Transcritical Carbon Dioxide Based Heat Pumps: Process Heat Applications

Jahar Sarkar
Indian Institute of Technology

Souvik Bhattacharyya
Indian Institute of Technology

M. Ram Gopal
Indian Institute of Technology

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Jahar SARKAR, Souvik BHATTACHARYYA*, M. RAMGOPAL

Department of Mechanical Engineering, Indian Institute of Technology
Kharagpur, India 721302

*91-3222-282904, souvik@mech.iitkgp.ernet.in

ABSTRACT

Ozone layer depletion has caused most of the conventional refrigerants used in high temperature heat pumps to be phased-out recently, and a suitable synthetic refrigerant is yet to be found. Several natural refrigerants such as hydrocarbons, water, and carbon dioxide have been presented as alternatives. In this context, environmentally benign transcritical CO_2 cycle based heat pumps offer widespread possibilities in process heat applications due to the large temperature glide present in the gas cooler and can be extensively used in food and dairy, chemical, textile, paper, sugar and other industries where process heat is a critical input.

A detailed performance analyses of heat pump systems have been carried out for several types of process heat applications with various possible heat sources. Important performance characteristics of the transcritical CO_2 process have been reviewed along with their optimization aspects. Performance based comparisons have been made between the conventional heat pump systems and CO_2 based ones. Suitability of the CO_2 system in specific process application has been reported; this arises because a particular process application imposes specific temperature and pressure requirements. Results show that although the CO_2 heat pumps give marginally lower COP (5% to 25%) compared to other conventional refrigerants in industrial heat pumps, they offer considerably lower pressure ratios (3 to 4 times) and higher volumetric capacity. It is observed that CO_2 systems have strong potential in process heating and cooling (refrigeration / water cooling / air conditioning / process cooling). It is expected that CO_2 heat pumps will be viable alternatives for process heat application due to its eco-friendliness, higher volumetric capacity and good heat transfer properties.

1. INTRODUCTION

Energy conservation efforts have prompted low-grade energy supplies such as waste heat, geothermal energy, etc. to be effectively employed to produce desired process heat using heat pump systems. Heat pumps are used for residential applications such as water and space heating, and a large number of industrial applications as well, such as chemical, food, paper plants, etc., where the process heat is required in the form of hot water or steam and hot air. The two most common types of heat pumps are the mechanical vapor compression system and the heat driven vapor absorption system. A reversed absorption heat pump, also known as a heat transformer, can increase the temperature of part of a stream of waste heat while simultaneously lowering the temperature of the rest of the stream. The current study focuses on the vapor compression system. The choice of refrigerants for heat pumps is a multi-criteria selection process dependent on thermodynamic, safety and technical aspects. Although a number of conventional refrigerants are found suitable for low temperature (40-70 ^{o}C) heating applications such as cleaning, drying, water heating, only a few working fluids have been tested for medium and high temperature applications (> 70 ^{o}C), which is addressed in this study. For a condensing temperature of up to 120 ^{o}C, R114 seemed to be an appropriate solution (Moser and Schniter, 1985). For very high temperature requirements, water will be a suitable fluid although its molecular weight is very low, and hence the specific volume is high. Between 120 ^{o}C and 170 ^{o}C none of the proposed working fluids seems to be an acceptable choice. Since the discovery of harmful effects of the synthetic refrigerant R114 on environment, many alternative fluids with zero ODP such as HFC-143, HFC-236ca, HFC-236cb, HFC-236ea, HFC-236fa, HFC-254cb and HFE-134 have been proposed (Devotta, 1995). Of these, HFC-143 performs the best although it is flammable (Devotta and Pendyala, 1994). Several other hydrofluorinated ethers such as HFE-143, HFE-245, HFE-245cb, HFE-245fa and HFE-254cb have also been proposed for high temperature heat pump applications (Goktun, 1995). However, typically characterized by high NBP, they are not suitable for low evaporator temperatures. As a result, presently effort is underway to develop non-azeotropic mixtures by combining these high temperature fluids with low temperature fluids such as R22 (negligible ODP). Quite a few refrigerant mixtures such as R22/R114, R22/R141b, R22/R42b, R22/R152a, R22/R123, etc. (Li *et al.*, 2002) have been proposed. But all these mixtures are synthetic and have very high GWP, although ODP values are negligible. This has led to the development of eco-friendly natural refrigerants such as ammonia, propane and carbon dioxide. Among these, ammonia based heat pumps are used for only low temperature process heat applications due to its low critical temperature. Propane is unsuitable for its flammability. That leaves us with CO_2 as the only suitable option, although water, in spite of its deficiencies, is a good candidate for very high temperature requirements.

Table 1. Comparison of refrigerants for heat pump applications.
(Lorentzen, 1994; Devotta and Pendyala, 1994; Devotta, 1995; Goktun, 1995)

Refrigerant	Formulas	Flammability	NBP (^oC)	P_c (bar)	t_c (^oC)	ODP	GWP
Natural substances							
R744	CO_2	NF	-------	73.72	31.10	0.0	1(0)
R717	NH_3	F*	-33.3	113.5	133.0	0.0	0.0
R600a	C_4H_{10}	F	-11.7	36.50	135.0	0.0	3.0
Hydrofluorocarbons HFCs							
R143	$C_2H_3F_3$	F	5.00	42.33	157.8	0.0	NA
R152	$C_2H_4F_2$	SF	30.70	43.30	202.8	0.0	NA
R236ca	$C_3H_2F_6$	NF	5.00	32.99	138.9	0.0	NA
R236cb	$C_3H_2F_6$	NF	-1.22	31.18	130.1	0.0	NA
R236ea	$C_3H_2F_6$	NF	6.50	35.33	141.1	0.0	NA
R236fa	$C_3H_2F_6$	NF	-1.11	31.77	130.6	0.0	NA
R245ca	$C_3H_3F_5$	NF	25.0	36.60	178.4	0.0	NA
R254cb	$C_3H_4F_4$	UC	-0.78	37.53	146.1	0.0	NA
Hydrofluorinated ethers HFEs							
E134	$C_2H_2F_4O$	UC	4.67	37.5	153.4	0.0	NA
E143	$C_2H_3F_3O$	SF	30.1	41.4	186.6	0.0	NA
E245	$C_3H_3F_5O$	NF	80.6	36.2	224.6	0.0	NA
E245cb	$C_3H_3F_5O$	SF	34.1	34.2	185.2	0.0	NA
E245fa	$C_3H_3F_5O$	NF	29.2	33.7	171.0	0.0	NA
E254cb	$C_3H_4F_4O$	SF	36.4	35.6	189.4	0.0	NA
F: Flammable, SF: Slightly flammable, UC: Uncertain, NF: Non-flammable, NA: Not available							

*Its lower ignition limit is as high as 15.5% by volume, 3-7 times that of common hydrocarbons.

Although CO_2 was used long back, it was phased out during the 1930s due to the development of synthetic refrigerants and it is now witnessing a revival since 1994. This has inspired subsequent development of transcritical CO_2 cycles where the condenser gets replaced by a gas cooler and in which heat rejection occurs in the supercritical regimes due to very low critical temperature (31.2 ^oC) of CO_2. CO_2 based heat pumps offer extensive possibilities in simultaneous process heating and cooling applications due to the large temperature glide present in the gas cooler. Beside zero ozone depletion potential (ODP) and zero effective GWP, CO_2 has several advantages over other refrigerants, such as compatibility with normal lubricants and common machine construction materials, non-flammability, non-toxicity, low compression ratio, high volumetric capacity, easy availability and very low price. Different applications of CO_2 heat pumps were presented by Neksa (2002), which included water heating, space heating, as a residential heat pump, in air heating systems, as a heat pump dryer and high temperature hydronic heating systems. Experimental results of CO_2 heat pumps employed in residential heating (Richter *et al.*, 2003) and in drying applications (Klocker *et al.*, 2001) have been presented. Although the CO_2 heat pump water heater has been marketed recently, extensive industrial application is still being studied and evaluated. Due to a lower critical temperature, a CO_2 heat pump is not suitable for high temperature heat sources and due to the higher boiling point some of the HFCs and HFEs (Table 1) are not suitable for low temperature heat sources. So, HFC-143, HFC-236ea, HFC-236fa, HFE-134 and non-azeotropic mixture R22/R141b (Li *et al.*, 2002) are chosen for a comparison with CO_2. Although the mixture R22/R141b presents a nonzero ODP choice, this type of mixtures is suitable for heat pump applications because of its gliding temperature in the condenser. In the present work, a detailed thermodynamic and heat transfer based comparative analyses of CO_2 with respect to these refrigerants have been carried out for several types of process heat applications. Performance and limitation of CO_2 heat pumps in high temperature process heat application and simultaneous heating and cooling application are also presented.

2. ANALYSIS FOR HEAT PUMP SYSTEMS

In CO_2 systems, a gas cooler instead of a condenser is used to produce heat for process heat applications; the secondary fluid may be water or air, and the evaporator is exposed to the heat source or used for cooling (refrigeration / water cooling / air conditioning / process cooling) applications. Both the heat exchangers are counter flow ones where refrigerant flows through the inner tube and secondary fluid flows in the outer annulus. Performance parameters such as heating COP and exergetic efficiency are given by:

$$COP_{heating} = \frac{q_{gascooler/condenser}}{w_{compressor}} \tag{1}$$

$$\eta_{II,heating} = COP_{heating} \Big/ (COP_{heating})_c \tag{2}$$

Following assumptions are considered: single-phase heat transfer for the external fluid, compression process is adiabatic and isentropic, evaporation and compression processes are isobaric. For all refrigerants, average heat source temperature has been taken as 30 ^{o}C, so the evaporator temperature is 20 ^{o}C by assuming an average temperature difference of 10 ^{o}C in the evaporator. Internal heat exchanger effectiveness is taken as 60% for the analysis. For the thermodynamic and transport properties of CO_2, an exclusive code *CO2PROP* based on available correlations (Span and Wagner 1996, Vesovic *et al.*, 1990), has been developed and employed, and for other refrigerants *REFPROP* (version 5/6.01) has been used. For the CO_2 system an optimum discharge pressure was assumed (Sarkar *et al*.., 2004):

$$P_{opt} = 4.9 + 2.256t_3 - 0.17t_6 + 0.002t_3^{\ 2} \tag{3}$$

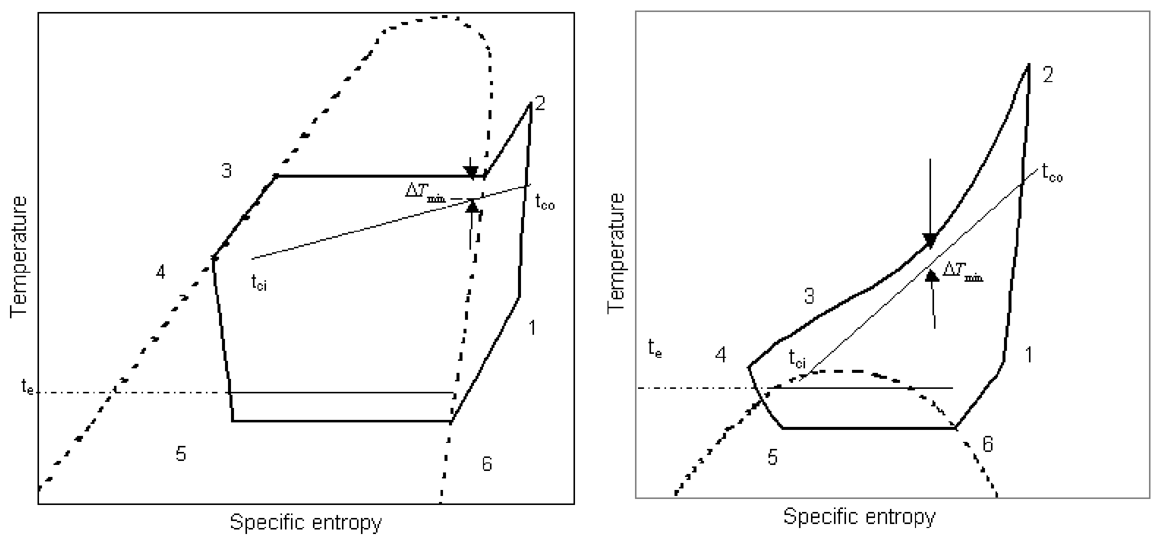

Figure 1: T-s diagrams of heat pump cycles for (a) conventional refrigerant and (b) CO_2

For the gas cooler or condenser side analysis, the refrigerant and secondary fluid are in counter flow arrangement and a minimum temperature difference of 5 ^{o}C has been assumed. To confirm this, for the CO_2 system the whole temperature range is divided into 50 sub ranges; thus at any point within the 50 elements, the temperature difference between refrigerant and secondary fluid should be more than or equal to 5 ^{o}C. For other systems, it is obvious that the minimum temperature will occur at point 2 (or t_{co}) or the saturated vapor point (pinch points as in figure 1), thus the condition is mathematically represented by,

$$\min\left[(t_2 - t_{co}), t_{cr} - \left(t_{ci} + \frac{h_v - h_3}{h_2 - h_3}(t_{co} - t_{ci})\right)\right] = 5^oC \tag{4}$$

3. PROCESS HEAT APPLICATIONS

Basic processes where a heat pump is used to provide process heat include evaporation, distillation, drying, blanching and pasteurization. Process heat is required in the form of hot air, hot water, saturated or superheated steam and direct heating in wide temperature ranges. The applicability of refrigerants is dependent on the critical temperature for all conventional cycles, but for CO_2 systems limitation is the gas cooler pressure. Generally a heat pump is used upto 100–120 °C, and beyond that a supplementary heater is used. Due to a temperature glide in the gas cooler, a CO_2 heat pump is useful upto 100 °C for water at atmospheric pressure, but for air or in direct heating mode higher temperatures can be used. The heat source includes ambient air, river or ground water, cooling water from engine, compressor or condenser of refrigeration systems or power plant, etc. Due to the low critical temperature, high temperature heat sources are not suitable for CO_2 systems. Here, comparisons of some applications are presented for three outlet temperatures of 80 °C, 100 °C and 120 °C and for an evaporator temperature of 20 °C.

Table 2: Performance comparison

Refrigerant	CO_2	R143	R236ea	R236fa	E134	R114	R141b/R22
Condition	t_{co} = 80 °C						
P_{ratio}	1.950	5.443	4.435	5.026	5.560	4.901	3.472
T_{dis} (°C)	90.48	118.6	85.44	96.24	107.5	97.52	106.4
VC (kJ/m^3)	21058	1773.8	1492.2	1765.6	1557.9	1420.9	3749.3
COP	5.073	5.153	5.794	4.773	5.008	4.854	6.156
η_{II} (%)	38.67	39.27	44.16	36.38	38.17	37.00	46.93
	t_{co} = 100 °C						
P_{ratio}	2.238	7.433	7.562	6.668	7.605	6.626	4.753
T_{dis} (°C)	107.2	140.1	114.4	112.5	127.2	116.0	132.8
VC (kJ/m^3)	20920	1709.5	1341.6	1634.2	1478.6	1336.5	3579.5
COP	4.110	4.157	3.812	3.789	3.994	3.825	4.645
η_{II} (%)	42.56	43.05	39.47	39.24	41.36	39.61	48.10
	t_{co} = 120 °C						
P_{ratio}	2.575	9.617	9.721	8.184	9.771	8.429	6.004
T_{dis} (°C)	125.3	158.7	129.3	125.1	144.2	131.3	153.9
VC (kJ/m^3)	20784	1650.1	1256.9	1530.9	1409.6	1256.6	3437.7
COP	3.414	3.533	3.196	3.191	3.373	3.205	3.848
η_{II} (%)	44.12	45.69	41.48	41.43	43.61	41.54	49.76

3.1 Food processing

The possible applications of heat pumps in the food and beverage industries are breweries, dairy, sugar refineries, grain drying and canning units. Most likely processes are pasteurization, sterilization, evaporation, blanching and drying. In most of the food processing applications, hot water at 70 °C is required for blanching. Pasteurization requires hot water at about 80 °C, sterilization requirement is of 121 °C, bottle cleaning requires hot water at 80 °C and various washing process requires hot air. In milk powder production units, air at 70 °C is first produced by a heat pump and then the hot air is further heated to 210 °C by an external heater for drying. Thus, it can be observed that in most cases, NH_3 would not be a suitable refrigerant. For all the applications, CO_2 will yield better performance than the other fluids, except R143 and R141b/R22 (Table 2). Actual COP of a CO_2 system is expected to be superior to other synthetic refrigerants because of a characteristic 3-4 times higher heat transfer coefficient (Table 3). The other refrigerants are obviously advantageous in steam production, although high temperature (more than 100 °C) heating water is possible by increasing pressure (up to 120 °C at 2 bar). It is obvious that, if air is heated to temperatures higher than 70 °C upstream of the air heater in a milk powder production system, performance will improve and for a CO_2 system this is quite feasible. In a sugar plants, several operations such as distillation, crystallizations, etc, require hot water and steam at difference temperature ranges (Bayrak *et al*., 2003). A CO_2 heat pump can also be used in drying facilities requiring hot water and hot air.

3.2 Pulp and paper industry

In paper manufacturing, several processes such as pulp digesting and bleaching, black liquor evaporation and paper drying require large amount of process heat. For drying, hot air at upto 70 ^oC is required; previously R12 was commonly used in heat pumps for this purpose. After the phase out, some mixtures have become popular. It is observed that for drying processes, CO_2 heat pumps can be effectively used. For evaporation, steam at 115 ^oC is required and R114 has been a very popular choice. Since the phase-out of R114, the replacements exhibit better performance, except R236fa, which results in a marginal loss in efficiency. For steam production, CO_2 based heat pumps are not recommended due to its higher exergy loss in the gas cooler and hence, these substitutes (R143, R236ea, R236fa, E134) offer a better choice for steam production. Refrigerants with very high critical temperature are useful for steam generation.

3.3 Chemical and petrochemical industry

Processes that require large amount of process heat are evaporation, distillation and drying. Heat pumps are also applicable in chemical reactors. Petroleum refinery systems require process heat in the form of steam upto a pressure of 5 bar and at a temperature of about 135 ^oC. There are many other processes such as propane-propylene splitter, propane-butane separator, polymerization, crystallization, etc, where hot water or steam at a temperature of 120 ^oC is required. Vapor compression heat pumps or absorption heat transformers or combined vapor compression heat pumps and organic-Rankine cycle are used in chemical processes. Eisa (1996) reported that both vapor compression and heat driven absorption heat pumps are suitable for distillation processes.

3.4 Other process applications

There are various proven industries where heat pump is required and there include plastics, agricultural, textile, ceramic, plaster, malting, etc. Moser and Moser (1985) described in detail a wide variety of other process applications of heat pumps where process heat is required at a wide range of temperatures. CO_2 heat pumps perform better when an open loop coolant system is employed.

It is obvious that the CO_2 system can be suitably employed in some limited applications where conditions such as open coolant loop, low temperature heat source and low temperature coolant inlet are satisfied. Merits include better performance, lower compressor discharge temperature, higher volumetric capacity, good heat transfer properties and obviously environmental impact factors (others have higher GWP). However in process heat applications where high temperature heat sources (i.e., power plant condenser, engine cooling water, etc) or closed cycles (means, high inlet temperature) are used and for steam production, CO_2 systems are not suitable.

4. HEAT TRANSFER CHARACTERISTICS

Heat transfer characteristic of the refrigerants is one of the important criteria for system performances. Better heat transfer properties will help to reduce heat exchanger size for same capacity and will ultimately yield a compact system. On the other hand, for the same heat exchanger, due to increase in heat transfer rate the required temperature difference between two fluids will be less, yielding higher exergetic efficiency. For a heating output of 100 kW and heating outlet temperature of 100 ^oC, the heat transfer coefficient for different refrigerants are given in the Table 3 at three state points: (a) in the evaporator (saturation temperature of 20 ^oC), (b) saturated liquid at condensing pressure, (c) compressor outlet. Heat transfer correlations have been used assuming constant thermo-physical properties (Kakac and Lui, 2002):

$$\alpha = \alpha_l \left(1 + 1.925 X_{tt}^{-0.83}\right); \alpha_l = 0.03 \frac{k_l}{d_i} \mathrm{Re}_l^{0.8} \mathrm{Pr}_l^{0.4} \quad \text{(Wattelet correlation for evaporator)} \tag{5}$$

$$\alpha_l = 0.05 \frac{k_l}{d_i} \mathrm{Re}_l^{0.8} \mathrm{Pr}_l^{0.33} \quad \text{(Cavallini \& Zecchin equation for condenser)} \tag{6}$$

$$\alpha = \frac{k}{d_i} \frac{(f/2)\mathrm{Re}\,\mathrm{Pr}}{1.07 + 12.7(f/2)^{1/2}(Pr^{2/3} - 1)} \quad \text{(Singe phase heat transfer)} \tag{7}$$

where, friction factor, $f = \left(1.58\ln(\mathrm{Re}) - 3.28\right)^{-1}$ (8)

In equation (5), X_{tt} is the Lockhart–Martinelli parameter. For simplicity, refrigerant heat transfer coefficients in the evaporator corresponding to the saturated liquid are presented in table 2. Equations (7) and (8) (Petukhov-Kirillov) are valid for $\mathrm{Re} \geq 10^4$. Gnielinski correlation has been used for $\mathrm{Re} < 10^4$. Gnielinski correlation is the same as equation (7), except that Re has to be replaced by (Re − 1000). It is observed that for the same heating output, non-azeotropic mixture R143 require less refrigerant mass flow rate. Calculations are based on a single tube of 8 mm inner diameter; in an actual system flow can be split among multiple tubes. Heat transfer coefficient for CO_2 in the evaporator corresponding to saturated liquid is 3-4 times more than that of other fluids. So, for the same evaporator design, the required minimum temperature difference between the two fluids will be less for CO_2, yielding lower exergy loss in the evaporator and higher exergetic efficiency. Among other fluids, R236ea and R236fa are comparatively better.

Table 3: Heat transfer coefficients at evaporator and condenser pressure

Refrigerants	$\dot{m}_r$ (kg/s)	Nu_l (e)	α_l (e) kW/m^2K	Nu_l (c)	α_l (c) kW/m^2K	Nu_2 (c)	α_2 (c) kW/m^2K
CO_2	0.6590	3353.1	35.9861				
E134	0.5184	908.3	10.7921	2860.3	23.5248	4507.1	11.6357
R236fa	0.8071	1537.1	12.7168	5430.2	32.5272	7269.4	17.7367
R236ea	0.7194	1478.2	13.2538	4298.3	28.4494	6486.1	15.8501
R143	0.3476	590.8	9.8795	1940.9	21.4901	3294.8	9.5888
R22/R141b	0.4688	858.9	10.1473	3546.8	24.3698	4049.5	9.3608
R114	0.8658	1463.2	11.3522	4847.1	28.3512	7402.9	14.4266

The variation of heat transfer coefficients for CO_2 across the gas cooler is shown in figure 2. On the heating side also CO_2 has better heat transfer coefficient (about double) than other fluids, although R236ea, R236fa and R114 are better among the rest. Hence CO_2 is superior to other refrigerants in terms of heat transfer characteristics for high temperature process heat applications, as we can afford a lower minimum temperature difference in both evaporator and gas cooler for CO_2. This will also result in an increase in the difference between actual COP of CO_2 systems and that of other systems. It may be noted that the heating COP listed in Table 2 are for certain specific conditions and actual system COPs will be lower; however CO_2 systems, nevertheless, will yield relatively higher COP and exergetic efficiency.

5. HIGH TEMPERATURE APPLICATIONS

High temperature applications require process heat at temperatures exceeding 100 ^oC. Many applications even require process heat at above 150 ^oC. Most of the alternative refrigerants are useful up to 120 ^oC and a few may be till 150 ^oC. Water is the only option left for very high temperature applications beyond 150 ^oC; however it is not useful for a low temperature heat source. For high temperature applications, CO_2 can be used but it has also some limitation due to high system pressure. It was experimentally established (White, 2002) that the CO_2 heat pump could produce hot water at 90 ^oC without any operational problem. Klocker *et al.* (2001) also used CO_2 in drying heat pumps to produce hot air up to 60 ^oC. White *et al.* (2002) claimed that CO_2 based heat pumps could produce hot water even up to 120 ^oC. But with respect to exergetic efficiency, it is not profitable to produce 120 ^oC steam, high water pressure can be used to get high temperature water such as 120 ^oC at 2 bar. It is also possible to use such systems for high temperature air heating. Deterioration in heat transfer properties (Figure 2) is another limitation for heating temperature. As an alternative to synthetic chemicals (other alternative refrigerants), CO_2 heat pumps can be effectively used to produce hot air at high temperature and water of high temperature and pressure. There is also a possibility to use CO_2 and water heat pumps in series to produce steam at high pressure and temperature from low temperature water.

6. SIMULTANEOUS HEATING AND COOLING APPLICATIONS

There are process heat applications such as dairy plants, which require simultaneous heating and cooling. Although CO_2 systems are not suitable for refrigeration alone, they are very useful in simultaneous heating and cooling applications. Yarral *et al*. (2001) experimentally established that CO_2 performs well for simultaneous heating and cooling. Among the four natural refrigerants, propane and iso-butane are useful only for refrigeration applications; although propane (NBP = –40 .48 °C) is useful for low temperature requirements, iso-butane (NBP = –11.61 °C) is not useful for low temperature due to its negative pressure in evaporator. NH_3 is useful for heating up to about 70 °C. Triple point of NH_3 is –77 °C and hence it can be useful for very low temperature applications; however the negative pressure (NBP = –33 °C) posses a problem and thus may be applicable up to –35 °C. Triple point of CO_2 is –56.6 °C and pressure is also positive, and hence it is useful for refrigeration up to an evaporator temperature of about triple point and is suitable high temperature heating as well. Therefore, in terms of applicable range, CO_2 appears superior to other natural refrigerants. For cooling purposes NH_3 and propane are good in terms of both performance and heat transfer properties and for heating CO_2 seems much better; Subsequently a good opportunity exists for cascade systems with CO_2 and NH_3 or propane to improve the system performance as well as the range of applicability.

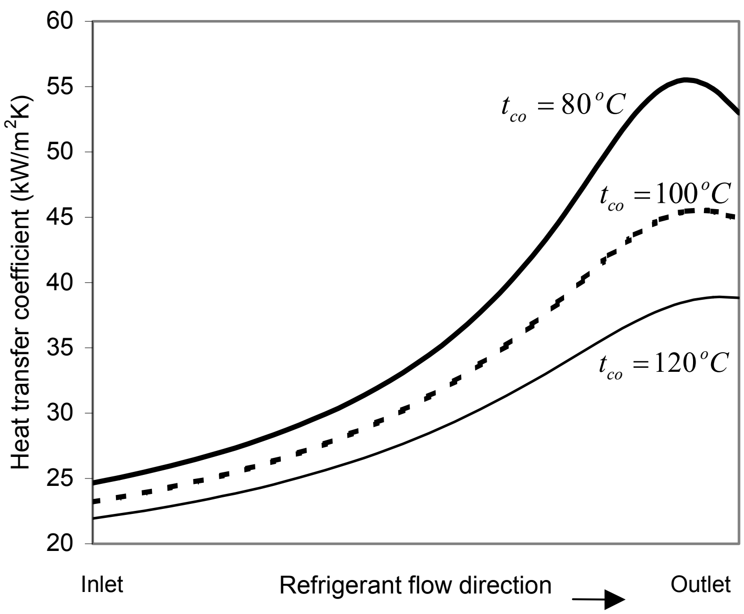

Figure 2. Variation of CO_2 heat transfer coefficients

7. CONCLUSION

Detailed comparison of CO_2 with other alternative refrigerants for high temperature heat pump applications have been demonstrated. Heat transfer analysis and applicability of CO_2 for high temperature heating as well as simultaneous heating and cooling applications are also discussed and the following conclusions can be drawn.

- CO_2 heat pumps can be successfully used for medium and high temperature process heat applications with some limitations such as low temperature heat source and low inlet temperature (open coolant system).
- CO_2 is superior to other alternative refrigerants with respect to heat transfer in both heating and cooling side for heating applications.
- CO_2 yields relatively better performance than other alternative refrigerants for process heat applications.
- Although the operating pressure is very high, CO_2 heat pumps require lower compressor discharge temperature and pressure ratio, and yields higher volumetric capacity than other alternative refrigerants.
- Refrigerant mixtures are better than other pure fluids for low inlet temperature due to its gliding temperature in condenser for high temperature heating.
- CO_2 heat pumps appear to be a good choice for simultaneous heating and cooling application in a wide range.

NOMENCLATURE

$COP_{heating}$	heating coefficient of performance	
$(COP_{heating})_c$	Carnot heating coefficient of performance	
d_i	inner diameter of tube	(m)
f	friction factor	
h	specific enthalpy	(kJ/kgK)
k	thermal conductivity	(W/mK)
P_{opt}	optimum pressure	(bar)
Pr	Prandlt number	
Q	heating output	(kW)
Re	Reynolds number	
T	temperature	(^{o}C)
W	compressor work input	(kW)
$\eta_{II,heating}$	exergetic efficiency for heating	
α	heat transfer coefficient	(kW/m^{2}K)

Subscripts

1-6	state point of refrigerant
co	condenser or cooler outlet
ci	condenser or cooler inlet
cr	saturated refrigerant vapor in condenser
v	saturated refrigerant vapor

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