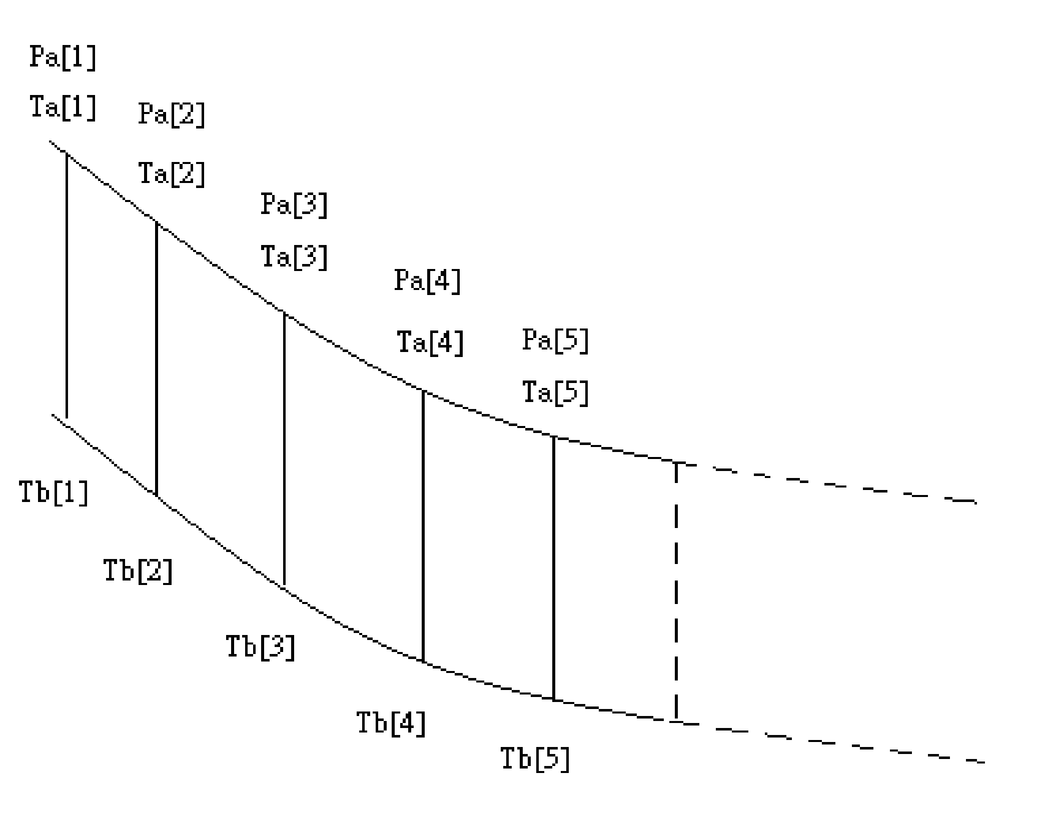
- Energy balance/pressure drop amid water and coolant can be written in the following equations:

$$Q_1 = m_c \times Cp_{c1} \times (T_{a1} - T_{a2}) = m_w \times Cp_{w1} \times (T_{b1} - T_{b2}) \qquad (1)$$

$$Q_1 = (UA)_1 \times (LMTD)_1 \qquad (2)$$

$$P_{a2} - P_{a1} = \frac{4L}{d} \times f \times \frac{G_c{}^2}{2\rho_c} \qquad (3)$$

Physical Configurations



Each segment has three unknown parameters. (water temperature, CO_2 temperature and CO_2 pressure).

There are three equations for each segment.

The unknown parameters can thus be solved through these equations.

HTCs in CO_2 & water

- For CO_2, Dang and Hihara (2004) correlation：

$$N_u = \frac{\left(\frac{f}{8}\right)(Re-1000)Pr}{1.07+12.7\sqrt{\frac{f}{8}}\left(Pr^{2/3}-1\right)}$$

$$Pr = \begin{cases} Cp_b\mu_b/\lambda_b, & for\ Cp_b \geq \overline{Cp} \\ \overline{Cp_b}\mu_b/\lambda_f, & for\ Cp_b < \overline{Cp}\ and\ \mu_b/\lambda_b \geq \mu_f/\lambda_f \\ \overline{Cp_b}\mu_f/\lambda_f, & for\ Cp_b < \overline{Cp}\ and\ \mu_b/\lambda_b < \mu_f/\lambda_f \end{cases}$$

- For water, Genielinski方程式：

$$N_u = \frac{\left(\frac{f}{2}\right)(Re-1000)Pr}{1.07+12.7\sqrt{\frac{f}{2}}\left(Pr^{2/3}-1\right)}$$

$$f = [1.58\ln(Re_b) - 3.28]^{-2}$$

Algorithm for Solution of the Gas Cooler

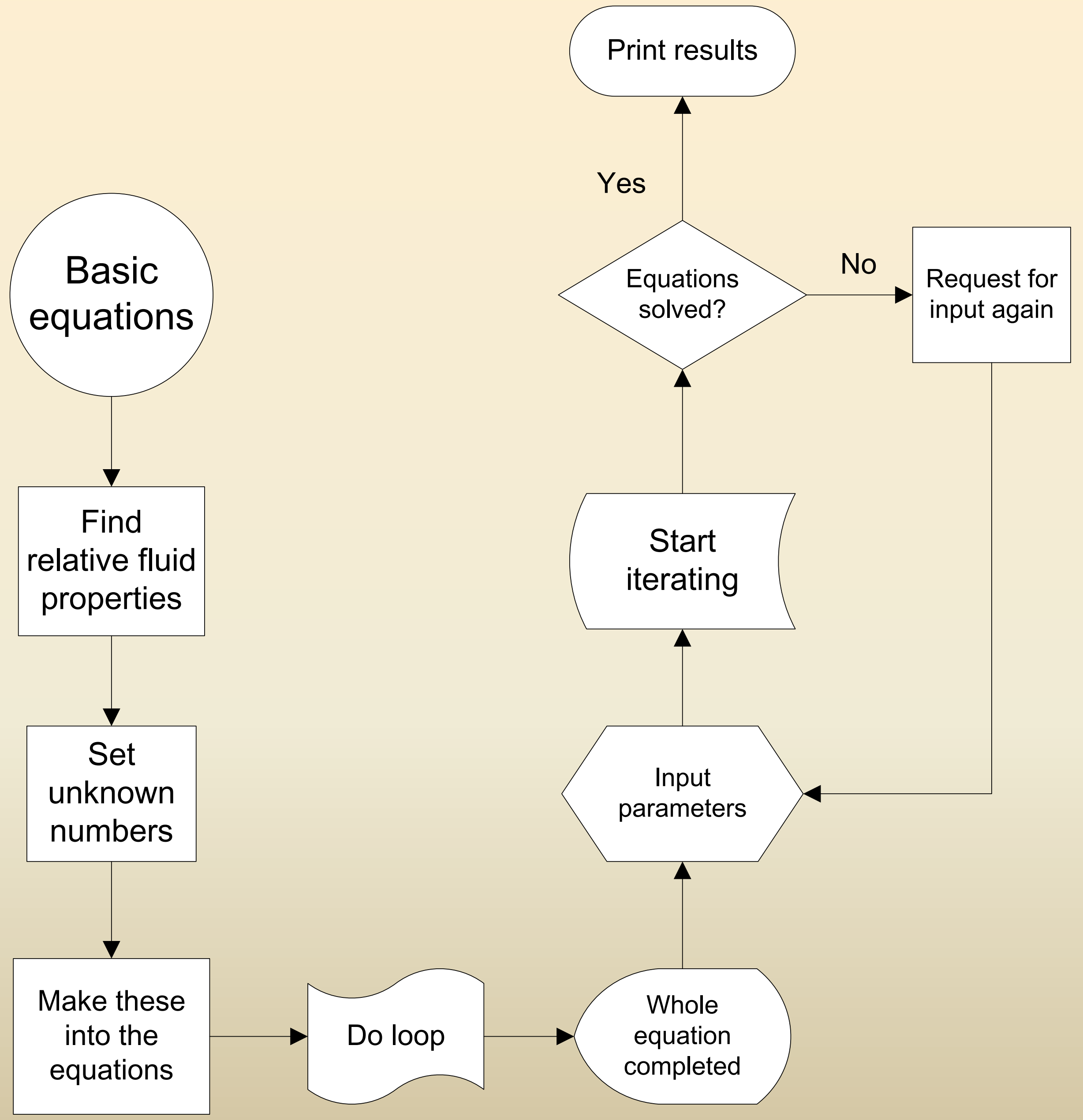
1. List the basic equations.

2. Make the properties of fluid into these equations.

3. Decide the segment number for calculation, and build the completed equation sets through do loops.

4. Start iterating.

5. Print out the results if the equations are convergent.

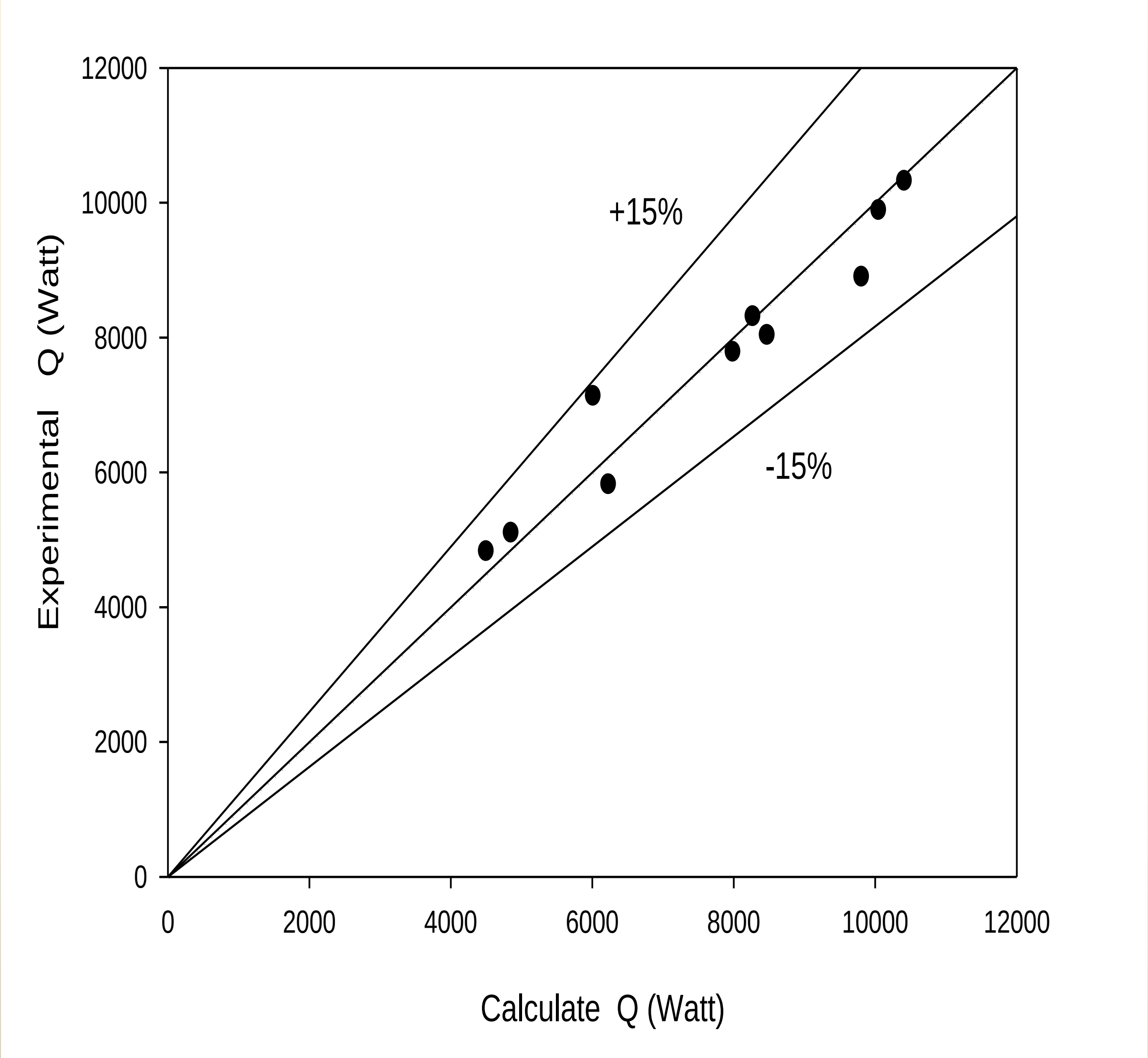

Basic equations
Find relative fluid properties
Set unknown numbers
Make these into the equations
Do loop
Whole equation completed
Input parameters
Start iterating
Equations solved?
Yes
Print results
No
Request for input again

Results and Discussion

- For validating the proposed model, calculation is compared with the measurements of Pitla et al.

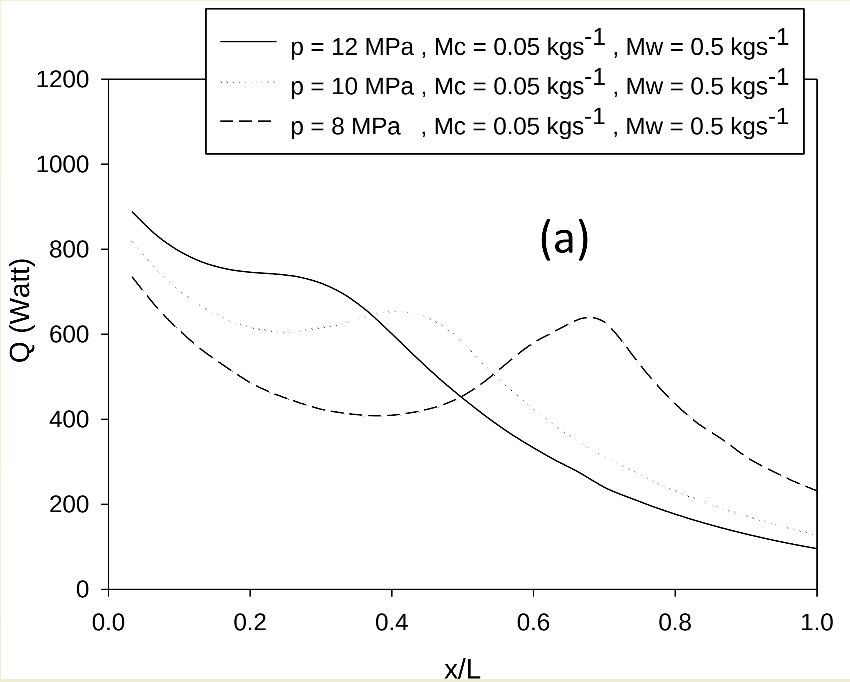
	$T_{c,in}$ (℃)	$P_{c,in}$(Mpa)	$T_{w,in}$ (℃)	Mc (kg/s)	Mw (kg/s)
Run1	121.2	9.44	20.8	0.01963	0.04011
Run2	126	11.19	24.2	0.0274	0.040497
Run3	73.3	13.33	36.12	0.02043	0.12914
Run4	123.5	10.8	27.21	0.02862	0.084087
Run5	107.2	8.11	24.2	0.0198	0.0455
Run6	123.4	8.98	22.3	0.02996	0.067864
Run7	118.3	7.79	21.2	0.02123	0.066434
Run8	115.8	8.60	18.9	0.03436	0.084087
Run9	114.9	8.76	18.9	0.03638	0.065091
Run10	113.4	9.50	15.9	0.03825	0.109052

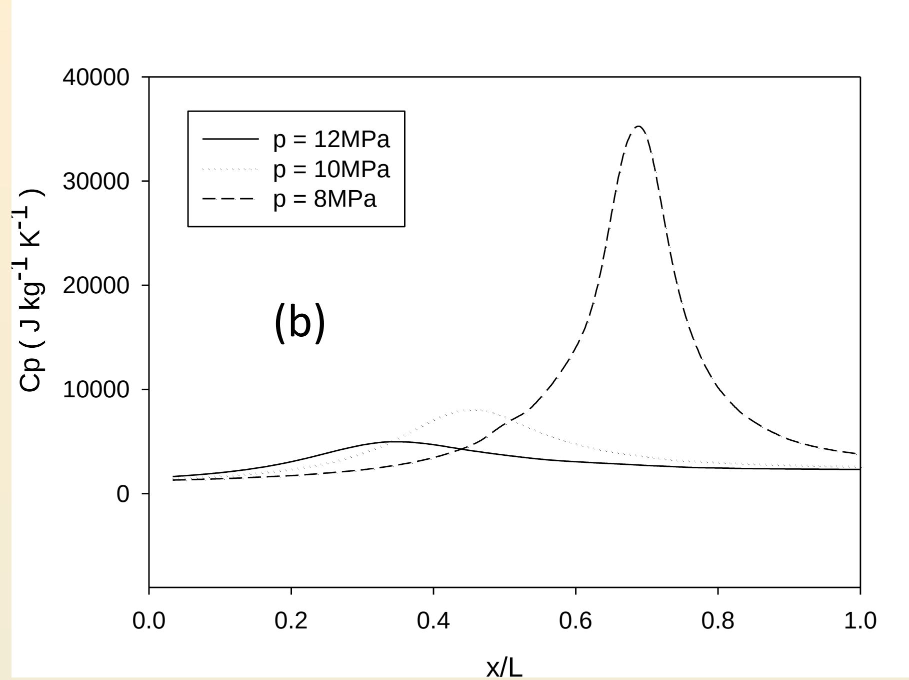
Pitla et al. had conducted CO_2 tube-in-tube heat exchanger with ID = 0.00472m and OD = 0.00635m for inner tube, ID = 0.01575m for outer tube.

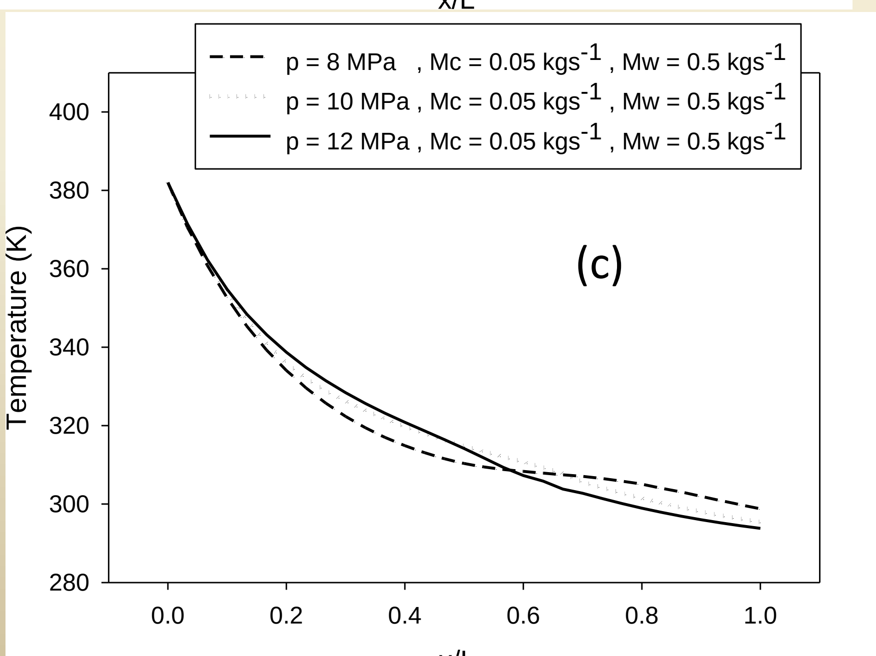


The calculations are in line with the experimental measurement, suggesting the applicability of the present modeling.

Results and Discussion

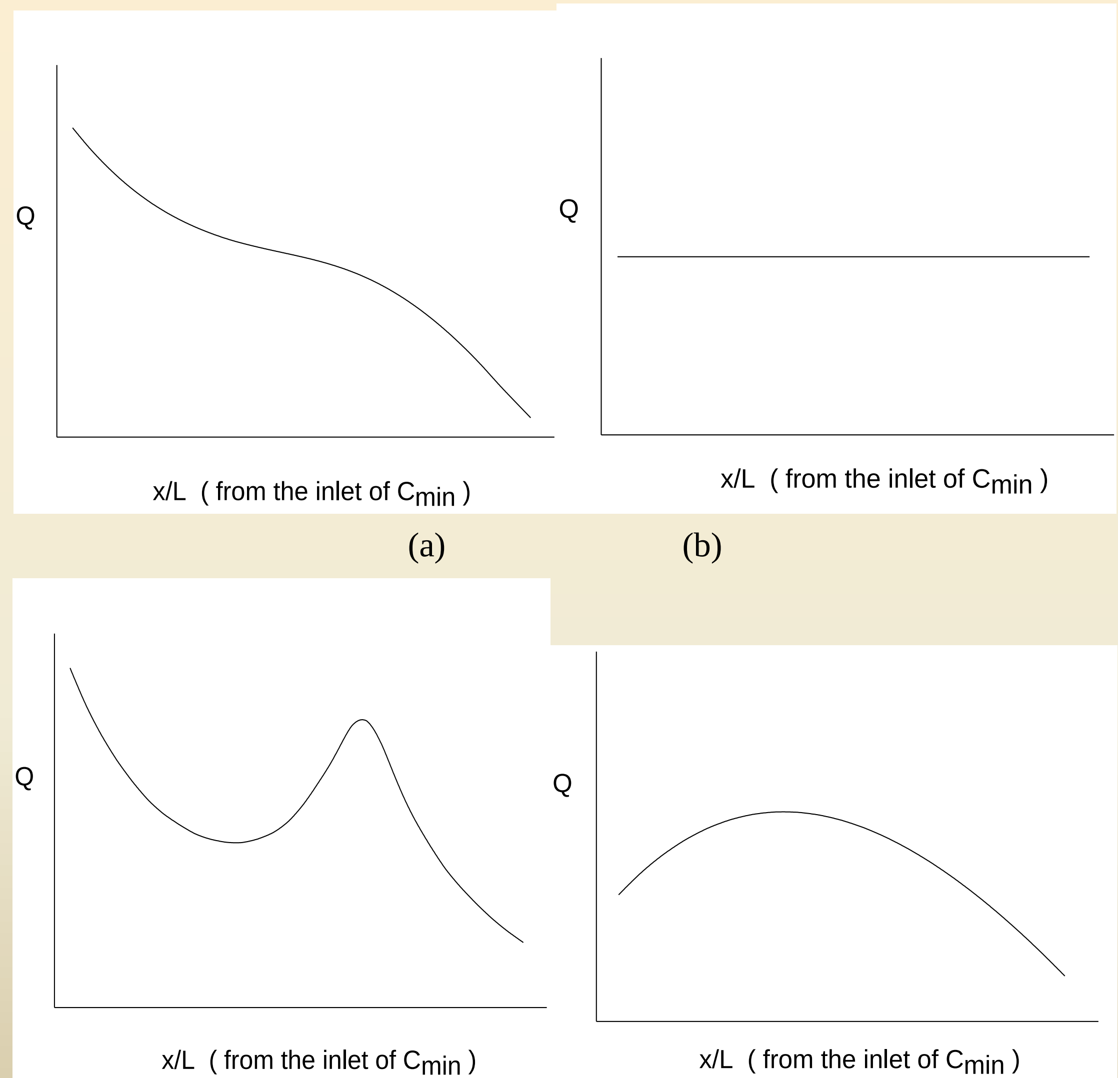






Effect of the inlet pressure on (a) the variation of the local heat transfer rate vs. the dimensionless tube length, (b) the variation of c_p vs. the dimensionless tube length and (c) the variation of the CO_2 temperature vs. the dimensionless tube length.

Schematic of the variation of the local heat transfer rate for a tube-in-tube heat exchanger: (a) Constant property, $C_{min} \neq C_{max}$; (b) Constant property, $C_{min} = C_{max}$; (c) CO_2 flow across the pseudo-critical point; and (d) Variable property.



(c) (d)

Concluding remarks for Gas Cooler

- Unlike conventional working fluid which normally shows a monotonically decrease of local heat transfer rate along the tube length, the CO_2 shows a different trend as compared to typical sub-critical fluids.
- The local heat transfer rate does not monotonically decrease with the tube length.
- In fact, a plateau occurs somewhere inside the heat exchanger. Moreover, a second maximum is seen when p is below 10 MPa, yet a significant of recovering of local heat transfer rate is encountered for p = 8 MPa despite the maximum temperature difference still occurs at the water inlet.
- The phenomenon is attributed to the significant rise of specific heat of CO_2 near pseudo critical temperature.
- The effect of tube diameter plays a crucial role of CO_2 HX, this is especially pronounced when the thermal resistance is on the CO_2 side.

Capillary tube Model

The calculation condition is based on the continuity equation, energy equation and pressure drop equations:

$$\dot{m}_{r,o} - \dot{m}_{r,i} = 0$$

$$\dot{m}_{r,o} h_{r,o} - \dot{m}_{r,i} h_{r,i} = 0$$

Agrawal et al. [14] pressure drop equation :

$$\frac{dp}{dL} = \left(\frac{dp}{dL}F\right) + \left(\frac{dp}{dL}a\right) \qquad \frac{dp}{dL} = -G^2\left(f_{tp}\frac{v}{2D} + \frac{dv}{dL}\right)$$

Lin et al. [15] empirical formula can calculate the friction coefficient of the two-phase flow :

$$f_{tp} = \varphi_{tp} f_{sp}\left(\frac{v_{sp}}{v_{tp}}\right) \qquad \varphi_{tp} = \left[\frac{(8/\mathrm{Re}_{tp})^{12} + (A_{tp}^{16} + B_{tp}^{16})^{-3/2}}{(8/\mathrm{Re}_{sp})^{12} + (A_{sp}^{16} + B_{sp}^{16})^{-3/2}}\right]^{1/12}\left[1 + x\left(\frac{v_g}{v_l} - 1\right)\right]$$

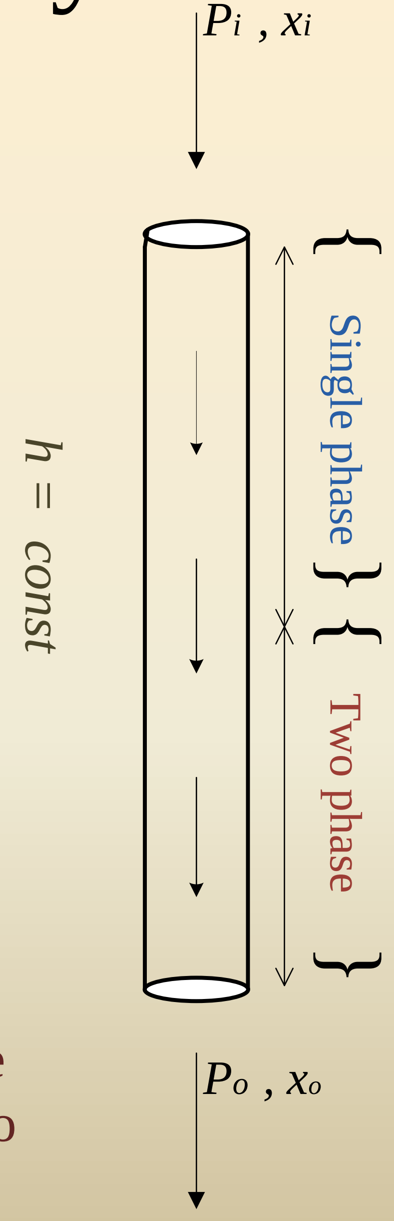
$$A = 2.457\ln\frac{1}{(7/\mathrm{Re}_{tp})^{0.9} + 0.27\varepsilon/D} \qquad B = \frac{37530}{\mathrm{Re}_{tp}}$$

pressure drop in capillary tube

The refrigerant flow out from the condensation region and through a smooth tube in single-phase liquid. It's change to two-phase in the tube in constant enthalpy.

$$\frac{1}{\mu_{tp}} = \frac{1-x}{\mu_l} + \frac{x}{\mu_g}$$

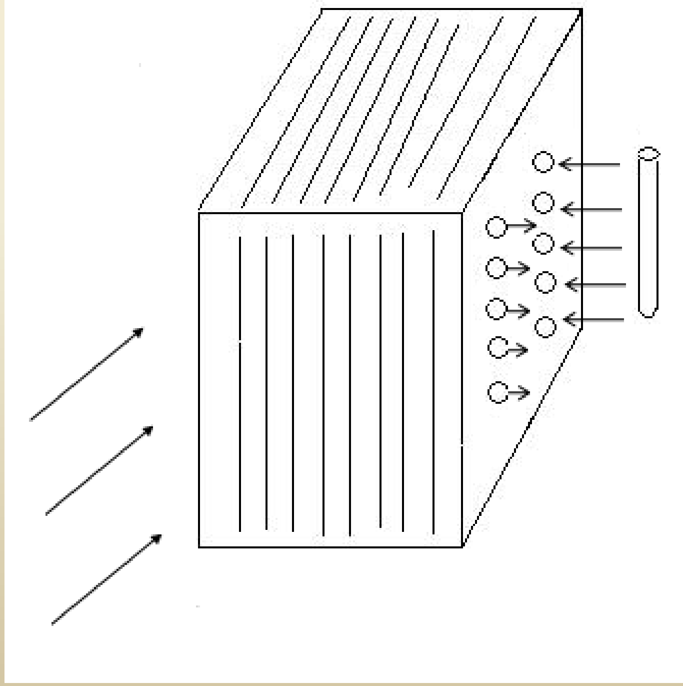
$$\frac{dp}{dL} = -G^2 \left(f_{tp} \frac{v}{2D} + \frac{dv}{dL} \right)$$



The capillary tube is further divided into many small segments to calculate pressure (Δp) and quality(x) alongside the capillary tube. The outlet quality and exit pressure is also obtained.

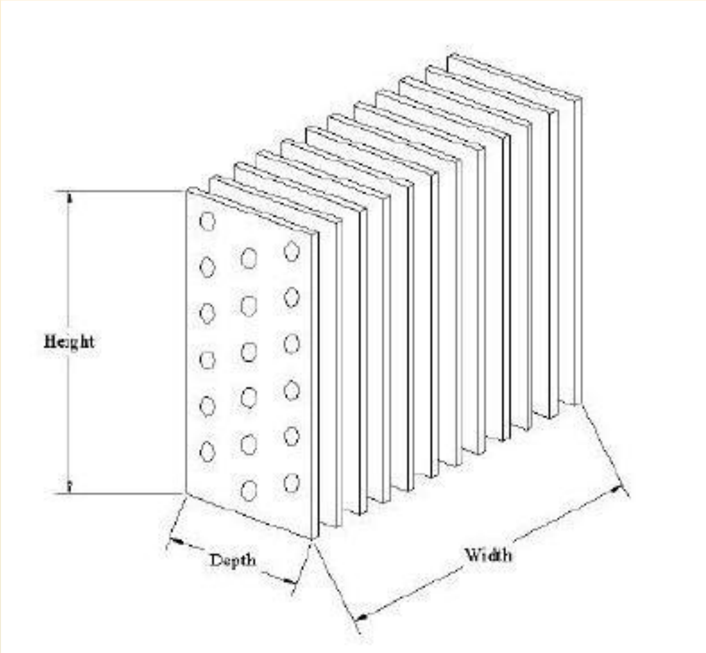
Evaporator Model

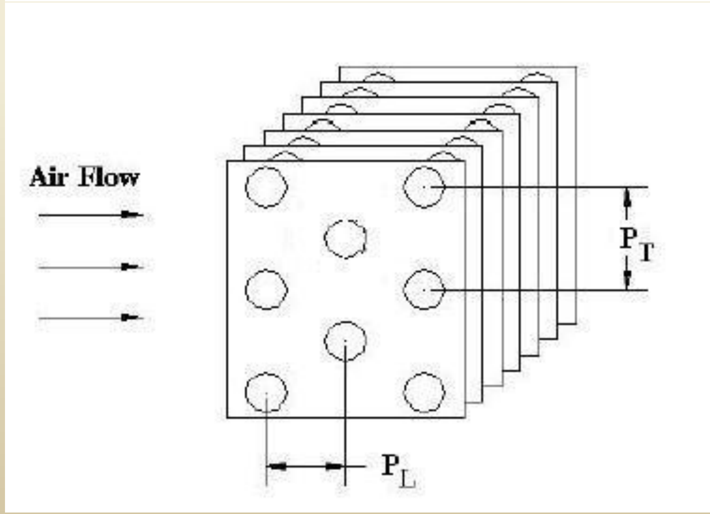
- Fin-tube heat exchanger in evaporator region. Fluid flow pattern is a counter-cross flow. CO_2 refrigerant is flowing in the tube and the air is flowing outside the tube.



Wet coil Analysis Method

Geometric parameters





Heat exchanger width	W
Heat exchanger height	H
Tube diameter	d_o
Fin type	Flat type
Pitch	F_p
Horizontal pitch	P_l
Vertical pitch	P_t
Fin thickness	δ_f
Wall material	Copper
Fin material	Aluminum
Tube row number	N
each row number of tube	Nt

Threlkeld (1970) for wet coil analysis:

$$Q_a = \dot{m}_a \left(i_{a,i} - i_{a,o}\right)$$ （ Air side ）

$$Q_c = \dot{m}_c C_{p,c} \left(T_{c,o} - T_{c,i}\right)$$ （ Refrigerant side ）

$$Q = U_{o,w} A_o F \Delta i_m$$

$$\Delta i_m = i_{a,i} + \frac{\left(i_{a,i} - i_{a,o}\right)}{\ln\left(\frac{i_{a,i} - i_{r,o}}{i_{a,o} - i_{r,i}}\right)} - \frac{\left(i_{a,i} - i_{a,o}\right)\left(i_{a,i} - i_{r,o}\right)}{\left(i_{a,i} - i_{r,o}\right) - \left(i_{a,o} - i_{r,i}\right)}$$

$i_{a,i}$, $i_{a,o}$: inlet and outlet of air enthalpy

$i_{r,i}$: corresponding saturated air enthalpy evaluated at the inlet refrigerant temperature

$i_{r,o}$: corresponding saturated air enthalpy evaluated at the outlet refrigerant temperature

Basic Equations

Two-phase region

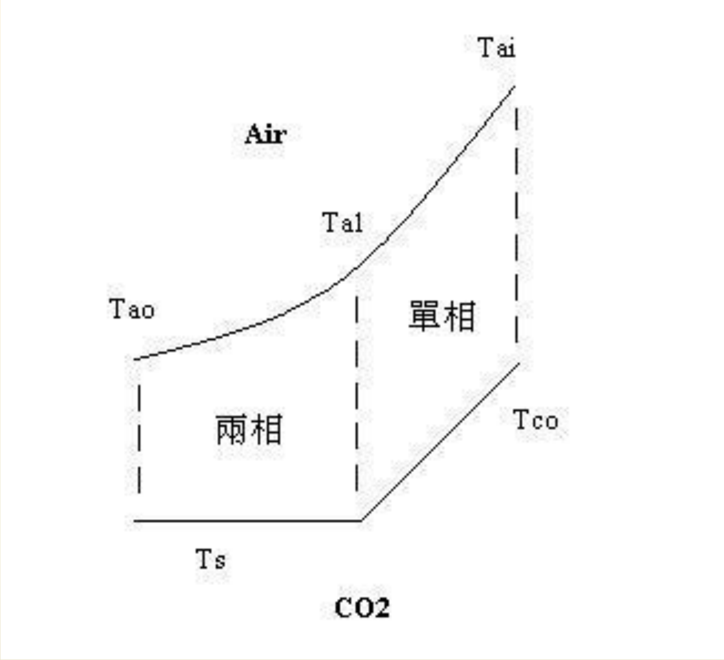
$$Q_{1c} = m_c \times i_{fg} \times (1 - x) = m_a \times (i_{a1} - i_{ao})$$

$$Q_{1c} = (UA)_1 \times (LMHD)_1$$

Single-phase region

$$Q_{2c} = m_c \times Cp_c \times (T_{co} - T_s) = m_a \times (i_{ai} - i_{a1})$$

$$Q_{2c} = (UA)_2 \times (LMHD)_2$$



Total heat exchange area $A_{total} = A_1 + A_2$

Evaporator

- The calculation of the air-side heat transfer coefficient
- Threlkeld (1970) Wet coil Analysis Method:

$$h_{o,w} = \frac{1}{\dfrac{c_{p,a}}{b'_{w,m} h_{c,o}}} = \left(\frac{c_{p,a}}{b'_{w,m} h_{c,o}} \right)^{-1}$$

$$b'_{w,m} = \frac{i \quad - i_{s,w,m}}{T \quad - T_{f,m}}$$

$$h_{c,o} = j_{wet} \times G_c \times C_{pa} \times \mathrm{Pr}_a^{-2/3}$$

Evaporator

– Refrigerant side

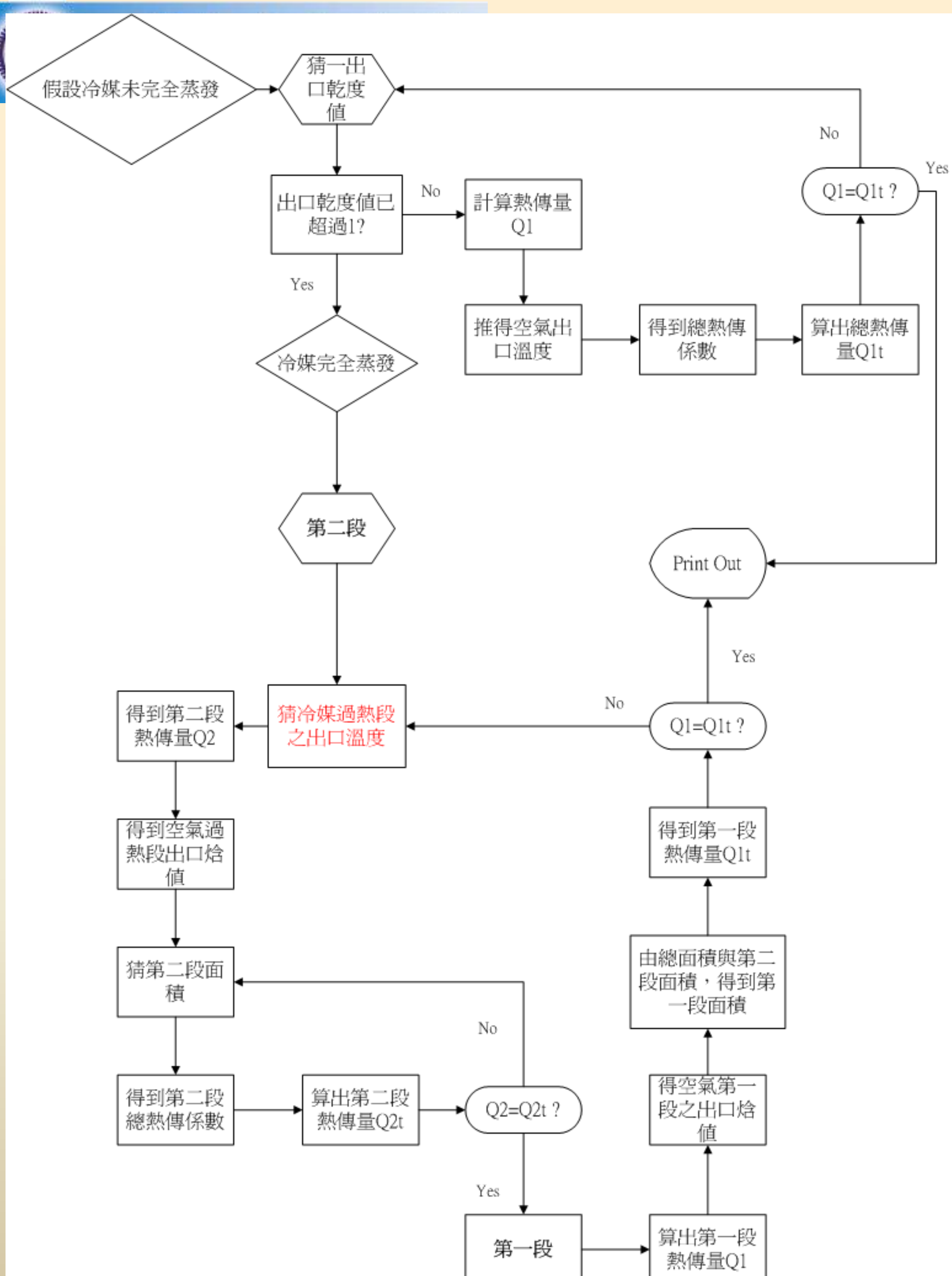
– Hihara and Tanaka (2000) correlation：

$$\frac{\alpha}{\alpha_{lo}} = K_1 Bo + K_2 \left(\frac{1}{X_{tt}}\right)^{2/3}$$

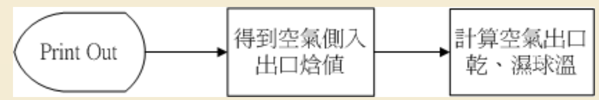
$$\alpha_{lo} = 0.023 \,\mathrm{Re}_l^{\,0.8}\, \mathrm{Pr}_l^{\,0.4} \frac{\lambda_l}{d_i}$$

$$K_1 = 1.4 \times 10^4 \quad , \quad K_2 = 0.93$$

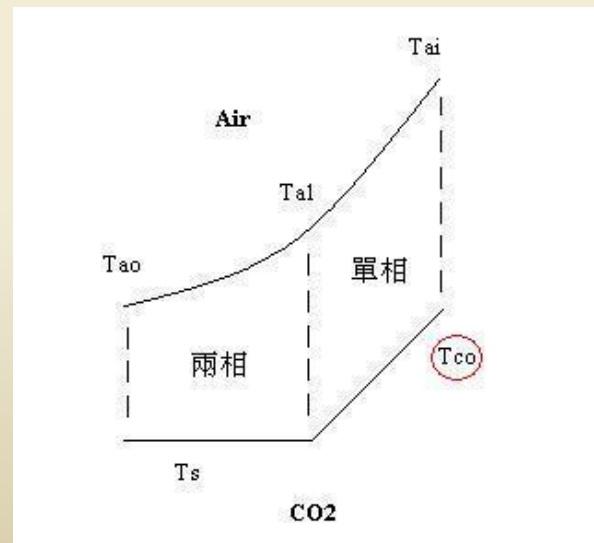
$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_g}{\rho_l}\right)^{0.5} \left(\frac{\mu_l}{\mu_g}\right)^{0.1}$$



假設冷媒未完全蒸發
猜一出口乾度值
出口乾度值已超過1?
No
Yes
計算熱傳量Q1
推得空氣出口溫度
得到總熱傳係數
算出總熱傳量Q1t
Q1=Q1t ?
No
Yes
冷媒完全蒸發
第二段
Print Out
猜冷媒過熱段之出口溫度
得到第二段熱傳量Q2
得到空氣過熱段出口焓值
猜第二段面積
得到第二段總熱傳係數
算出第二段熱傳量Q2t
Q2=Q2t ?
No
Yes
第一段
算出第一段熱傳量Q1
得空氣第一段之出口焓值
由總面積與第二段面積，得到第一段面積
得到第一段熱傳量Q1t
Q1=Q1t ?
No
Yes



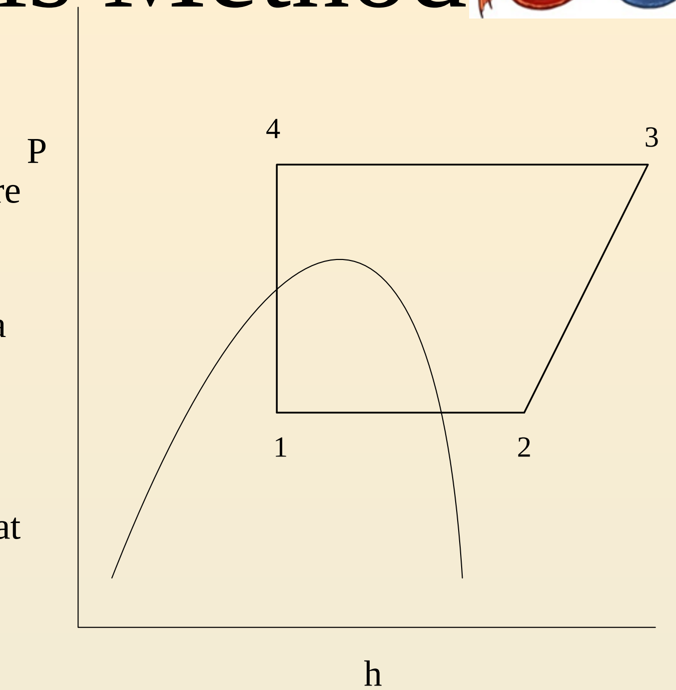
Print Out
得到空氣側入出口焓值
計算空氣出口乾、濕球溫



Tai
Air
Ta1
Tao
單相
兩相
Tco
Ts
CO2

Cycle Analysis Method

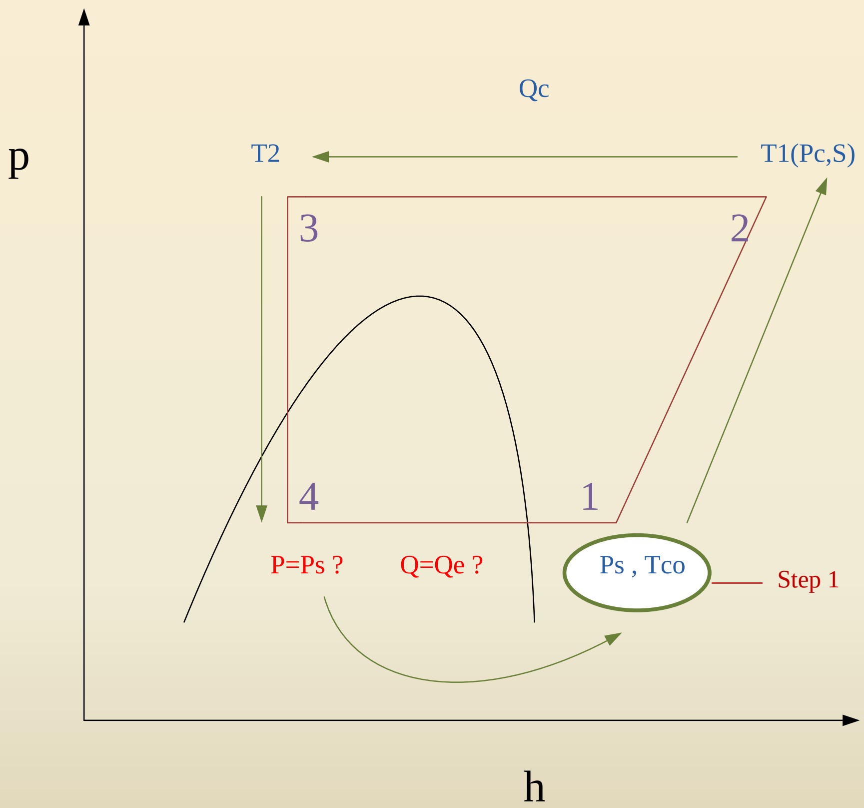
The first step is assuming the compressor inlet pressure, inlet temperature, outlet pressure. With these three values, one can calculate the mass flow rate of refrigerant. After 2 → 3 adiabatic process, the discharge compressor temperature can be obtained. 3 → 4 One can obtain a outlet temperature and enthalpy using the gas cooler model. Thus the exit state after capillary tube can be obtained via the 4 → 1 using capillary model. Checking whether the outlet pressure is the same pressure as the original guess. If it is different, we must re-assume the compressor outlet pressure. 1 → 2 Entering the evaporator and calculate the heat transfer rate with air. If it does not satisfied the energy balance equation in evaporation region, we should re-assume the initial value from the first step.



Input parameters
Gas cooler dimensions (d_i, d_o, D_i, L)
Evaporator dimensions (d_i, d_o, D_i,W, H...)
Expansion device dimensions (d_c, L_c)
Compressor data: Vs, N
Water inlet conditions (m_w, T_{wi})
Air inlet conditions (m_a, T_{db}, T_{wb})

The calculation process make use the following assumptions:

1. Heat loss is negligible
2. Steady state
3. No pressure drop in gas cooler & evaporator
4. Ignore the change of kinetic and potential energy in capillary tube



Input Evaporator dimensions, Gas cooler dimensions, Capillary tube dimensions, Compressor data, Air inlet parameters and water inlet parameters.

Initial gauss: compressor inlet pressure

Gauss: evaporator outlet temperature

Gauss: compressor outlet pressure

Calculate: refrigeration mass flow rate

Calculate: gas cooler outlet conditions

Calculate: capillary tube outlet conditions

Capillary tube outlet pressure = compressor inlet pressure?

No

Update compressor outlet pressure

Yes

Evaporator heat balance?

No

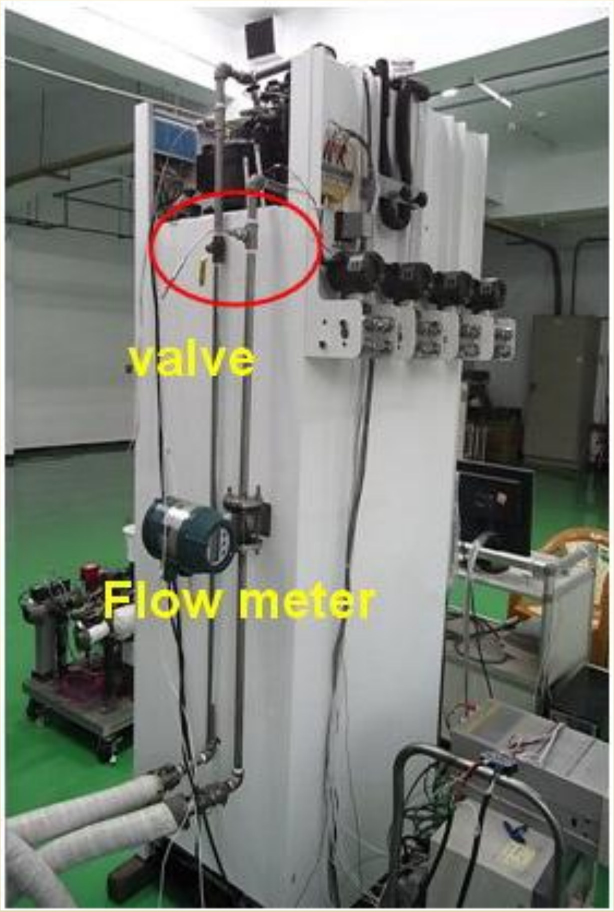
Update compressor inlet conditions

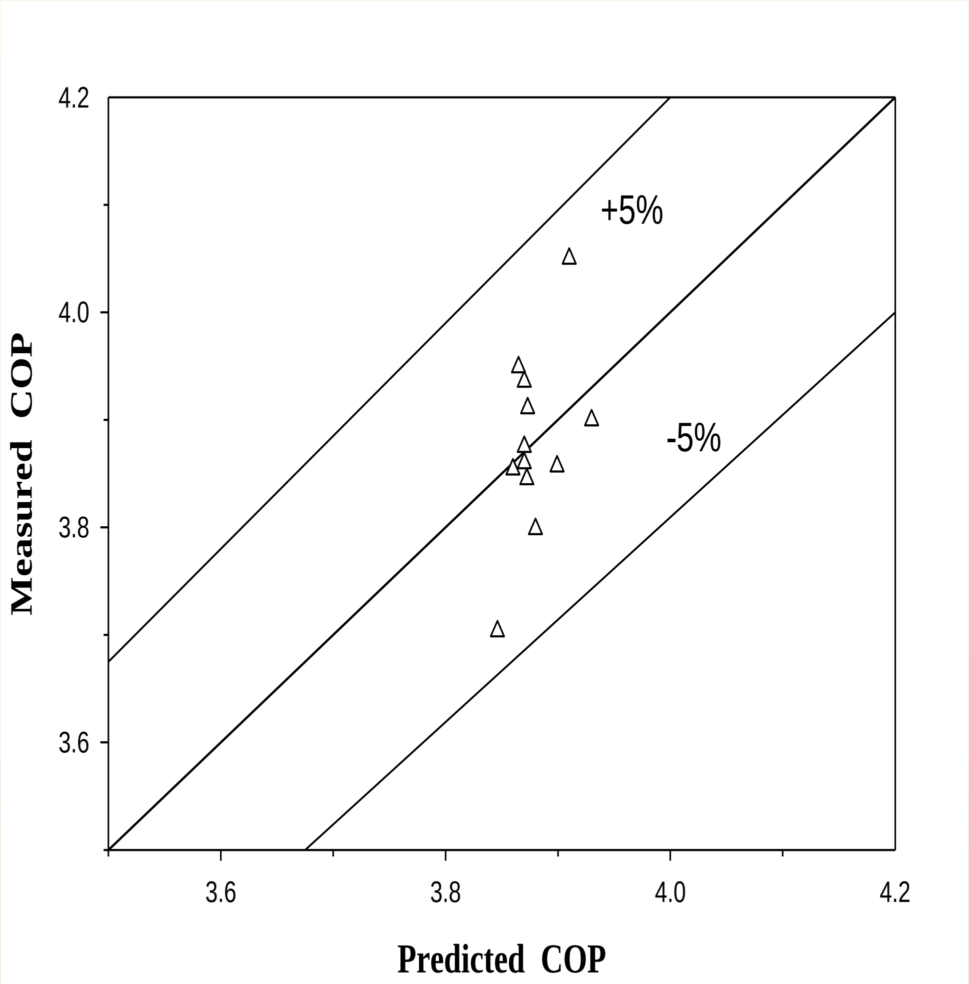
Yes

Outputs: state points, COP

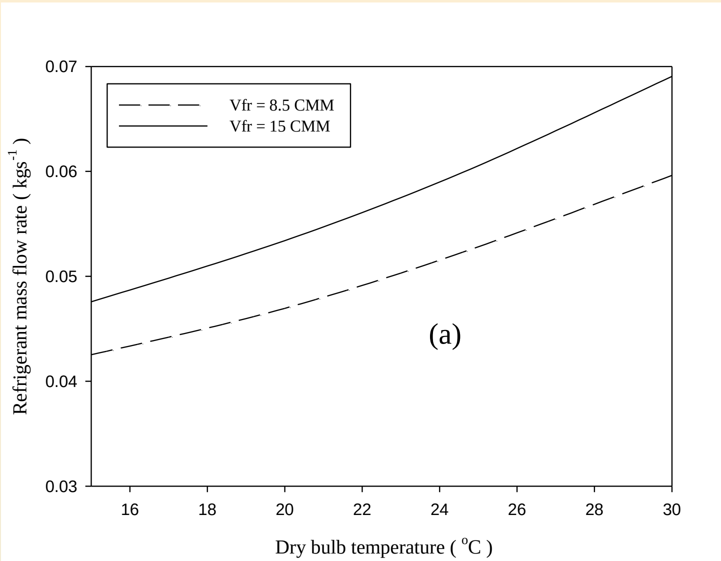
Simulation results

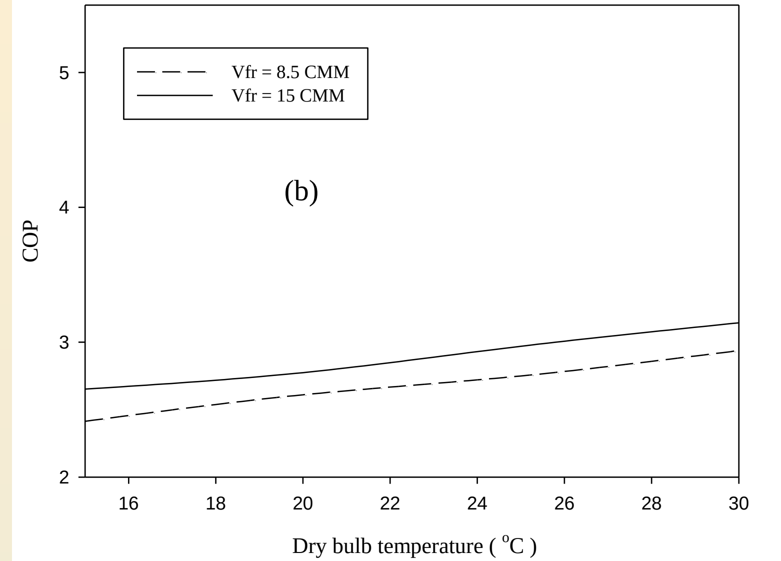
Predictions vs. Measurements

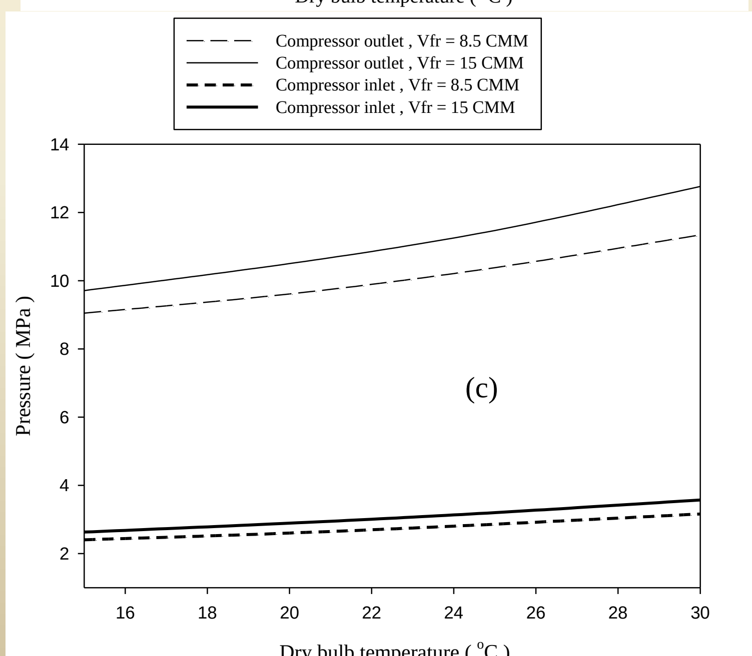


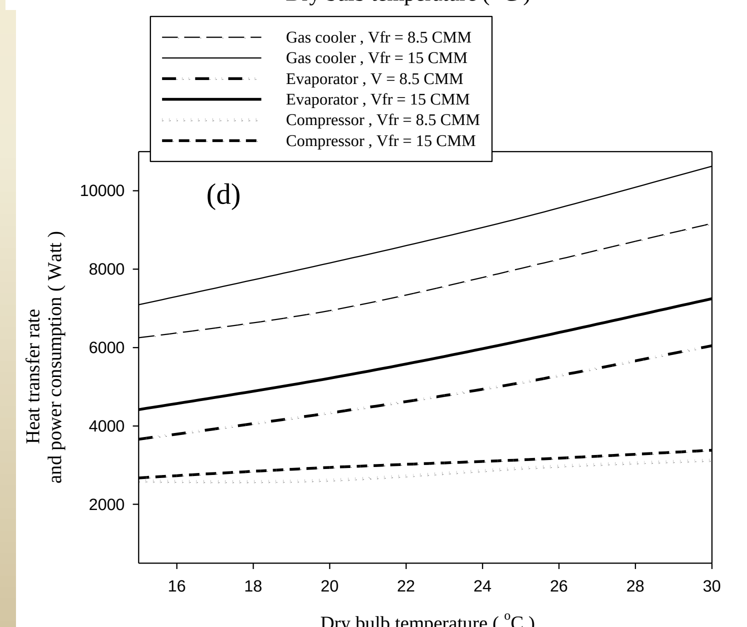


Effect of the dry bulb air temperature on (a) refrigerant mass flow rate (b) COP (c) pressure (d) heat transfer rate. With RH=50%, Twi=20℃, Mw=0.08kgs^{-1}, Lcap=2.0m, d,cap=1.4mm

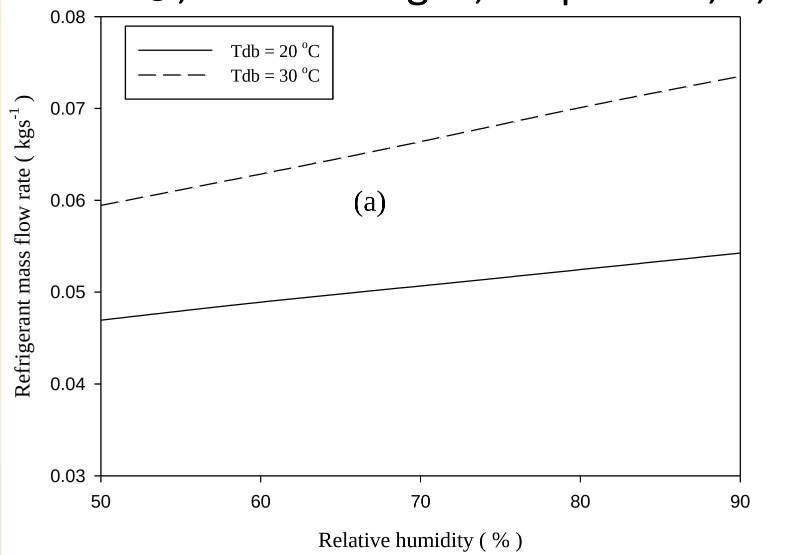


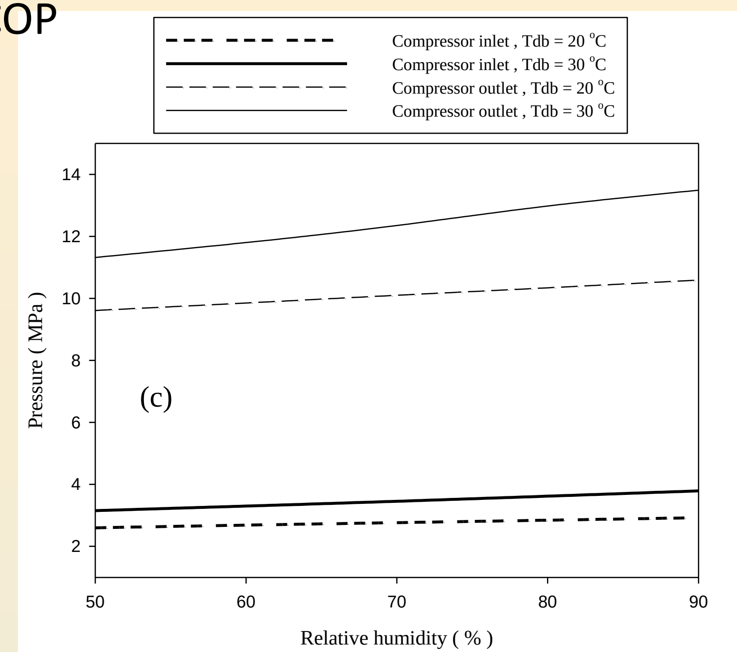


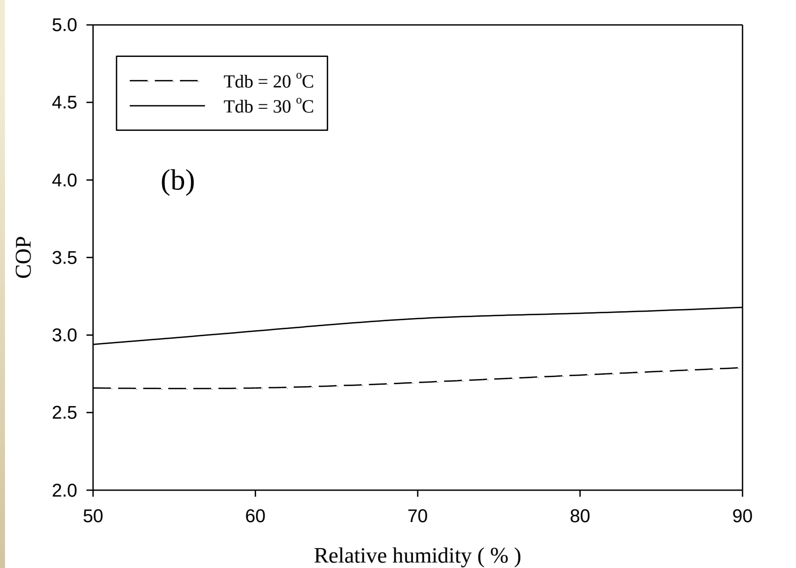


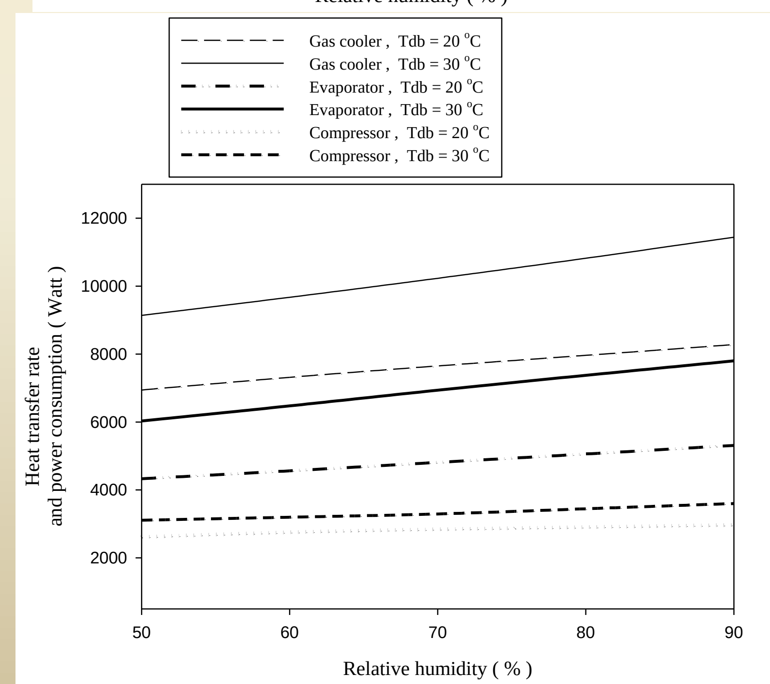


Effect of the RH on the (a) refrigerant mass flow rate (b) COP (c) pressure (d) heat transfer rate. With V = 8.5 CMM, Twi=20℃, Mw=0.08kgs^{-1}, Lcap=2.0m, d,cap=1.4mm,

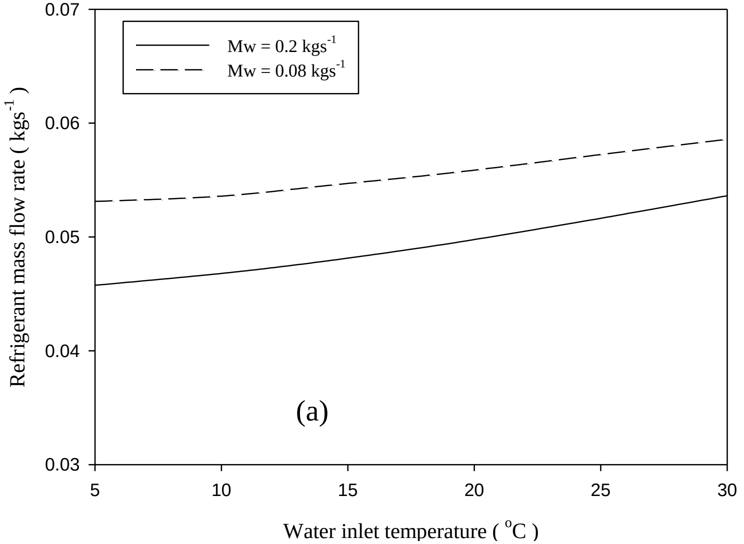


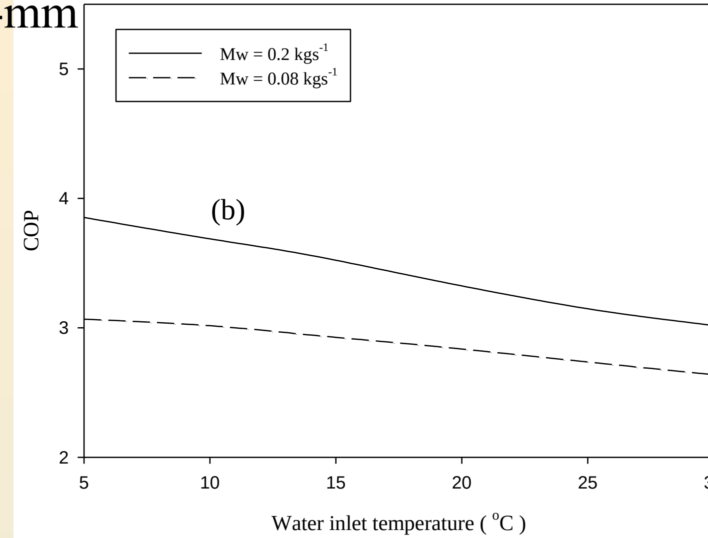


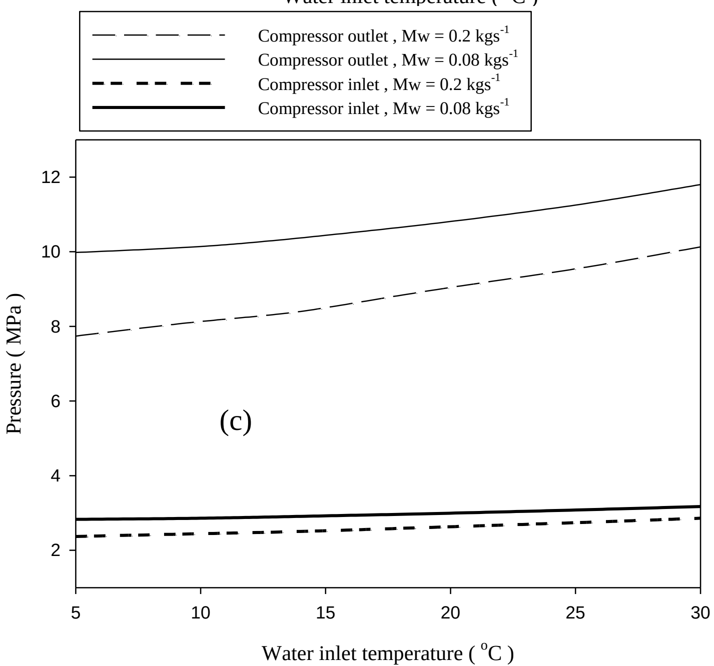


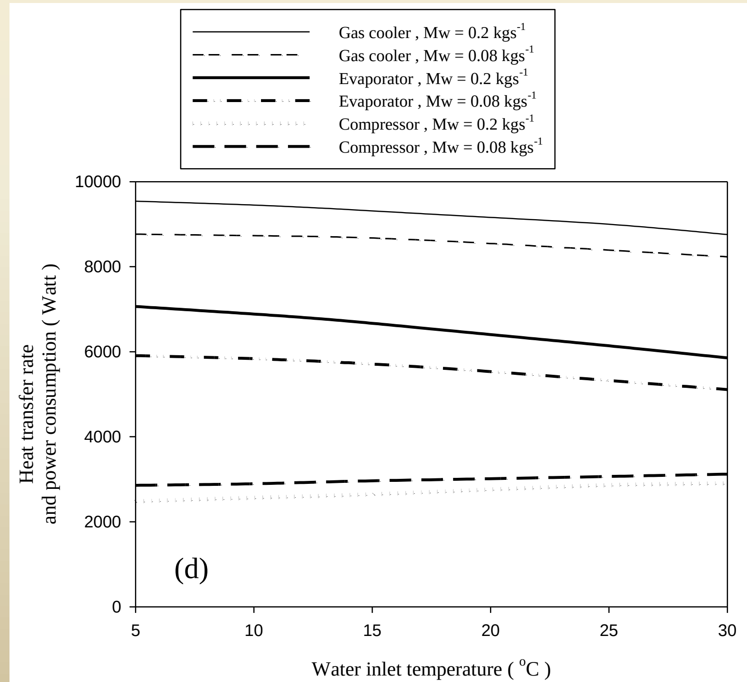


Effect of water inlet temperature on (a) refrigerant mass flow rate (b) COP (c) pressure (d) heat transfer rate. With Tdb=27℃, Twb=20℃, Vfr=8.5CMM, Lcap=2.0m, d,cap=1.4mm

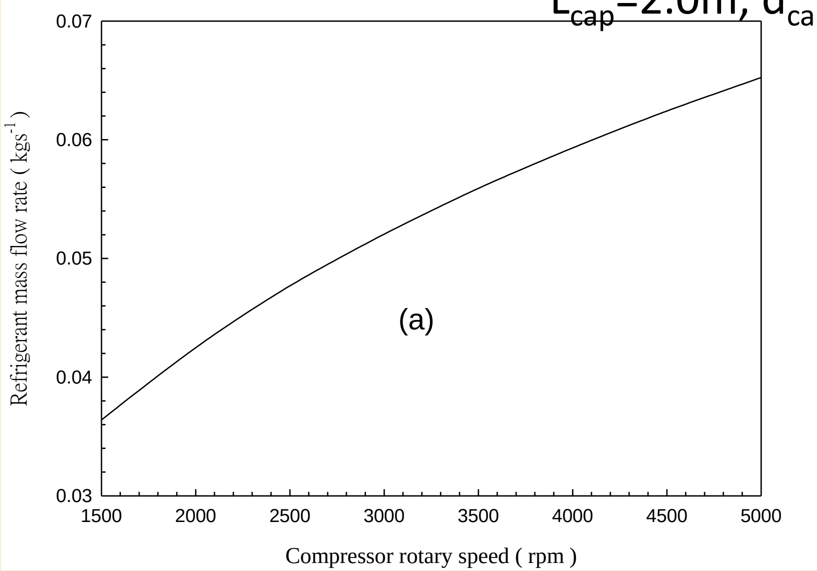


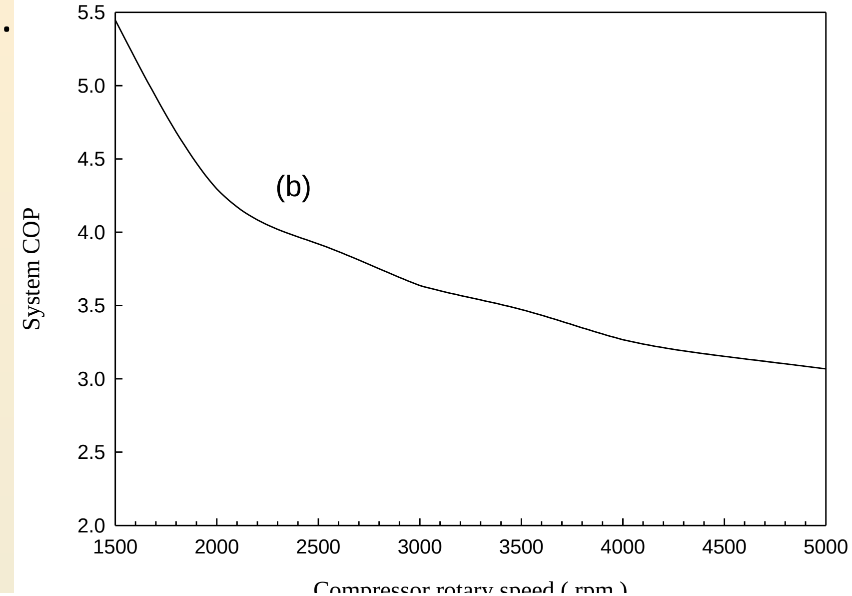


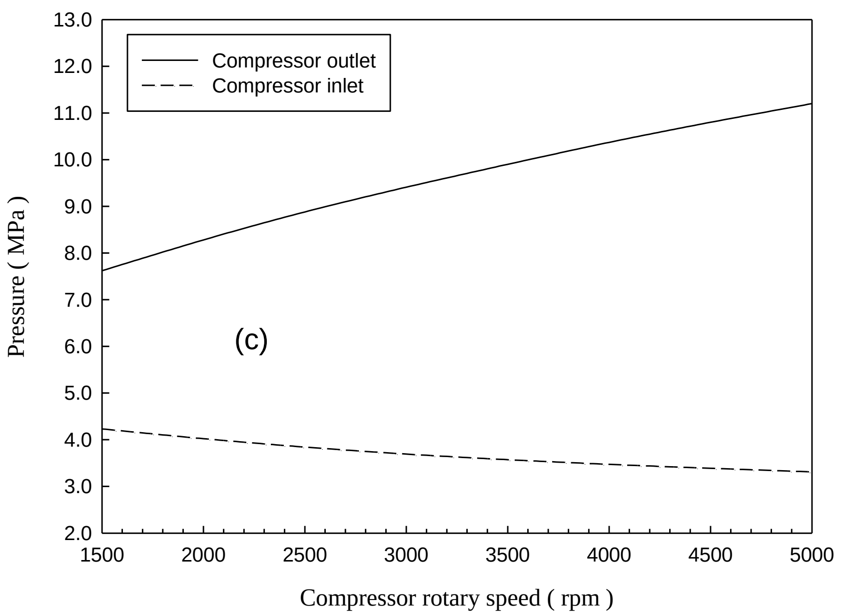


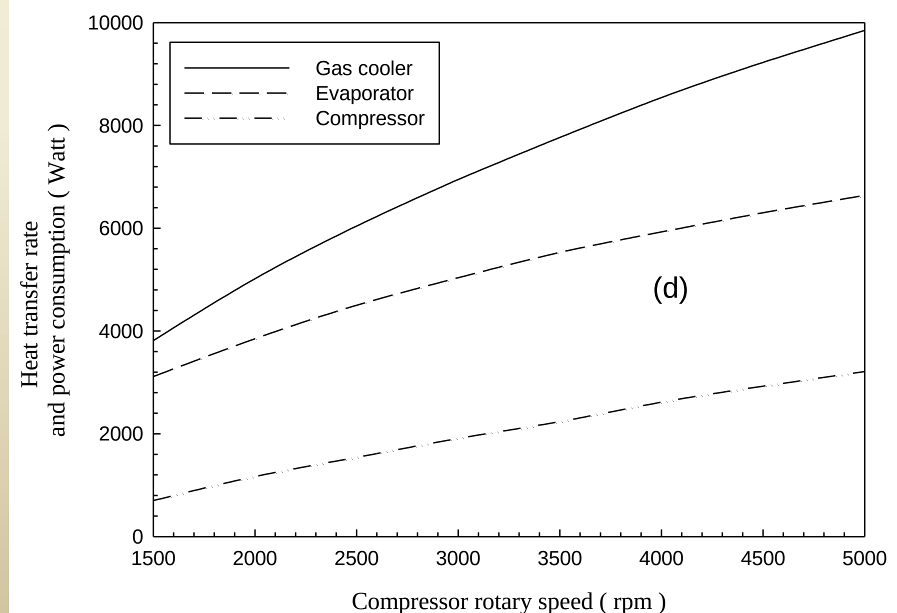


Effect of compressor speed on (a) refrigerant mass flow rate (b) COP (c) pressure (d) heat transfer rate. With T_{db} = 27℃, Twb=20℃, V=8.5CMM, Twi=10℃, Mw=0.2kgs^{-1}, L_{cap}=2.0m, d_{cap} = 1.4mm.

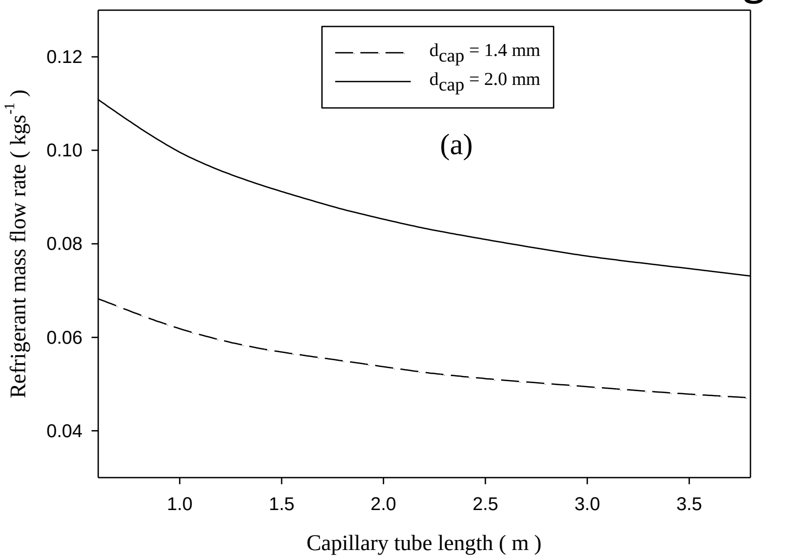


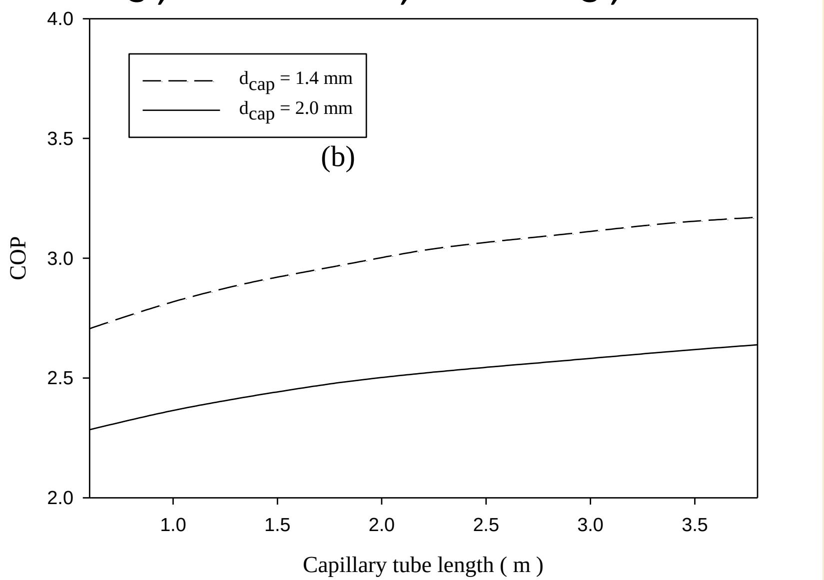


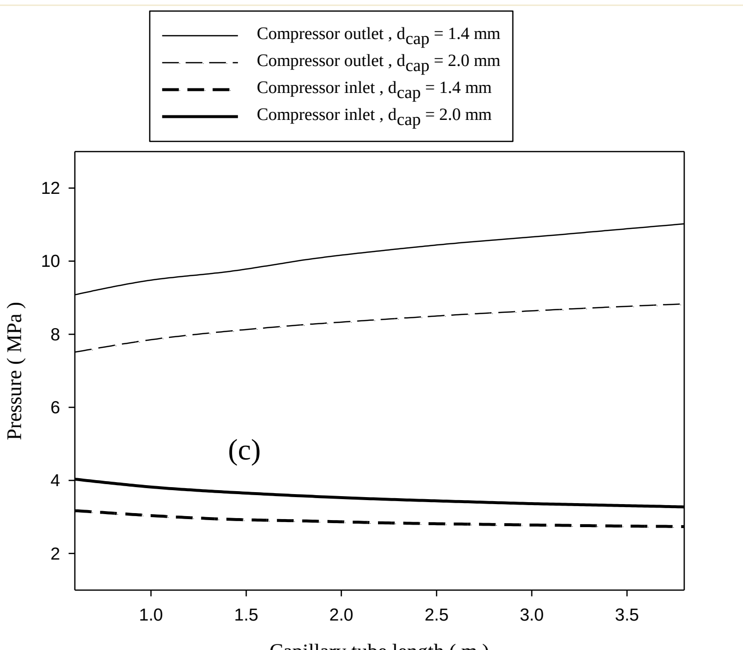


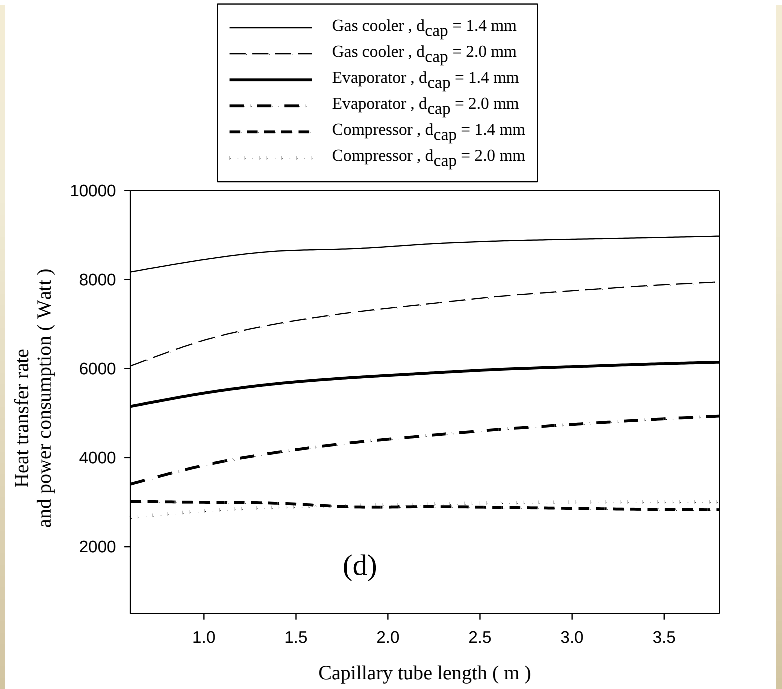


Effect of the capillary tube length on (a) refrigerant mass flow rate (b) COP (c) pressure (d) heat transfer rate. With Tdb=27℃, Twb=20℃, V=8.5CMM, Twi=10℃, $Mw=0.2kgs^{-1}$









Conclusions

- This study develops a simulation program of CO_2 refrigerant cycle system. The simulation program is capable of handling the variation of indoor conditions, gas cooler, compressor speed, and geometry of capillary tube without any prescribed conditions (i.e. fixed evaporation temperature, fixed condensing temperature, and the like).
- In gas cooler, the CO_2 may present a local minimum and a local maximum along the length of the heat exchange, provided CO2 passes through the pseudo-critical temperature, and this phenomenon becomes more and more pronounced when the pressure is close to the critical pressure
- The RH of the indoor air is an important factor affecting the overall system performance.
- Increasing the water side inlet temperature or increasing the compressor speed will decrease COP of the system.
- An increase of the length of capillary tube also increases the COP moderately.

Thanks for your attention

Major Graduate Student in this project: Kai-Hsiang Lin

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