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TRANSCRITICAL CO_2 REFRIGERATION SYSTEMS FOR BUILDING COOLING AND HEATING REDUCE ENERGY AND COOLING WATER CONSUMPTION, EMISSIONS AND THE LEGIONELLA DANGER

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Abstract—**In many large office buildings, hospitals and hotels AC cooling is affected by chilled water generated by a refrigerating plant which rejects its heat to the environment by means of cooling tower water or, increasingly so, by air cooled condensers and adiabatically assisted air-cooled condensers. Hot water for space heating, AC reheat and sanitary purposes is usually provided by a boiler.**

Using a COAG, the Council of Australian Governments, data base on annual energy consumption per m^2 of building, it is shown that significant reductions in the electrical energy, natural gas and cooling water consumption may be achieved when using transcritical CO_2 refrigerating systems for building cooling, heating and sanitary hot water, thus saving gas and attendant emissions, and cooling water. It is shown that the specific energy consumption measured in kWh/m^2.annum reduces about 30% when retrofitting trans-critical CO_2 systems to existing Australian buildings and about 55% in new buildings. These numbers range from about 10% to 50% in hospitals. Supermarkets with all CO_2 refrigeration would save about 37%.

The natural refrigerant CO_2 is non-toxic, non-flammable, low cost and energy efficient. CO_2 also has superior heat transfer properties and its GWP=1. As such CO_2 is a sustainable refrigerant and will future proof any system against the serious implications of the impending phase down, and ultimate phase out, of the high GWP HFC refrigerants by the amendment to the Montreal Protocol, known as the Kigali amendment, where it was agreed to by the 197 signatories to the Montreal Protocol on 15 October 2,016.

Keywords—**CO_2 refrigerating systems, energy, water, global warming, Montreal protocol, Kigali amendment.**

I. INTRODUCTION

The late Prof. Gustav Lorentzen publicly called for the revival of CO_2 refrigeration in 1993, see Fig. 1. Since then air cooled gas cooling – some with adiabatic assistance by spraying water onto the air inlet face of the finned coil gas cooler – has been applied almost universally. This has resulted in virtually all CO_2 refrigerating systems needing to run in transcritical mode some or all of the time because the cooling air temperature is close to or exceeds the CO_2 critical temperature of 31.1°C in many cases.

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This results in the compressors needing to operate at a discharge pressure of 8 MPa or higher to ensure a reasonable COP. The summer design COPs of transcritical CO_2 compressors are generally lower than those of air cooled HFC or evaporatively cooled ammonia systems. As a consequence, to date most CO_2 refrigeration applications have been in the form of CO_2/HFC cascade systems with air cooled HFC condensing. Larger industrial plants have employed CO_2/NH_3 cascades with evaporative condensers.

As observed by Pearson [1] the obvious solution is to reduce the temperature of the condenser cooling medium to allow a complete subcritical CO_2 refrigeration cycle all or most of the time. This is easily accomplished with an evaporative condenser, where the effective cooling medium temperature is the ambient air wet bulb temperature rather than the ambient air dry bulb temperature in the case of an air cooled condenser or gas cooler. Furthermore, existing CO_2/HFC cascade systems may be converted to multifunction two stage transcritical CO_2 refrigerating systems in future when the Montreal Protocol (MP) has completed its new onerous task of reducing the levels of HFC production and consumption, as agreed to under the provisions of the Kigali Amendment to the Montreal Protocol on 15 October 2016.

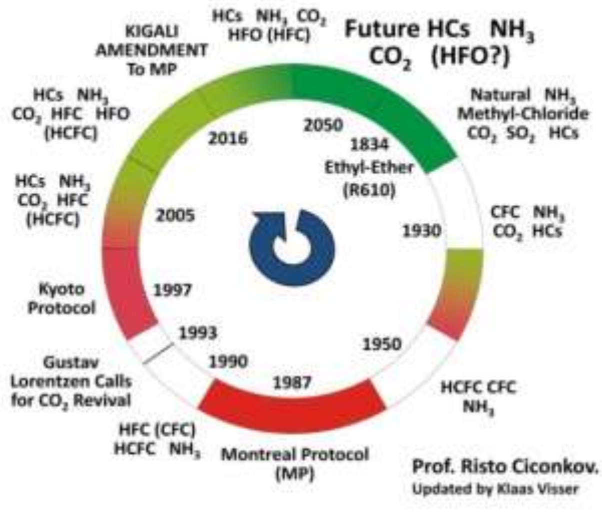

Figure 1. History of refrigerants

A. Main advantages of CO_2

- CO_2 is non-toxic, non-flammable, has a low Global Warming Potential of 1, is low cost and environmentally sustainable, and reduces cooling water and energy consumption.
- CO_2 has better heat transfer properties than Ammonia and HFC refrigerants at all temperature levels.
- CO_2 pressure vessel and suction piping diameters are about half the diameter of NH_3 vessels and piping.

B. Disadvantages of CO_2

- CO_2 systems operate at higher pressures than NH3 and chemical refrigerants.
- In confined spaces CO_2 can be fatal at concentrations exceeding 8% by weight. CO_2 detectors need to be installed in all refrigerated chambers close to floor level.
- An Ammonia cooled CO_2 cascade condenser is an expensive piece of equipment. Should CO_2 leak into the Ammonia side of the heat exchanger, as reported recently, ammonium carbamate will be formed, which is a highly abrasive salt. It would quickly damage compressors significantly if not detected.

II. DATA USED IN ANALYSIS

A. USA data

Westphalen et al. [2] prepared a report dealing with Thermal Distribution, Auxiliary Equipment and Ventilation in the Energy Consumption Characteristics of Commercial Building Heating, Ventilating, Air Conditioning (HVAC) systems comprising 3,345 million m^2 of cooled building floor space plus 4,459 million m^2 of heated building floor space. The total annual HVAC primary energy consumption was 4.85 x 109 GJ. See Fig. 2 [3]. Of the total Parasitic

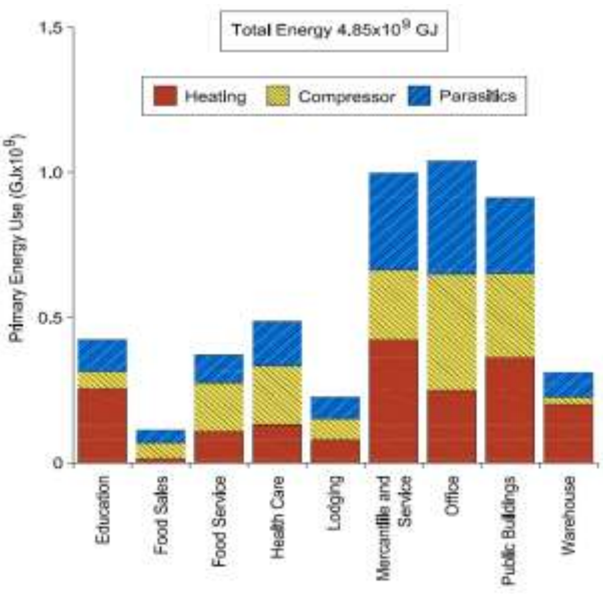

Figure 2. Total HVAC primary energy use by building type

Primary Energy use in HVAC systems in office buildings, the supply and return air fans plus the exhaust fans consumed 80% of a total of 0.39 x 10^9 GJ i.e. 0.31 x 10^9 GJ. In the hospital category the supply and return fans plus exhaust fans consumed a total of 0.12 GJ of primary energy. See Fig. 3.

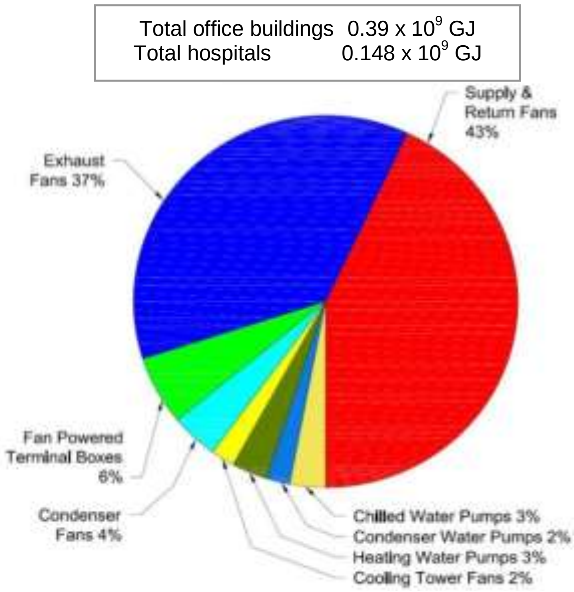

Figure 3. Parasitic energy use by type of equipment

In Table 1, seasonal efficiencies for different cooling equipment and the distribution of the equipment as applied in USA office and hospital categories according to floor areas served [2] have permitted the evaluation of a weighted COP for the entire office building and hospital categories.

B. Australian data

In November 2012 the Council of Australian Government (COAG) published Part 1 of a Report [4] as part of COAG's National Strategy on Energy Efficiency. The report deals with seven building types; offices, hotels, retail buildings, hospitals, schools and tertiary education and public buildings. The data on the former four building types has been used in this study to arrive at potential energy performance in those buildings should they be retrofitted with CO_2 refrigeration technology applied to both heating and cooling. In Figures 4, 5 and 6 the cooling COPs of a commercially available semi-hermetic CO_2 compressor with a swept volume of 27.2 m^3/hr are shown at varying suction and discharge conditions.

Figure 4 shows the COP variation with Saturated Suction Temperatures (SSTs) ranging from +10°C to −5°C and at Saturated Condensing Temperatures (SCTs) from 16 to 30°C with varying degrees of liquid subcooling.

Figures 5 and 6 show the COPs at 7.5, 8, 9 and 10 MPa discharge pressures at CO_2 gas cooler exit temperatures ranging from 5 to 35°C and 5K Suction Super Heat (SSH) at +10 and +5°C SST respectively. It is clear from Figures 4, 5 and 6 that CO_2 behaves much like any other refrigerant in that the lower the discharge pressure and the higher the suction pressure, the higher the COP will be and vice versa.

TABLE 1. BUILDING AND HOSPITAL EQUIPMENT SEASONAL EFFICIENCIES – COEFFICIENTS OF PERFORMANCE (COP'S)

Equipment type used	COP [1] Existing equipment	COP Correction Factor for fans	COP corrected	USA Offices: Office area applied to [2] $m^2 \times 10^6$	USA Offices: Office area applied to [2] %	USA Offices: Weighted contribution to mean COP	USA Hospitals: Floor area[1] applied to $m^2 \times 10^6$	USA Hospitals: Floor area[1] applied to %	USA Hospitals: Weighted contribution to mean COP
Centrifugal chiller	4.4	1	4.4	135	15	0.66	45.5	22.0	0.97
Water cooled screw chiller	4.14	1	4.14	18	2	0.083	6.2	3.0	0.12
Water cooled recip. chiller	3.91	1	3.91	27	3	0.117	3.3	4.0	0.16
Air cooled screw chiller	3.17 [3]	1.1 [5]	3.49	36	4	0.140	20.7	10.0	0.353
Air cooled recip. chiller	2.71 [3]	1.1 [5]	2.98	144	16	0.477	58.0	28.0	0.84
PTAC, PTHP	2.49 [4]	1.3 [6]	3.24	59	7	0.227	-	-	-
Room AC	2.34 [4]	1.3 [6]	3.04	58	6	0.182	12.4	6.0	0.18
Packaged AC	2.13 [4]	1.3 [6]	2.9	351	39	1.13	44.2	21.3	0.62
Heat pump	2.13 [4]	1.3 [6]	2.9	62	7	0.203	7.5	3.6	0.11
Absorption chiller	0.98	1.0	0.98	9	1	0.001	4.2	2.1	0.02
TOTAL				**899**	**100**	**3.22**	**207**	**100**	**3.35**

[1] Westphalen et al (1999)
[2] Westphalen et al (1999)
[3] Includes condenser fans
[4] Includes condenser and evaporator
[5] Adjustment factor for condenser fans
[6] Adjustment factor for condenser and evaporator fans

It is also clear from Figures 4, 5 and 6 that, provided the gas cooler CO_2 exit temperatures do not exceed 30-31°C, transcritical CO_2 COPs have quite acceptable values compared to those shown in Figure 7 for R717, R22, R507A, Propane and R134a at SCTs ranging from 16 to 40°C. When comparing the three CO_2 COP values at −5°C SST in Figure 4 with the COPs for conventional refrigerants shown in Figure 7, it is clearly evident that the performance of subcritical CO_2 refrigeration systems compares favorably with conventional ammonia and HFC refrigeration systems. This is partially due to much reduced boiling point suppression in the suction lines in the case of CO_2, which is due to the low boiling point suppression in K per unit pressure expressed in bar. This is another major advantage of CO_2.

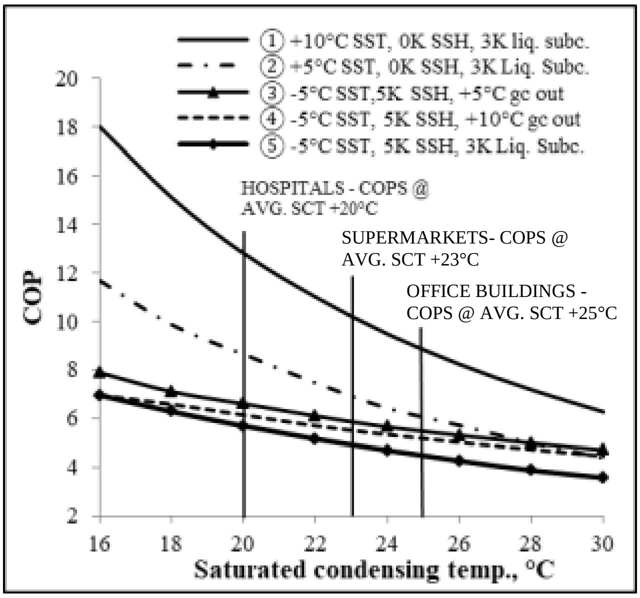

Figure 4. COPs vs condensing temperature for 27.2 m^3/h compressor

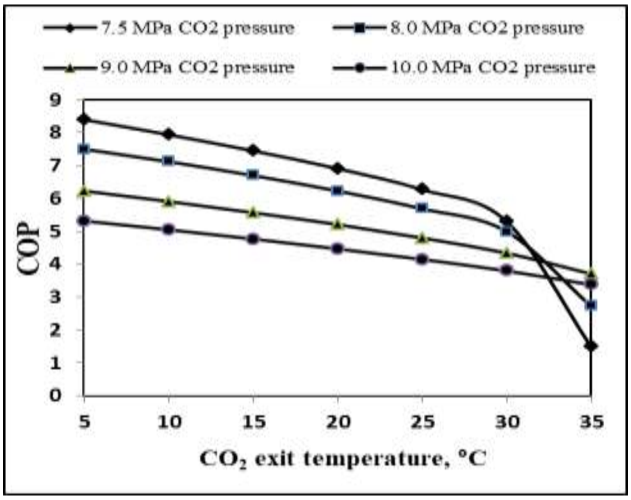

Figure 5. Compressor COP vs CO_2 exit temperature at 10°C SST and 5 K superheat

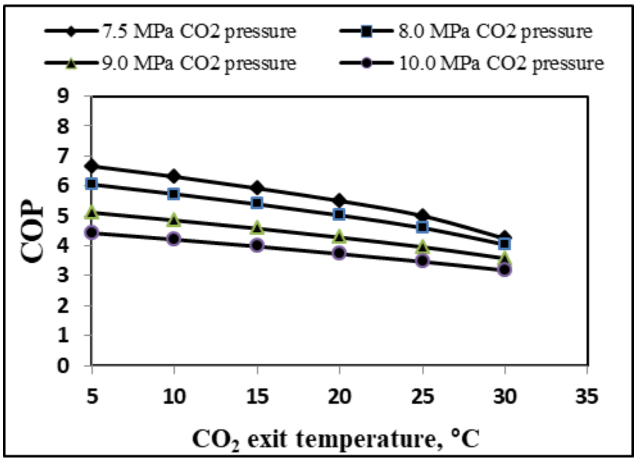

Figure 6. Compressor COP vs CO_2 exit temperature at 5°C SST and 5 K superheat

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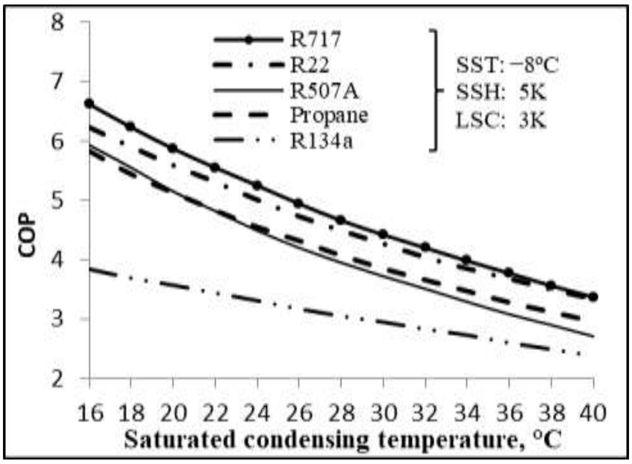

Figure 7. COP vs condensing temperature of a commercially available compressor with a swept volume of 637 m^3/h

III. ANALYSIS BASED ON COAG REPORT

A. *Existing situation*

In Table 2 the specific energy consumption in MJ/m^2 is listed for Australia Stand Alone Office Buildings (SAOBs), Hotels (HTLs), Hospitals (HSPs) and Supermarkets (SASMs). Furthermore, the percentages of Electrical Energy (EE) and Fuel (FL) are listed and broken up in MJ/m^2. They are based on the projected building areas in m^2 and the total energy consumption PJ in 2015.

The total energy consumption is the sum of the fossil fuel – mostly natural gas – used in MJ plus the EE as electrical fuel in MJ obtained by multiplying kWh EE consumption by 3.6 as 1 kWh = 3.6 MJ. The analysis is carried out by defining the amount of EE and FL used and the percentage proportion of the use of these two fuels by of the total specific energy consumption per m^2 of building area.

EE is used for HVAC, lighting, equipment, domestic hot water, pool heating and other undefined duties. In the case of the supermarkets the MT and LT refrigeration duties also consume EE. Natural gas is used for space heating, domestic hot water, pool heating, laundry, sterilization and other undefined duties. In Table 3 we have summarized the specific HVAC EE consumption in kWh/m2.a and Table 4 shows the total amount of heat required for space heating and hot water. In the case of hospitals the 6% sterilization heat has not been added. In all cases “Other uses” have not been added as it is not known what those “Other uses” are.

B. *Evaluation of energy consumption using transcritical CO_2 refrigeration systems in office buildings, hotels and hospitals.*

All of these facilities require HVAC, space heating and domestic hot water (DHW), particularly hotels and hospitals. The low critical point of CO_2 is a distinct advantage when heating DHW from the mains temperature to a desirable level of 55 to 65°C. The advantage consists in the fact that in transcritical operations there is no condensing phase and

TABLE 2. SPECIFIC ENERGY CONSUMPTION IN MJ/M^2 FOR AUSTRALIAN SAOBS, HTLS, HSPS, SASMS

Parameter			Type of Building				
No	Description	Unit	SAOBs		HTLs	HSPs	SASMs
			T[(1)]	BB[(2)]			
1	Building area	m^2x10^3	40,911		11,424	13,747	5,926
2	Annual Energy Consumption (AEC) / Emissions (E)	PJ/a;MtCO_2e/a	35.6/9.2		17.7/3.5	22.2/3.7	20/5.7
3	Specific AEC	MJ/m^2.a(kWh/m^2.a)	870(241)		1,549(430)	1,615(449)	3,375(938)
4	Electrical Energy (EE)	PJ (%)	15.4(43.21)	16.7(47)	11.3(64)	10.9(49.1)	19.8(99)
5	Natural Gas	PJ (%)	3.4 (9.8)		6.3(36)	10.5(59.9)	0.2(1%)
6	LPG/Diesel	PJ (%)	0.1 (9.8)		0	0.710	0
7	EE End Use						
.1	HVAC	%	18	67	52.0	47	35
.2	MT Refrig.	%	-	-	-	-	25
.3	LT Refrig.	%	-	-	-	-	15
.4	Lighting	%	37	15	20.0	17	
.5	Equipment	%	31	11	11.0	7	
.6	Domestic Hot Water	%	3	2	9.0	2	25
.7	Pool Heating	%	-	-	6.0	-	
.8	Other	%	11	5	2.0	27	
8	Gas End Use						
.1	Space heating	%	-	49	26.0	32	
.2	Domestic Hot Water	%	-	8	23.0	12	
.3	Pool Heating	%	-	-	6.0	3	100
.4	Sterilization	%	-	-	-	6	
.5	Laundry	%	-	-	13	-	
.6	Other	%	-	43	32	46	

(1) T = Tenant (2) BB = Base Building

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TABLE 3. DEFINITION OF TOTAL SPACE HEATING AND DOMESTIC HOT WATER HEAT REQUIRED BASED ON TABLE 2

Parameter				**SAOB**		**Hotels**	**Hosp.**
No.		**Description**	**Unit**	**T**	**BB**		
1		Electrical Heating					
	.1	Total electrical, derived from Table 2, Item 4	$kWh/m^2.a$	104	113	275	220
	.2	Heating Proportion, Table 2, Items 7.6 & 7.7	%	3	2	15	2
	.3	Total electrical space heat required, 1.1 x 1.2	$kWh/m^2.a$	3	2	41	4
2		Gas Heating					
	.1	Total gas, Table 2, Item 5	$kWh/m^2.a$		23	153	22.7
	.2	Water heating proportion, Items 8.1 to 8.5, minus 8.4	%		57	68	54
	.3	Total gas water heating, 2.1 x 2.2	$kWh/m^2.a$		13	104	123
	.4	Hot water boiler efficiency	%		80	80	80
	.5	Available heat in hot water	$kWh/m^2.a$	-	10	83	98
3		Total Heat Water Heating Load	$kWh/m^2.a$				
	.1	Item 1.3 above	$kWh/m^2.a$	3	2	41	4
	.2	Item 2.5 above	$kWh/m^2.a$	-	10	83	98
	.3	Total water heating required	$kWh/m^2.a$	3	12	124	102
	.4	Total water heating required, 3.3 x 3.6	$MJ/m^2.a$	11	43	446	367
	.5	Total space heating required, Item 1.3	$MJ/m^2.a$	11	7	148	14
	.6	Total heat required	$MJ/m^2.a$	22	50	594	381
4		Specific Global Warming Emissions (GWE)					
	.1	Electrical GWE @ $0.8kg/kWh/m^2.a$	$kgCO_2.e/m^2.a$	104	113	275	220
	.2	Natural gas GWE @ 59.3 kg/GJ	$kgCO_2.e/m^2.a$	5		33	5
	.3	Total specific GWE	$kgCO_2.e/m^2.a$	222		308	225

therefore the HVAC CO_2 compressors Total Heat Rejection (THR) may be used to heat water. This is entirely different to the conventional refrigerants (R717, R22, etc.) where up to 90% of the THR is rejected at a constant temperature during the condensing phase.

In Australia mains water temperatures rarely exceed 20°C enabling a gas cooler exit temperature of 23°C. As is clear from Fig. 5 and Fig. 6, at these conditions the COPs of CO_2 compressors are high. Fig. 5 is based on Direct Expansion (DX) CO_2 for AC cooling at an SST of 10°C, whilst Fig. 6 is based on 5°C SST for chilled water. If higher hot water temperatures are required, the compressor discharge temperature may be increased by lifting the discharge pressure or increasing the amount of suction superheat or a combination of both.

C. *Examination of additional possibilities*

Fig. 8 shows the proportion of latent and sensible heat to be disposed of in the condenser gas cooler. At 30°C Saturated Condensing Temperature (SCT) the amount of sensible heat to be removed from the compressor discharge is 68% of the total heat leaving only 32% of the heat to be removed as latent heat when condensing at 30°C. The advantage of this is that a considerable amount of heat may be removed by aircooling thus saving water in evaporative condenser/gas coolers and cooling towers.

Fig. 9 shows an interesting phenomenon in industrial air blast freezing. A time limited freezing cycle requires a combination of air temperature and air velocity, which increase and reduce together, i.e. as the air temperature reduces, the air velocity reduces. But as the air velocity increases the fan EE consumption increases as the cube ratio of the air velocity. Thus if the air velocity, i.e. the air volume, reduces by 25% the fan EE reduces to 42.2% (0.753) saving 57.8% of the fan EE. Invariably the fan EE becomes heat to be removed. Thus a reduction in air volume not only reduces fan EE consumption but also the refrigeration load. But when the air velocity reduces a lower air temperature is required necessitating a lower evaporating temperature which reduces the compressor COP. Fortunately this is compensated by the reduced refrigeration capacity. Therefore the minimum EE consumption of a

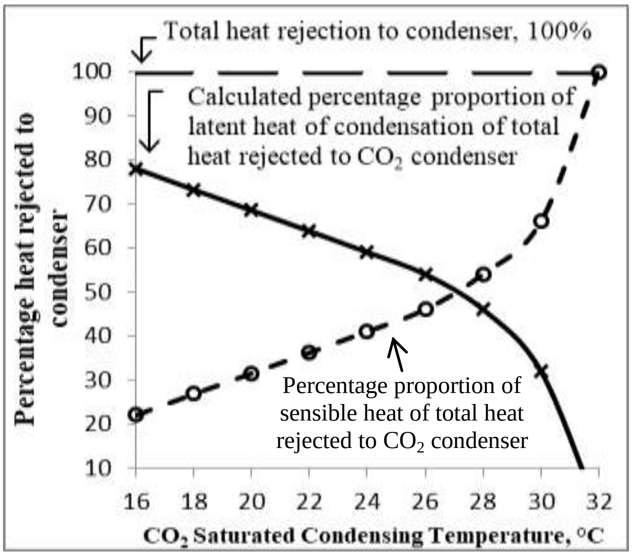

Figure 8. Heat rejection profile for a commercial compressor at 5°C satd. suction, 5 K useful suction superheat and 5°C CO_2 liquid temp.

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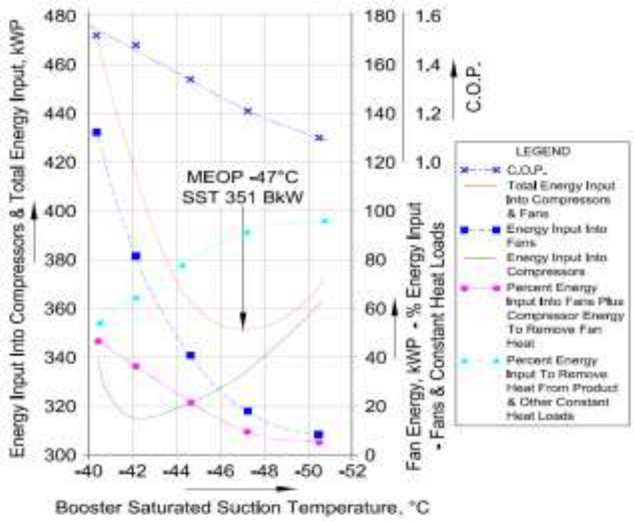

Figure 9. Total blast freezer energy demand variation with saturated suction temperature due to reducing air temp. and air velocity

freezing process is when the sum total of compressor and fan EE is a minimum. The irony is that the compressor COP for the minimum energy consumption of the system is much lower than when conventional design practices are followed.

The same problem exists in AC systems where there is a strong emphasis on compressor COP. This invariably means that a high chiller Saturated Suction Temperature is chosen. This leads to relatively high water temperatures with low ranges. Adding to that the almost obsessive practice to design for maximum Net Lettable Area in a building frequently reduces the space for ducting. This results in high air velocities in ducts requiring high fan EE consumption. This is clearly shown in Fig. 2 where 80% of the parasitic heat loads are contributed by the supply and return fans!

In Fig. 10 and Fig. 11 bars A show the existing situation in Melbourne office buildings and hospitals. Bars B show the total energy consumption when replacing existing compressors, having COPs developed in Table 1, with CO_2 compressors which far superior COPs, as shown in Fig. 4 and Fig. 5. A lot of the heat required in a building may be recovered from the CO_2 compressor discharge saving a lot of FL energy and cooling water as the amount of heat

TABLE 4. EVALUATIONS OF COOLING CAPACITY, TOTAL HEAT REJECTION TO CONDENSER AND HVAC ENERGY WITH CO_2

Parameter				**SAOB**		**Hotels**	**Hosp.**
No.		**Description**	**Unit**	**T**	**BB**		
1		Electrical Energy (EE) HFC cooling					
	.1	Total, Item 4, Table 2	kWh/m^2.a	104	113	275	220
	.2	HVAC proportion	%	18	67	52	47
	.3	HVAC EE use, 1.1 x 1.2	kWh/m^2.a	19	76	143	103
	.4	Compressor energy, % of total from Fig. 2, excluding heating	%	50	50	50	54
	.5	Compressor energy use 1.3 x 0.4	kWh/m^2.a	10	38	72	56
	.6	COP from Table 1		3.22	3.22	3.3	3.35
	.7	Refrigeration Cap. 1.5 x 1.6	kWh/m^2.a	32	122	238	188
	.8	Total heat rejection, 1.5 + 1.7	kWh/m^2.a	42	160	310	244
	.9	Total heat rejection, 1.8 x 3.6	MJ/m^2.a	151	576	1,116	878
2		CO_2 Cooling					
	.1	Capacity required	kWh/m^2.a	32	122	238	188
	.2	COP @ 5°C SST/+24°C SCT from Figure 4		6.38	6.38	6.38	6.38
	.3	EE consumption CO_2 compressor	kWh/m^2.a	5	19	36	30
	.4	EE consumption HFC system – Item 1.5	kWh/m^2.a	10	38	72	56
	.5	EE reduction with CO_2 plant	kWh/m^2.a	5	10	36	26
	.6	EE reduction with CO_2 plant	MJ/m^2.a	18	68	130	94
	.7	CO2 compressor heat rejection 2.1 + 2.3	kWh/m^2.a	37	141	274	218
	.8	CO2 compressor heat rejection 2.7 x 3.6	MJ/m2.a	133	508	986	785
	.9	Total heating load, Item 3.6 Table 3	MJ/m^2.a	22	50	594	381
	.10	Surplus compressor heat, 2.8 – 2.9	MJ/m^2.a	111	458	392	404
	.11	Total energy savings, 2.6 + 2.9	MJ/m^2.a	40	118	724	475
	.12	Present energy consumption	MJ/m^2.a	870		1,549	1,615
	.13	CO_2 energy consumption, 2.12 – 2.11	MJ/m^2.a	712		825	1,140
	.14	Saving %, 2.11 ÷ 2.12	%	18.2		46.7	29.4
3		Reduction in energy use and emissions					
	.1	Electric heating reduction, Table 3 Item 1.3	kWh/m^2.a	5		41	4
	.2	Compressor EE reduction, Table 4 Item 2.5	kWh/m^2.a	15		36	26
	.3	Total reduction in EE, 3.1 + 3.2	kWh/m^2.a	20		77	30
	.4	GWE from EE, @ 1 kg/kWh	kgCO_2-e/m^2.a	20		77	30
	.5	Total gas water heating reduction	MJ/m^2.a	47		374	44
	.6	GWE reduction @ 59.3 kg/GJ	kgCO_2-e/m^2.a	3		22	3
	.7	Total reduction in GWE, kg 3.4 + 3.6	kgCO_2-e/m^2.a	22		99	33
	.8	Current GWE, Table 3 , Item 4.3	kgCO_2-e/m^2.a	222		308	225
	.9	GWE reductions, 3.7 ÷ 3.8	%	10		32	14.7

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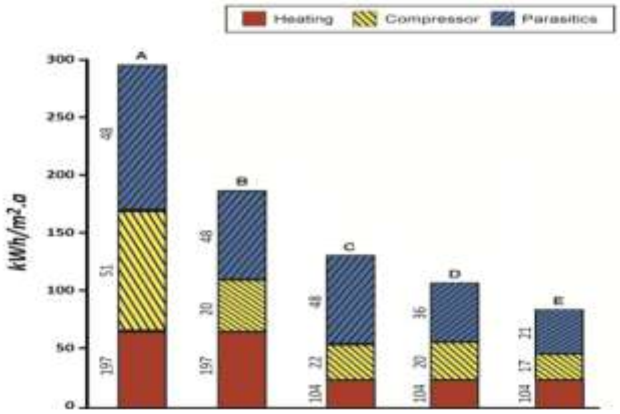


A = Existing situation in Hospitals in Melbourne
B = Retrofit CO_2 compressor
C = Retrofit CO_2 heat recovery
D = Slow down fans
E = New Building with pumped CO_2 and slow fans

Energy Consumer	A	B	C	D	E	Remarks
Parasitics	48	48	48	36	21	
Compressor	51	20	22	20	17	
Heating	197	197	104	104	104	Gas
Total	296	265	174	160	142	
Reduction $kWh/m^2.a$	-	31	122	136	154	
Reduction %	-	10.5	41.2	45.9	52.0	

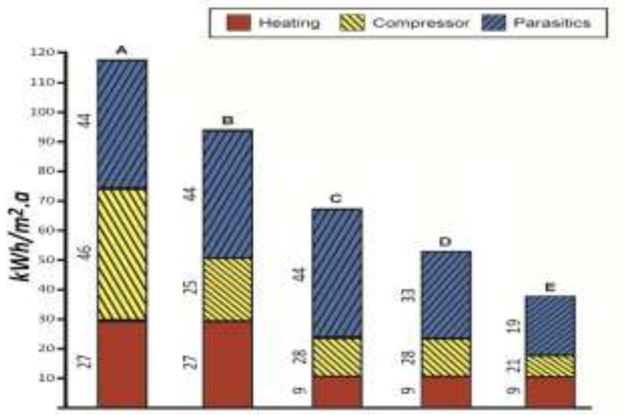
Figure 11. Reduction in energy consumption when retrofitting CO_2 cooling and heating to existing & equipping new hospitals in Melbourne with transcritical CO_2 cooling and heating systems

rejected to an evaporative condenser/gas cooler or cooling tower reduces. The energy reduction benefits are shown in bars C. Bars D show the benefit of reducing the air volume by slowing down the fans 25%. Bars E show the results if we pump liquid CO_2 around the building to DX evaporators in the Air Handling Units in new buildings. Retrofitting CO_2 refrigeration with heat recovery would reduce the energy consumption by about 42% in existing buildings and 45% in hospitals in Melbourne.

D. *SUPERMARKETS* *See Table 5*

In Australian supermarkets Electrical Energy Consumption (EEC) accounts for about 99% of all supermarket energy cost. In this discussion the remaining 1% other fuel consumption is ignored. Supermarkets in Australia have little need for heating but they are invariably cooled, which on average consumes about 35% of all electrical energy. MT and LT Refrigeration together consume about 40% of all supermarket EE, 25% by MT Refrigeration and the balance of 15% by LT Refrigeration. The remaining 25% is used for all other purposes like lighting, equipment, etc. Australian supermarkets are notoriously inefficient in terms of energy consumption. They consume more than twice the amount of energy per m^2 than hospitals do and about four times as much as office buildings.

Because presently most sales coolers and refrigerated cabinets are designed for HFC refrigerants it is difficult to retrofit CO_2 refrigeration to supermarket refrigeration. In



A = Existing situation in Office Buildings in Melbourne
B = Retrofit CO_2 compressor
C = Retrofit CO_2 heat recovery
D = Slow down fans
E = New Building with pumped CO_2 and slow fans

Energy Consumer	A	B	C	D	E	Remarks
Parasitics	44	44	44	33	19	
Compressor	46	25	28	25	21	
Heating	27	27	9	9	9	Gas
Total	117	96	81	67	49	
Reduction $kWh/m^2.a$	-	21	36	50	68	
Reduction %	-	17.9	30.8	42.7	58.1	

Figure 10. Reduction in energy consumption when retrofitting CO_2 cooling and heating to existing & equipping new buildings in Melbourne with transcritical CO_2 cooling and heating systems

the past nearly 15 years there has been an increasing trend to CO_2/HFC cascade systems and several CO_2/ammonia cascade systems. The ultimate solution to improve the energy efficiency of supermarket refrigeration is at hand in the form of Multi Function 2 Stage Trans Critical CO_2 Systems with Parallel Compression ($MF2STCCO_2RSPC$) systems in AC, MT and LT Refrigeration duties. They will be water cooled and would operate subcritically 100% of the time in 90% of Australia where the 1% ambient wet bulb design temperature does not exceed 24°C and most of the time in the remaining 10% of Australia. This is also very relevant to the other building types. In the following Table this proposition will be examined for 90% of Australia.

There are no firm figures available on the amount of HFC refrigerant losses from supermarket refrigerating systems in Australia. In the USA annual refrigerant losses in supermarkets are estimated at 25% of the system charge [5]. The large manufacturers of HFCs budget for annual refrigerant losses of 16% in their systems (Dupont, Honeywell, Arkema) [6]. USA supermarkets are inherently much more energy efficient than Australian supermarkets with a total energy consumption of 2,442 $MJ/m^2.a$, 27.5% less than Australian supermarkets. 81% of USA supermarket energy use is electrical energy (549 $kWh/m^2.a$) and the balance of 19% is natural gas (466 $MJ/m^2.a$). The R404A leaks from USA supermarkets contribute 409 kg of CO_2-e/m^2 out of a total of 629 $kgCO_2$-e/m^2 of USA supermarket floor area. Thus 65% of GWE from USA supermarkets is due to refrigerant leaks.

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TABLE 5. EVALUATION OF ENERGY CONSUMPTION BENEFITS IN AUSTRALIAN SUPERMARKETS FROM TOTAL INTEGRATED SUBCRITICAL CO_2 REFRIGERATING SYSTEMS IN 90% OF AUSTRALIA

Parameter				System Type				
				Existing HFC		kWR	Integrated CO_2	
No.	Sub	Description	Unit	Energy	COP	Refrig Comp.	COP	Comp. kWP
1		Specific energy consumption	MJ/m2.a	3,375	-			
2		Specific energy consumption, 1÷3.6	kWh/m2.a	938	-			
3		Electrical energy	%	99	-			
4		Qty. electrical energy, 100%	kWh/m2.a	929	-			
5		Base load & parasitic, 25%	kWh/m2.a	232	-			
6		Total refrigeration	kWh/m2.a	697				
	.1	AC air-cooled DX, 35% (5)	kWh/m2.a	326	3.5	1,141		
	.2	MT air-cooled DX, 25% (5)	kWh/m2.a	232	3.25	754		
	.3	LT air-cooled DX, 15% (5)	kWh/m2.a	139	1.25	174		
7		CO_2 AC compressor +10 SST/+24 SCT avg. from 6.1				1,141	9.5	119
8		CO_2 LT refrig. cap. −30/−5 from 6.3				174	4	44
9		MT refrig. cap. from 6.2				754	5.3	142
10		MT Hi Stage for 8, Item 6.3	kWR + kWP			218	5.3	41
11		Total CO_2 compressor, 7+8+9+10	kWP			-	-	347
12		Existing compressor, 6.1+6.2+6.3	kWP					697
13		Compressor EE reduction	kWP					351
14		Compressor EE reduction	%					50.4
15		% EE reduction total operation	%					37.8
16		Emissions						
	.1	Current GWE from electrical energy 929 kWh/m^2.a @ 1.0 kg/kWh						= 929 kg/m^2.a
	.2	Reduction in GWE from electrical energy 351 kWh/m^2.a @ 1.0 kg/kWh						= 351 kg/m^2.a
	.3	Reduction in supermarket GWE, %						= 37.8%

IV. KIGALI AMENDMENT TO THE MONTREAL PROTOCOL

The Kigali Amendment to the Montreal Protocol was agreed to on 15 October 2016. The Agreement is summarized in Table 6. Fig. 12 shows the rate and timeframe for the HFC reduction in the 50 developed countries – 45 countries starting in 2019 and 5 starting in 2020.

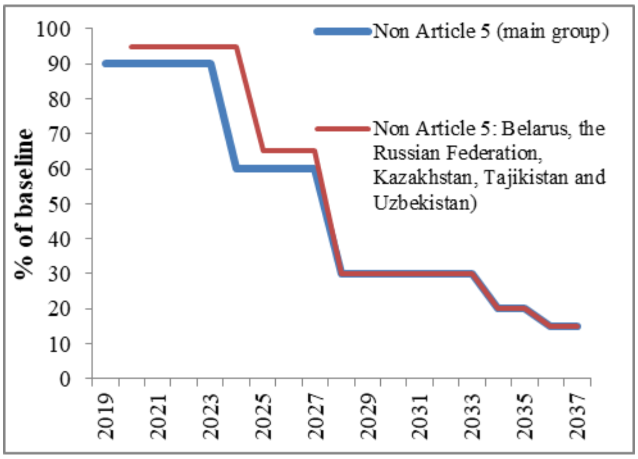

Figure 12. HFC phase down schedule for developed countries

Fig. 13 shows the same for the 137 Group 1 developing countries and the 10 Group 2 developing countries, commencing respectively with a freeze in 2024 and 2028 with the first mandated reductions four years after the freeze start.

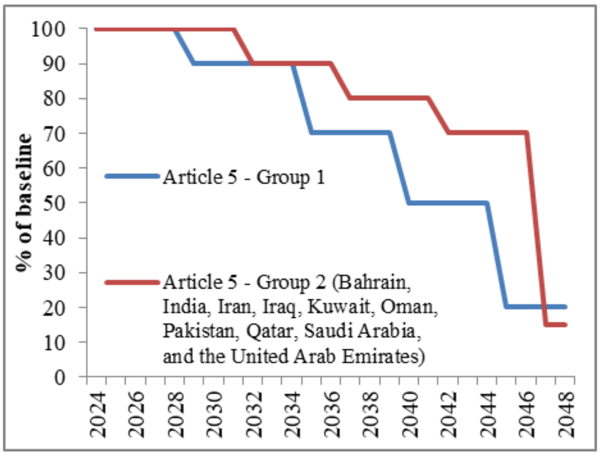

Figure 13. HFC phase down schedule for Groups 1 & 2 Article 5 developing countries

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TABLE 6. PHASE-DOWN SCHEDULE FOR HFCS IN ARTICLE 5 AND NON-ARTICLE 5 PARTIES

	Non-A5 parties (developed countries)	**A5 parties (developing countries) -** *Group 1*	**A5 parties (developing countries) -** *Group 2*
Baseline formula	Average HFC consumption levels for 2011-2013 + 15% of HCFC baseline – see Fig. 12	Average HFC consumption levels for 2020-2022 + 65% of HCFC baseline – see Fig. 13	Average HFC consumption levels for 2024-2026 + 65% of HCFC baseline
Freeze	-	2024	2028
1st step	2019 – 10%	2029 – 10%	2032 – 10%
2nd step	2024 – 40%	2035 – 30%	2037 – 20%
3rd step	2029 – 70%	2040 – 50%	2042 – 30%
4th step	2034 – 80%		
Plateau	2036 – 85%	2045 – 80%	2047 – 85%

* For Belarus, Russian Federation, Kazakhstan, Tajikistan, Uzbekistan, 25% HCFC component of baseline and different initial two steps (1) 5% reduction in 2020 and (2) 35% reduction in 2025

V. 6. CONCLUSIONS

It is clear that retrofitting subcritical CO_2 refrigerating systems for cooling and heating into office buildings, hotels, hospitals and supermarkets has significant energy consumption reduction benefits plus attendant reductions in both indirect and direct Global Warming Emissions. Furthermore, the amount of heat rejected to the atmosphere reduces by 20% in the case of SAOBs to 64% for hotels and 54% for hospitals. Subcritical CO_2 discharge vapor has a high proportion of sensible heat. Thus we estimate the reduction in cooling tower cooling water at 50% in the case of SAOBs, 80% for hotels and 65% for hospitals with the use of hybrid CO_2 evaporative condensers which remove up to 50% of the heat in the CO_2 compressor discharge vapor by air cooling with the balance of the heat being removed in the evaporative condenser suction. Equipping new facilities with water cooled CO_2 systems for cooling and heating future proofs such buildings against the potentially severe impact of the implementation of the Kigali Amendment by the MP.

Much greater energy savings would be possible in new buildings if the CO_2 were pumped around the buildings in a Liquid Recirculation fashion. This would permit lifting the evaporating temperature which would increase the compressor COP. But perhaps more importantly it would be possible to reduce airflow, which would drastically reduce the energy consumption of the supply and return fans and thus reduce the heat load on the AC compressors. The adoption of CO_2/HFC cascade systems by Australian supermarkets since 2003 was largely driven by the high cost of replacing CFC and HFC refrigerants with attendant GWEs.

About 90% of Australia's population lives South of a line from Geraldton, Western Australia through Alice Springs, Northern Territory to the NT/Queensland boarder and from there South of a line through Kingaroy, Queensland to Ballina, New South Wales. The Ambient Wet Bulb Design Temperature (AWBDT) in this southern half of Australia does not exceed +24°C, thus subcritical CO_2 refrigeration with an evaporative condenser is possible during virtually 100% of the year.

Nomenclature

a	annum (year)
AC	Air Conditioning
AEC	Annual Energy Consumption
AWBDT	Ambient Wet Bulb Design Temperature
BB	Base Building
CFC	Chloro Fluoro Carbon
CO_2	Carbon Dioxide
CO_2-e	Carbon Dioxide equivalent
COP	Coefficient of Performance
°C	Degrees Celsius
DHW	Domestic Hot Water
DX	Direct Expansion
EE	Electrical Energy
EEC	Electrical Energy Consumption
GC	Gas Cooler
GJ	Gigajoule - 10^{12} Joules
GWE	Global Warming Emissions
HFC	Hydro Fluoro Carbon
HVAC	Heating, Ventilating, Air Conditioning
K	degree Kelvin
kWh	kilowatt hour
kWP	Compressor EE consumption
kWR	Refrigeration capacity
LPG	Liquid Petroleum Gas
LTR	Low Temperature Refrigeration
m^2	Square Meters
m^3/h	Cubic Meters per Hour
MJ	Mega joule – 10^6 Joules
MTR	Medium Temperature Refrigeration
NH_3	Ammonia
PJ	Peta joule - 10^{15} Joules
SCT	Saturated Condensing Temperature
SSH	Suction Super Heat
SST	Saturated Suction Temperature
T	Tenant
THR	Total Heat Rejection

"…significant reductions in the electrical energy, natural gas and cooling water consumption may be achieved when using transcritical CO_2 refrigerating systems for building cooling, heating and sanitary hot water"

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