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TITLE

A COMPARATIVE EVALUATION OF CO_2 AND HCFC-22 RESIDENTIAL AIR-CONDITIONING SYSTEMS IN A JAPANESE CLIMATE

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ABSTRACT

The natural refrigerant carbon dioxide (CO_2) offers ecological and personal safety, as well as low price, excellent availability and compact equipment. Results are presented from a comparative study on the performance of a small inverter-driven CO_2 split A/C system, in relation to a baseline HCFC-22 circuit. A typical Japanese climate is assumed. Based on the detailed simulation results, it is concluded that the CO_2 system can offer competitive seasonal energy efficiency both in cooling and heating mode, and the power consumption in cooling operation at extreme ambient temperature is more or less the same for both systems. A TEWI reduction of 10% is possible compared to HCFC- or HFC-based equipment. The trans-critical CO_2 cycle with heat rejection at gliding temperature can offer efficient hot water production and/or increased air delivery temperatures in heating mode.

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1. ABSTRACT

The natural refrigerant carbon dioxide (CO_2) offers ecological and personal safety, as well as low price, excellent availability and compact equipment. Results are presented from a comparative study on the performance of a small inverter-driven CO_2 split A/C system, in relation to a baseline HCFC-22 circuit. A typical Japanese climate is assumed. Based on the detailed simulation results, it is concluded that the CO_2 system can offer competitive seasonal energy efficiency both in cooling and heating mode, and the power consumption in cooling operation at extreme ambient temperature is more or less the same for both systems. A TEWI reduction of 10% is possible compared to HCFC- or HFC-based equipment. The trans-critical CO_2 cycle with heat rejection at gliding temperature can offer efficient hot water production and/or increased air delivery temperatures in heating mode.

2. INTRODUCTION

Carbon dioxide is one of the few natural vapor compression cycle refrigerants that is non-combustible and non-toxic, thereby offering not only environmental but also personal safety. During this decade remarkable progress has been made in the development of motor vehicle air conditioning systems based on CO_2 (Pettersen 1994, Wertenbach 1996), and the results have triggered interest in evaluating this concept also for residential air conditioning purposes.

Considerable work is now being made by the air conditioning industry to evaluate various HFC mixtures for replacement of HCFC-22, and the first units based on such blends (R-407C, R-410A) are already offered on the market. All these replacement fluids are artificial substances with global warming potentials (GWPs) that are 1600 to 2000 times higher than CO_2. In the case of R-410A, the GWP (1900) is even higher than that of HCFC-22 (1700). A number of industrial countries are now focusing on the growing emissions of HFC gases, and measures are introduced to limit this growth (RAC 1996). Zero-GWP refrigerant alternatives are therefore met by considerable interest, provided that factors like energy efficiency, safety and cost are competitive to those of present systems. The "double-zero" safety refrigerant CO_2 is therefore a very interesting option, and the major remaining question is what energy efficiency such a system can obtain.

Simple theoretical cycle calculations generally result in poor efficiency of CO_2 systems, since a number of important factors are neglected, such as the need for optimization of high-side pressure, realistic heat exchanger and compressor performance, heat rejection with gliding temperature from a single-phase super-critical gas, and differences in heating capacity characteristics.

In order to avoid these shortcomings and to provide more realistic data, the present study is based on detailed models of a HCFC-22 baseline system and a CO_2 system of equal size and capacity. Since the same simulation tools are used for both systems the results can be compared and analyzed, although the absolute level may not be entirely correct, especially since the compressor is assumed to have constant efficiency.

The present work represents both a refinement and a change of conditions compared to previous publications based on simpler models and a North-American climate (Aarlien et al. 1996). In the following text, specifications of the baseline and CO_2 systems are outlined, and the simulation model and calculation methods are briefly explained. Results for cooling and heating operation are then presented at rating conditions and at varying ambient air temperature, and also in the form of seasonal performance data. These results are then discussed and analyzed, focusing on the main parameters that affect the practical performance. Finally, some conclusions are drawn.

3. SYSTEM SPECIFICATIONS

Baseline system data were taken from a typical Japanese standard HCFC-22 split system (room air-conditioner). This was a ¾ hp inverter-driven unit with design-point cooling capacity 2.4 kW. Table I gives some additional data for the system at rating conditions as specified by JIS C 9612 (JIS 1994).

***Table I**: Capacities and rating conditions of the baseline HCFC-22 system.*

	Cooling Mode	Heating Mode
Cooling capacity, kW	2.4 (49 Hz)	-
Heating capacity, kW	-	3.5 (70 Hz)
Outdoor air temperature, °C	35.0	7.0
Indoor air temperature, °C	27.0	22.0
Indoor air relative humidity, %	47.4	86.9

The CO_2 system, which was based on a trans-critical vapor compression cycle with high-side pressure control (Lorentzen et al. 1993), had the same rating-point cooling capacity. A simplified circuit and a pressure/enthalpy diagram of the CO_2 cycle is shown in Figure 1. The supercritical high-side pressure is regulated by the expansion valve, and the evaporator outlet condition is maintained as a vapor/liquid mixture (Pettersen 1994).

The baseline HCFC-22 system had mechanically expanded tube-in-fin heat exchangers with enhanced tubes and fins. In order to predict the performance of the systems, heat exchanger performance data were needed. Since test results were not available, these data were obtained partly by an evaporator simulation program (EVAC), and partly by a condenser/gas cooler model that was integrated in the system simulation program *RACsim*. Specifications of the HCFC-22 heat exchangers were taken from the baseline system, and CO_2 heat exchangers were then designed within the same core dimensions and with equal or lower air-side pressure drop.

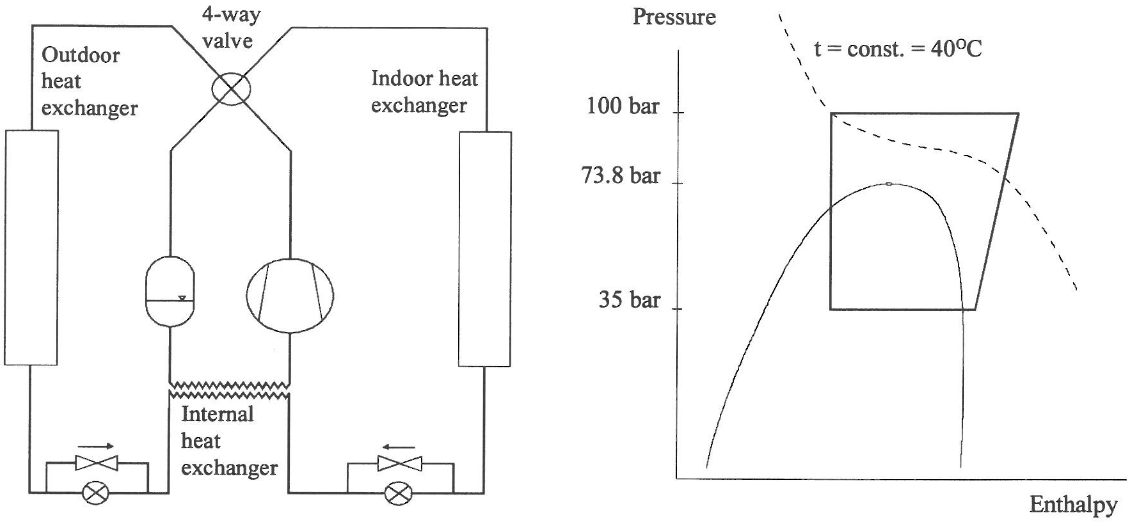

***Figure 1** - Flow circuit and pressure/enthalpy diagram of CO_2 system.*

Specifications of both units are listed in Table II. The CO_2 units had a larger number of small-diameter tubes. Fin pitch of the outdoor units were equal to avoid problems with frosting, and this gave lower air-side pressure drop and fan power consumption for the CO_2 outdoor unit (see the following text). Fin pitch of the CO_2 indoor unit was reduced to obtain air-side pressure drop equal to the HCFC-22 heat exchanger. All the units had staggered tube arrangement, plain aluminum fins, and 0.15 mm fin thickness. Although the HCFC-22 units had copper tubes, aluminum was assumed as tube material in the CO_2 heat exchangers. The main reason was the experience with this type of heat exchanger tube geometry in motor vehicle A/C prototypes. The effect of tube-side enhancements were not included in the simulations for any of the units.

***Table II**: Heat exchanger specifications.*

Heat exchanger	Outdoor unit		Indoor unit	
Refrigerant	HCFC-22	CO_2	HCFC-22	CO_2
Core dimensions (HxWxD), mm	530 x 600 x 40		280 x 620 x 30	
Face area, m^2	0.32		0.17	
Tube dimensions (OD/ID), mm	8.0 / 7.0	3.2 / 2.0	7.0 / 6.0	3.2 / 2.0
Number of tubes, -	48	120	28	76
Fin pitch, mm	2.0	2.0	1.70	1.35
Air-side surface, m^2	12.05	12.93	5.71	7.61
Refrigerant-side surface, m^2	0.68	0.48	0.34	0.30
Tube pitch - vertical, mm	20.0	10.0	15.0	7.5
Tube pitch - horizontal, mm	11.0	8.8	10.0	7.4

The circuiting of the CO_2 indoor and outdoor heat exchangers is shown in Figure 2.

The efficiency characteristics of hermetic CO_2 compressors are still uncertain, although preliminary results indicate that at least the same efficiency level as in HCFC-22 machines will be obtained (Fagerli 1996). In the present calculations, constant and equal compressor total efficiency (isentropic power requirement divided by electric power input) was therefore assumed at 65%. A maximum power input of 1.8 kW and a maximum frequency of 110 Hz was assumed for the compressor.

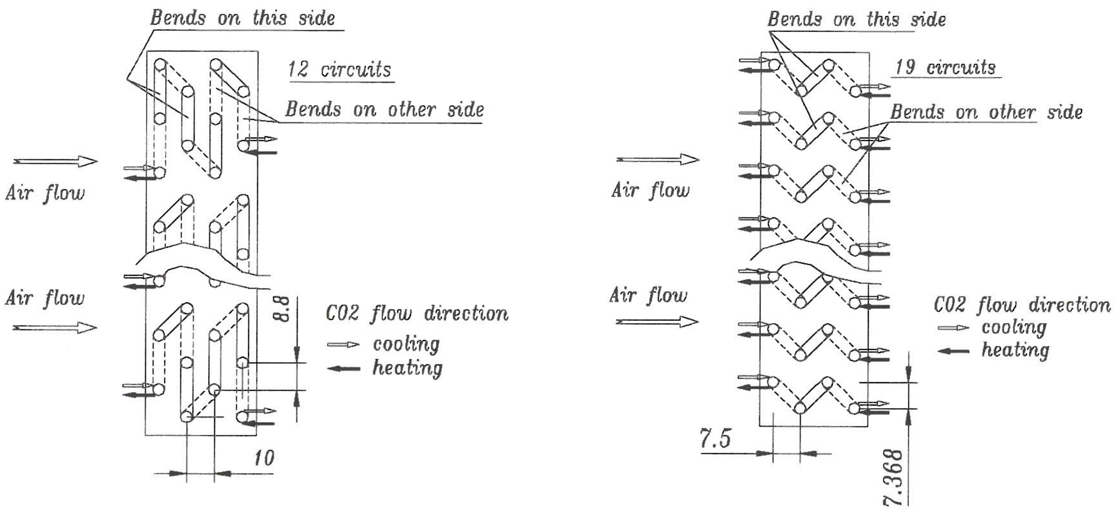

***Figure 2** - Tube geometry and refrigerant/air flow directions for CO_2 heat exchangers. Outdoor unit (left) and indoor unit (right)*

A suction line of ID 8 mm and 5.0 m length was assumed for both systems, and a double-tube internal heat exchanger of 1.5 m length was included in the CO_2 system.

4. SIMULATION MODEL AND CALCULATION METHODS

Capacity data from the evaporator simulations are shown in Figure 3 for the indoor (left) and outdoor (right) units. The EVAC program (Nesje and Skaugen 1989) is a detailed model based on stepwise calculation of local heat transfer and pressure drop data for the air side and refrigerant side, and integration/balancing through an iterative process to provide overall results. This program has recently been refined and adapted to compact heat exchanger types, and also to include CO_2 as a refrigerant. The indoor unit data were calculated at a constant air inlet temperature of 27°C and a humidity of 47.4%, and the outdoor unit evaporator data were calculated at varying air inlet conditions as experienced in the chosen climate (see following text). Air face velocities (w) are indicated in the diagrams (the HCFC-22 curve for 0.81 m/s air face velocity is more or less equal to the CO_2 curve for 0.53 m/s). The reversed direction of refrigerant flow as compared to heating-mode operation was taken into account, as indicated in Figure 2.

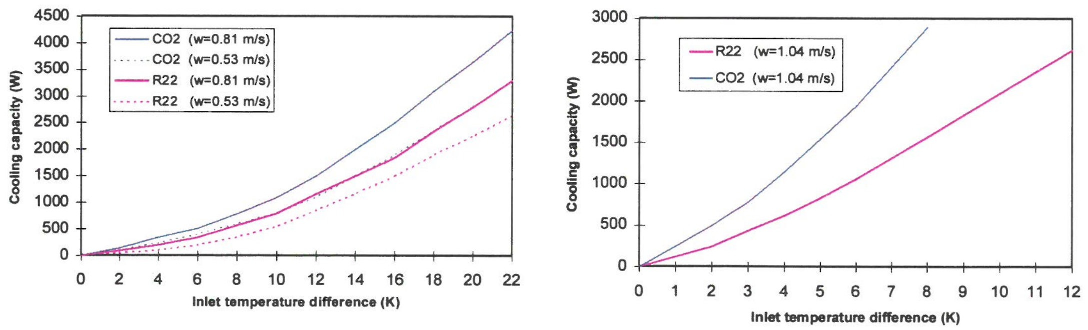

***Figure 3** - Simulated evaporator capacity data for indoor unit (left) and outdoor unit (right). Air face velocities (w) are shown in the diagrams.*

In order to account for the effect of varying refrigerant superheat in the HCFC-22 indoor unit, a correction curve as shown in Figure 4 was used. This curve is based on simulated evaporator performance at varying superheat.

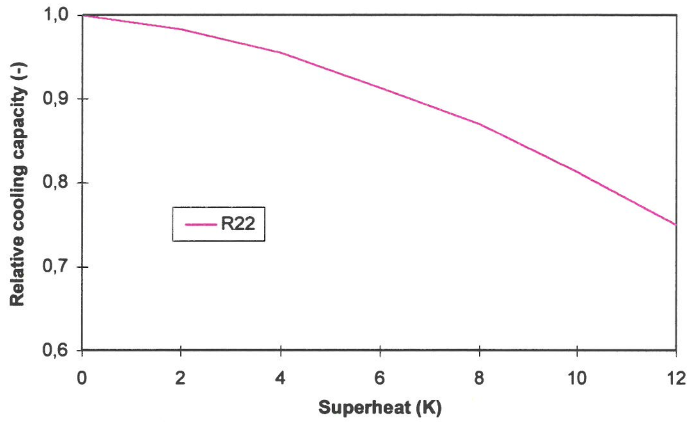

Figure 4 *- Assumed effect on capacity by varying superheat for the HCFC-22 indoor unit.*

The condenser and gas cooler were mathematically split into 10 sections, and the heat exchanger performance was then calculated as part of the system simulation, based on similar principles as in the EVAC model. This integration into the system model was selected due to the problem of defining gas cooler performance data in a simple way (equations or tables).

The indoor unit air face velocities were varied in accordance with the load conditions as shown in Table III. The outdoor unit air face velocity was 1.04 m/s for all operating conditions.

Table III*: Indoor unit air face velocities.*

Type of operation	Cooling operation		Heating operation	
Load condition	Low load	High load	Low load	High load
Ambient air temp., °C	Lower than 30	Higher than 30	Higher than 11	Lower than 11
Air face velocity, m/s	0.53	0.81	0.53	0.81

Superheat for the evaporators and subcooling for the condensers were also defined in relation to the load conditions, i.e. depending on the ambient air temperature. Table IV shows the assumed values for HCFC-22 cooling and heating mode. For the CO_2 system simulations in both cooling and heating mode, saturated evaporator outlet conditions were assumed, and during operation at sub-critical high-side pressures, 1K subcooling was assumed.

Table IV: *Evaporator superheat and condenser subcooling in HCFC-22 system simulations.*

Mode of operation	Cooling mode			Heating mode		
Load	Low	Medium	High	Low	Medium	High
Range of ambient temperature, °C	24-27	28-31	32-38	12-16	6-11	-6-5
Superheat, K	0.1	4.7	10.0	2.0	2.0	2.0
Subcooling, K	0.7	0.8	1.5	1.2	4.8	10.2

The internal heat exchanger in the CO_2 system was modeled based on a constant UA-value (constant heat transfer capacity per degree mean temperature difference).

By using the recently developed *RACsim* program, the performance of the complete HCFC-22 and CO_2 systems could then be simulated at specified operating conditions. These conditions were defined based on a simple model for the cooled/heated room, and climate data from JIS C 9612 (JIS 1994), "warm area" i.e. Tokyo climate with 1271.5 cooling hours and 3007.6 heating hours. The load characteristics of the room and the ambient temperature duration curve for heating and cooling operation are shown together with the results in Figures 6 (cooling) and 7 (heating).

RACsim is a C-coded system simulation program that can handle both sub-critical and trans-critical operation. The program is based on models of each component in the circuit, either as equation sets or tables, or as built-in subroutines. A general purpose equation solver and optimization procedure is used, and the program is linked to libraries of thermodynamic data for the refrigerants. In the present calculations, the compressor speed and the CO_2 high-side pressure were optimized in order to find COP maximums for given capacity requirements. COP values with and without supplementary heating were provided by the program

Based on the *RACsim* result output, seasonal performance data were calculated in a spreadsheet, in accordance with the JIS standard (JIS 1994).

5. RESULTS

Table V shows some of the main simulation results at the design points for cooling and heating operation, respectively.

Table V: *Simulation results for HCFC-22 and CO_2 systems at the rating points for cooling and heating operation.*

Mode of operation	Cooling mode 35°C outdoor 27°C indoor		Heating mode 7°C outdoor 22°C indoor	
Refrigerant	HCFC-22	CO_2	HCFC-22	CO_2
Cooling load and -capacity, kW	2.40	2.40	-	-
Heating load and -capacity, kW	-	-	3.53	3.53
Compressor power, kW	0.69	0.81	1.02	0.94
Compressor frequency, Hz	49	49	62	79
Fan power, indoor/outdoor unit, W	11 / 40	11 / 22	24 / 40	24 / 22
COP based on compressor power, -	3.50	2.98	3.45	3.75
COP including fan power, -	3.26	2.86	3.25	3.57
Evaporator refrigerant inlet temperature,°C	9.0	11.4	-0.7	-0.5
Evaporator refrigerant outlet saturation temperature, °C	6.4	11.3	-4.8	-0.6
Evaporator superheat, K	10	0	2	0
Condenser/gas cooler refrigerant outlet temperature, °C	45.0	36.0	40.4	26.4
High-side pressure (condenser/gas cooler outlet), bar	17.4	83.3	17.4	83.3
Compressor suction pressure, bar	5.8	45.8	3.9	33.7
Compressor discharge pressure, bar	17.9	95.6	17.7	83.8
Compressor pressure ratio, -	3.1	2.1	4.6	2.5

As may be observed from the table, the simulated cooling COP of the CO_2 system is 12% lower than that of the HCFC-22 system at the design point. In the heating-mode design point, however, the simulated COP is 10% higher for the CO_2 system than for the HCFC-22 system. Supplementary heat is not needed at this temperature for any of the systems.

The evaporating temperatures are a few degrees higher in the CO_2 system (equal capacities), due to a larger air-side surface (indoor unit), absence of superheat, and lower refrigerant-side temperature drop. This, together with the much narrower temperature approach in the CO_2 gas cooler than in the HCFC-22 condenser, improves the relative efficiency of the CO_2 system. The temperature approach is defined as the refrigerant exit temperature minus the air inlet temperature.

Figure 5 shows the simulated COP of the two systems (including fan power) for cooling operation at varying ambient air temperature. The distribution of cooling hours and the cooling load is also shown in the diagram. As may be observed, the COPs are equal at about 27°C, slightly lower for the CO_2 system at higher temperatures, and better than for the HCFC-22 system at lower temperatures. Apart from at the lowest temperatures, the difference in COP between the HCFC-22 and CO_2 systems is not more than 15%. At 38°C ambient temperature, the difference is only 8.4%.

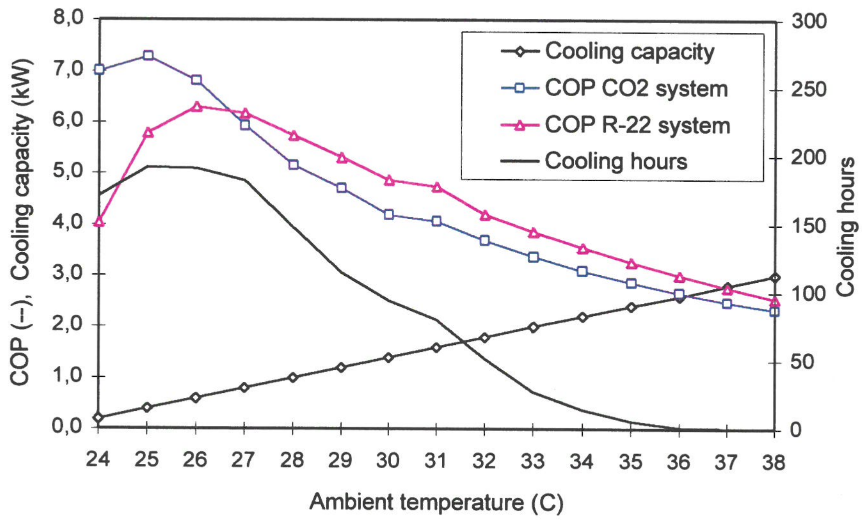

Figure 5 *- Simulated cooling COP of the CO_2 and HCFC-22 systems for operation at varying ambient air temperature. The distribution of cooling hours and the cooling load is also shown.*

COP and capacity data for heating operation are shown in Figure 6, together with the distribution of heating hours and the heating load curve.

As may be observed, the CO_2 system is able to maintain a higher heating capacity (Q_{hp}) than the HCFC-22 system as the ambient temperature is reduced below 5°C, and less supplementary heat is thus needed. The heating output from the CO_2 heat pump is boosted by raising the high-side pressure, and although this reduces the COP, it gives a higher overall efficiency thanks to reduced supplementary heat. Figure 7 shows that for ambient temperatures below 2°C, the CO_2 compressor is operating at maximum load, i.e. 1.8 kW, and the high-side pressure is raised as the outdoor temperature is reduced, up to a maximum of approximately 95 bar at -6°C ambient. The HCFC-22 compressor is operating at maximum frequency 110 Hz from an ambient temperature of 5°C, and the CO_2 machine from 2°C. The shift in COP values around 11-12°C is due to the change in indoor unit air flow rate at this temperature.

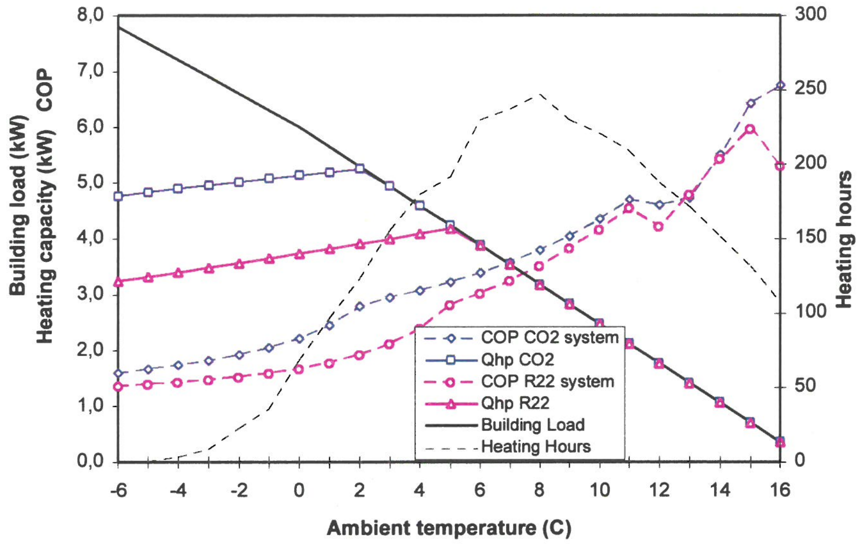

Figure 6 *- Simulated heating COP and heating capacity of the CO_2 and HCFC-22 systems at varying ambient air temperature. Distribution of heating hours and building heating requirements are also shown.*

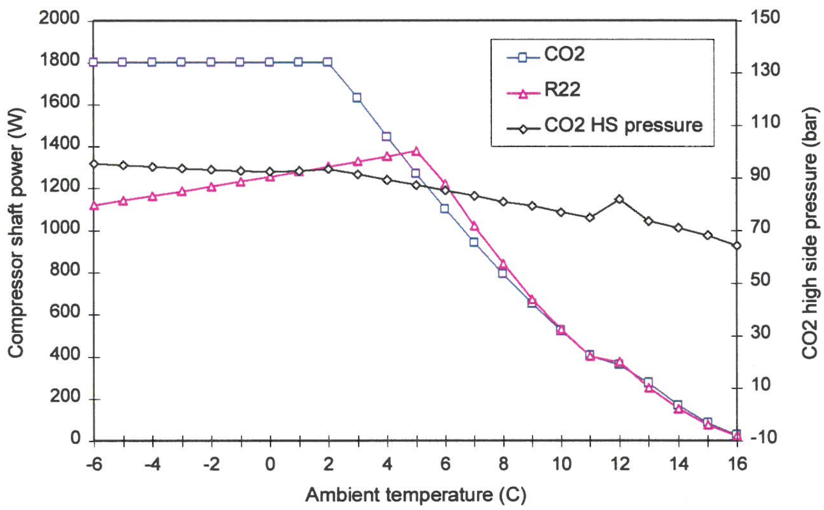

Figure 7 *- Compressor shaft power for HCFC-22 and CO_2 systems, and CO_2 system high-side pressure, in heating operation.*

Based on a combination of performance data from each ambient temperature, and duration data for the temperatures (cooling and heating hours), the *seasonal* performance of the two systems can be calculated (JIS 1994). Table VI shows the simulated seasonal performance factors resulting from these calculations.

Table VI *- Simulated seasonal performance factors for the two systems*

Mode	Cooling operation		Heating operation	
Refrigerant	HCFC-22	CO_2	HCFC-22	CO_2
Seasonal COP, -	5.0	4.7	2.7	3.3
SEER, Btu/kW	17.2	16.1	-	-
HSPF, Btu/kW	-	-	9.2	11.1

The simulated seasonal cooling COP is thus about 6% lower for the CO_2 system than for the HCFC-22 baseline system, while for heating operation, the CO_2 system is about 20% better than the baseline. These differences are not very large, and at least the seasonal cooling data are well within the accuracy of the simulations.

6. DISCUSSION

Apart from the compressor performance characteristics, the present calculations include most of the important practical effects that would exist in a actual system, such as heat exchanger performance and pressure drops, as well as all the refrigerant properties. Although the absolute level may not be entirely correct, the results should give a reasonably good estimate for the *relative* performance of a CO_2 system, as compared to a HCFC-22 baseline circuit.

In the following text, some of the main parameters that affect the practical performance of the CO_2 system as compared to the HCFC-22 unit are discussed.

Evaporator temperature: As shown by the evaporator capacity curves in Figures 3 and 4, the CO_2 evaporators generally show better performance than the corresponding HCFC-22 units. As already pointed out, the main reasons are absence of superheat, better heat transfer coefficients, lower pressure drops, and larger air-side surface (indoor unit). Although the curves represent simplified data, the simulation program consistently gave better performance for the CO_2 evaporators. As a result, the CO_2 system will operate with higher evaporating temperature and thereby higher COP. Figure 8 shows simulated evaporating temperatures for cooling operation (left) and heating operation (right). The break in the curve between 30°C and 31°C (cooling operation) is due to the change in fan speed of the indoor unit.

An interesting question related to the "cooling" temperatures is the ability of the CO_2 system to dehumidify the indoor air. In order to evaluate this, some evaporator simulations were performed with equal cooling capacity 2.4 kW, at air inlet temperature 27°C, and at two different humidity levels, 80% and 60% r.h. The results are listed in Table VII, where the evaporator conditions and the dehumidification characteristics are shown for the HCFC-22 and CO_2 evaporators. Despite refrigerant outlet saturation temperatures that are 2-3 K higher in the CO_2 evaporator, there is not much difference in the water condensation rate. At 80% r.h. the two evaporators perform equally, and at 60% the simulated rate is about 9% lower for the CO_2 system. This indicates that the dehumidifying effect can be maintained even with the higher CO_2 evaporating temperatures, although some adjustments in surface area and temperature may be necessary.

The main reasons for this result is again the absence of a superheat zone in the CO_2 evaporator, as well as lower refrigerant-side temperature drop, and a better fin efficiency that gives a quite low average fin surface temperature.

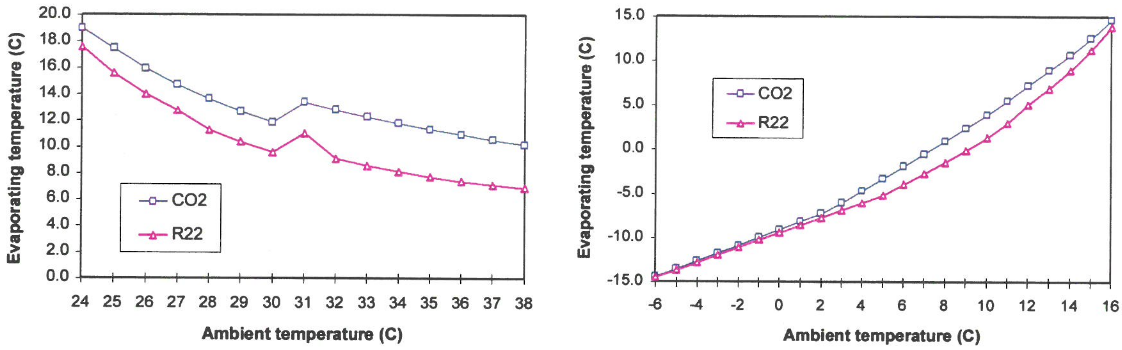

Figure 8 *- Simulated evaporating temperatures for the HCFC-22 and CO_2 systems in cooling operation (left) and heating operation (right).*

Table VII*: Calculated dehumidifying effect of HCFC-22 and CO_2 evaporators. Cooling capacity 2.4 kW, air face velocity 0.81 m/s.*

Air inlet condition	27°C, 80% r.h.		27°C, 60% r.h.	
Type of system	HCFC-22	CO_2	HCFC-22	CO_2
Saturation temperature at evaporator outlet pressure, °C	16.0	18.3	12.0	14.9
Superheat, K	5.0	0	5.0	0
Inlet refrigerant temp., °C	18.1	18.5	14.7	15.1
Water condensation rate, g/min	33	33	23	21
Part of fin area with water condensation, %	95	100	94	100
Air outlet temperature, °C	19.5	18.9	17.0	15.6

Gas cooler vs. condenser temperature approach: The COP and capacity of the CO_2 system is affected to a large extent by the gas cooler refrigerant outlet temperature, which ideally should approach the air inlet temperature, since heat rejection from the CO_2 takes place at gliding temperature in (super-critical) single-phase heat transfer. Owing to the outdoor unit design with four tube rows in the direction of air flow, the cooler approaches the behavior of a counterflow heat exchanger. The CO_2 indoor unit will also operate as a counterflow gas cooler during heating operation, with similar ability to give a close temperature approach. As a result, the refrigerant outlet temperatures will be quite close to the air inlet temperatures. In contrast, the HCFC-22 condenser is designed like a cross-flow heat exchanger, in principle with only one tube row (from a thermal point of view) without the same possibility for a narrow temperature approach.

The temperature profiles in the two types of outdoor heat exchangers are illustrated in Figure 9, showing temperatures along the refrigerant circuit, as a function of position from the refrigerant inlet.

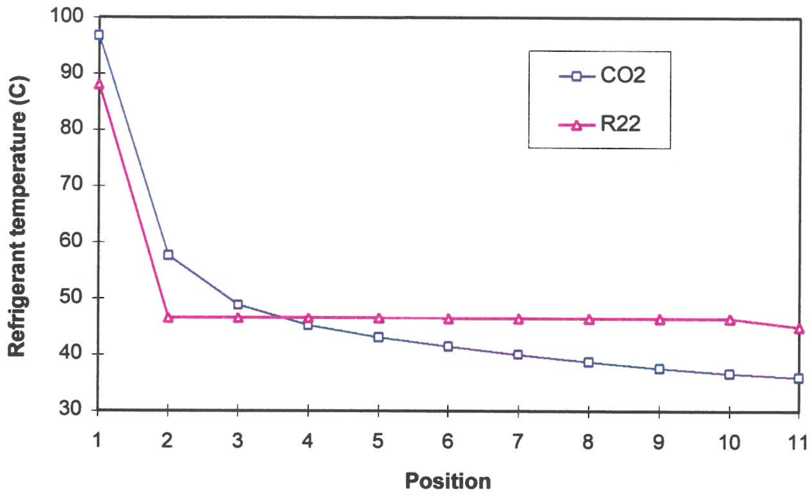

Figure 9 *- Refrigerant temperatures in HCFC-22 condenser and CO_2 gas cooler, during cooling operation at 35°C ambient air temperature.*

Figure 10 shows the simulated temperature differences in the outdoor heat exchanger during cooling operation at varying ambient air temperature.

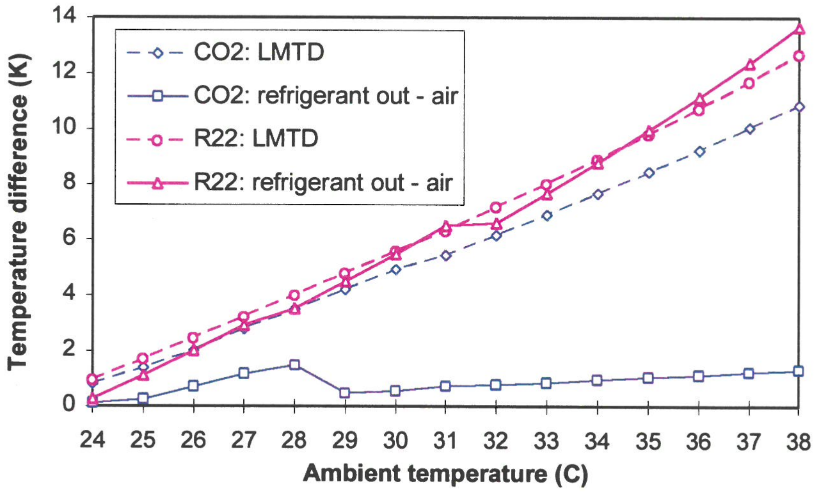

Figure 10 *- Temperature differences in the outdoor heat exchanger during cooling operation. Approach, i.e. refrigerant outlet minus air inlet temperature, and LMTD based on inlet and outlet temperatures on both sides.*

The temperature approach of the CO_2 gas cooler is always less than 2K, while the corresponding difference in the HCFC-22 condenser varies more or less linearly with the load, up to a maximum of almost 14K. The break in the CO_2 curve around 28-29°C ambient air temperature is due to the change from sub-critical to supercritical high-side pressure.

Regarding the LMTDs, there is not much difference between the two refrigerants, indicating that the "average" temperature differences are more or less the same in the two heat exchangers.

Suction line pressure drop: Figure 11 shows the calculated pressure drop in the suction line of the two units, during cooling operation (left) and heating operation (right). The pressure drop in the CO_2 system is lower both in absolute terms and in relation to the pressure level, and the reduced losses associated with this also contribute to the good practical efficiency of the CO_2 system.

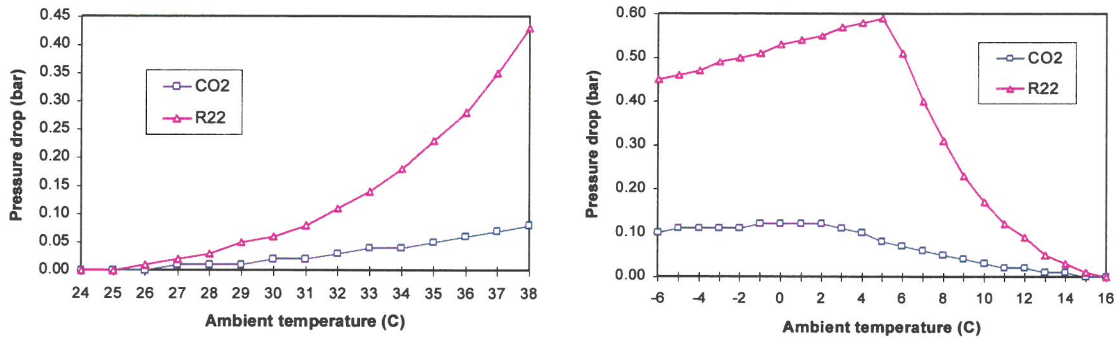

Figure 11 *- Simulated suction line pressure drops in the HCFC-22 and CO_2 systems, during cooling operation (left) and heating operation (right).*

Compressor Pressure Ratio: The pressure ratio encountered by the compressor in the two systems is shown in Figure 12 for cooling operation (left) and heating operation (right). As pointed out above, the lower pressure ratios in the CO_2 compressor could give improved efficiency, further contributing to a better practical efficiency. This effect is not included in the current analysis, however.

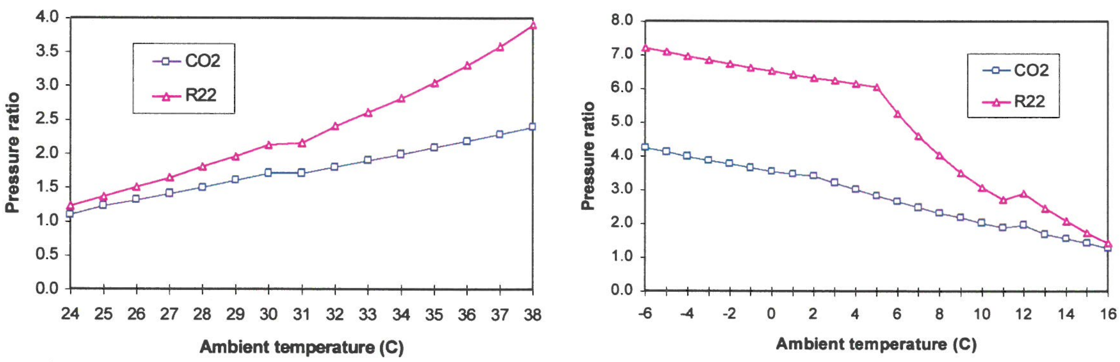

Figure 12 *- Compressor pressure ratios for HCFC-22 and CO_2 system during cooling operation (left) and heating operation (right).*

Supplementary heating: It has been argued that supplementary heating is not common in Japan, and, despite the clear procedures given in the JIS standard (JIS 1994), that heat pump operation should be analyzed without including any supplementary heating. Simulations were therefore carried out for the two heat pumps without any additional heat input to the room, and the resulting indoor temperatures calculated, Figure 13.

The benefit of the higher heating capacity from the CO_2 system can clearly be observed as 2-3 K higher room temperatures on the coldest days, and the ability to keep the room temperature at 22°C down to 3°C ambient air temperature, while the

HCFC-22 system is only able to meet the heating requirements down to 6°C ambient air temperature.

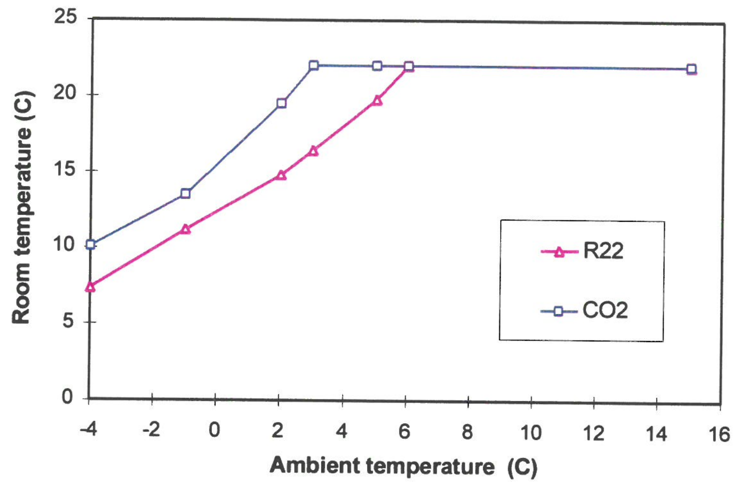

Figure 13 *- Simulated room temperatures with supplementary heating switched off and HCFC-22 or CO_2 heat pump as only heating means.*

If priority is given to COP instead of capacity, the high-side pressure in the CO_2 system should be optimized for maximum efficiency instead of capacity. The resulting seasonal performance data are listed in Table VIII. In these calculations, it is assumed that the room temperature is maintained at 22°C, but the energy requirements for supplementary heat is not taken into account in the calculation of seasonal COP. The heating capacity for the CO_2 system is adjusted to the same level as the HCFC-22 system at all temperatures below 6°C.

Table VIII *- Simulated seasonal performance factors for the two systems, assuming that the supplementary heat is not included in the energy requirements.*

Refrigerant	HCFC-22	CO_2
Seasonal COP, -	3.3	3.6
HSPF, Btu/(h·kW)	11.3	12.4

Even under these assumptions, the simulated seasonal performance of the CO_2 heat pump is very competitive, although the difference is reduced to approximately 9% compared to 20% when the supplementary heat is included (Table VI).

Comparison with earlier results: Earlier studies (Aarlien et al. 1996) suggested that the cooling power requirements for the CO_2 system would be significantly higher than the baseline at extreme ambient air temperature. The present results does not show this effect, and the main reason seems to be a more realistic model that includes the effect of increasing evaporator superheat and pressure drops as the capacity is increased, as well as increasing condenser approach temperature differences in the HCFC-22 system at higher loads.

The present model estimates 6% lower cooling SEER for the CO_2 system, while the previous study (North-American climate) predicted an increase by 10%. Both studies indicate about 20% higher HSPF, however. All in all, the main conclusions are still valid, although the expected rise in power consumption at high temperature may not be the case in practice. All these results will have to be verified by experimental investigations, which are planned in the coming year.

Relation to Japanese energy efficiency standards: Provided that the simulated COP levels are realistic, both the HCFC-22 system and the CO_2 system satisfy the new requirements of the Japanese "Energy Savings Law" to become effective October, 1997. For a split unit smaller than 4.0 kW, the minimum cooling COP requirement will be 2.67 and the minimum heating COP will be 3.2 (Kunugi et al. 1995). It should be noted that these minimum levels are not related to *seasonal* performance, but to the performance at JIS standard conditions (Table I).

7. CONCLUSIONS

Based on the results and findings from this study, the following conclusions may be drawn:

- The seasonal energy efficiency of the CO_2 system is competitive with the baseline HCFC-22 system data, both for cooling and heating operation. A slightly lower SEER is simulated, but this difference is not significant in relation to the uncertainty of the data. The results indicate that the HSPF may be somewhat higher for the CO_2 system, at least when the supplementary heat is accounted for.

- In contrast to earlier results, the present study does not indicate significantly higher power consumption for the CO_2 system during cooling operation at extreme ambient temperatures.

- Assuming equal seasonal energy consumption, a CO_2 room air conditioner will be able to reduce the Total Equivalent Warming Impact by approximately 10% as compared to HCFC or HFC-based equipment (ORNL 1996).

- The trans-critical CO_2 cycle with heat rejection at gliding temperature offers interesting options for integration of hot service water production and/or increased air delivery temperatures in heating mode.

8. ACKNOWLEDGEMENTS

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