A CASE STUDY INTO THE APPLICATION OF CO_2 COOLING AND HEATING IN AMERICAN OFFICE BUILDINGS

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ABSTRACT

A major benefit of transcritical CO_2 refrigeration is the gliding temperature available when cooling the transcritical fluid from the final high temperature at completion of compression to as low a temperature as practicable. This property may be exploited when requiring simultaneous cooling and heating in building air conditioning applications. The CO_2 compressor discharge can usually heat enough water for heating, reheating and hot water purposes thereby obviating the need for a boiler and its fuel supply.

Using USA Department of Energy data for energy consumption by USA office buildings it is shown that CO_2 cooling reduces primary energy consumption, energy cost, water consumption at the buildings and CO_2 greenhouse gas emissions by an estimated 55%, 57%, 75% and 53% respectively in the existing building stock when coupled with a 25% reduction in supply and return duct air velocities, and exhaust duct air velocities. In the case of new buildings these beneficial reductions would be about 63%, 65%, 79% and 61% respectively. Incorporation of energy recovery from exhaust air and economizing cycles will produce additional reductions in all three areas. The absence of cooling towers eliminates the danger of Legionella disease. The absence of HFC refrigerants eliminates the danger of high GWP fugitive gases escaping.

1.INTRODUCTION

Westphalen et al (1999) prepared a report dealing with Thermal Distribution, Auxiliary Equipment and Ventilation in the Energy Consumption Characteristics of Commercial Building 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 4.85×10^9 GJ. See Fig. 1 [Westphalen et al (2001)].

Figure 2 [Westphalen et al (1999)], shows that of the total Parasitic Primary Energy use in HVAC systems in the office building category, the supply and return air fans consumed 43% of a total of 0.39×10^9 GJ i.e. 0.17×10^9 GJ. The exhaust fans consumed a further 37% of parasitic primary energy, i.e. 0.14×10^9 GJ.

In Table 1, seasonal efficiencies for different cooling equipment and the distribution of the equipment as applied in the office building category [Westphalen et al (1999)] have permitted the evaluation of a weighted COP for the entire office building category.

2. PRINCIPLES OF REDUCTION IN ENERGY CONSUMPTION

2.1 Building Heat Load Components and AC Plant and Heating

The heat loads generated in a building are due to heat gains through the building structure windows, power consumed by office machines, lighting, air leakage into the building, ventilation air, people and so on.
Once the heat load has been determined, an air conditioning plant usually comprising both heating and cooling functions is designed and installed. The compressor heat is rejected to atmosphere via a cooling tower providing cooling water to a condenser or via an air cooled condenser. Chilled and hot water is pumped around to Air Handling Units (AHU's) for cooling and reheating or heating only during Winter. Fans circulate air through ducts and return air fans return air to the AHU's. Exhaust fans remove a minimum quantity of air, which is usually regulated by public health authorities

Figures 1 and 2 show that the fans constitute a very significant energy consumption in buildings. It is worthy of note that the fans are installed in the AHU's and return air streams and consequently all the energy supplied to the fan motors, including losses, shows up as heat in the air stream, which thus increases the heat load to be handled by the compressor.

2.2 Reduction in Energy Consumption

To reduce energy consumption in buildings it is necessary to reduce the amount of heat generated during hot weather by all constituent components and ensure that the cooling plant runs at the Most Energy Efficient Operating Point (MEEOP).

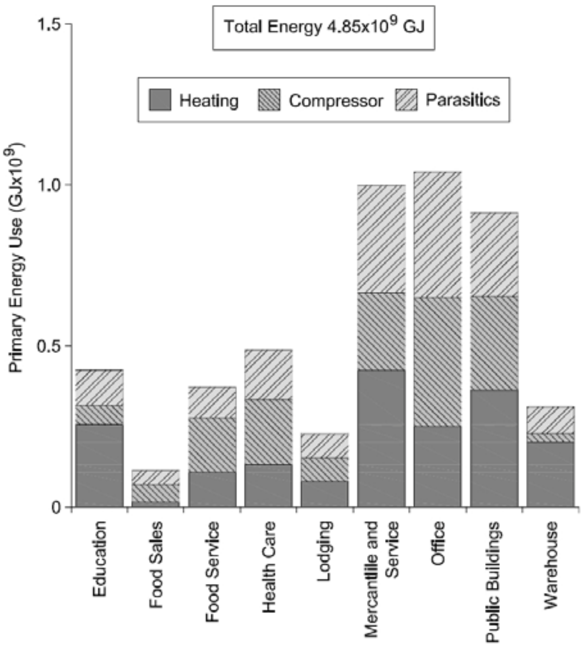


Figure 1 Total HVAC Primary Energy Use by Building Type

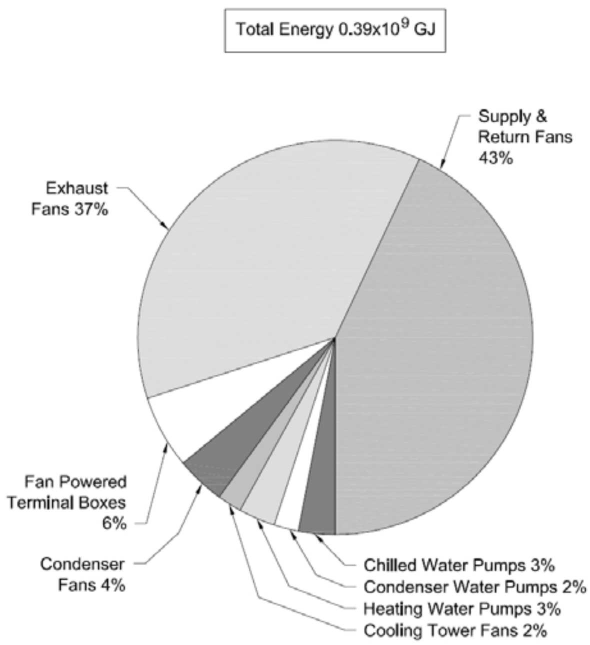


Figure 2 Parasitic Primary Energy use by Type of Equipment

Table 1 Equipment Seasonal Efficiencies– Coefficients of Performance (COP's)

Equipment Type Width Adjustment	COP(1)	COP Correc–tion Factor for Fans	COP Corrected	Office Area Applied to (2)		Weighted Contri-bution to Mean COP
				m^2 x 10^6	%	
Centrifugal chiller	4.4	1	4.4	135	15	0.66
Water cooled screw chiller	4.14	1	4.14	18	2	0.083
Water cooled recip chiller	3.91	1	3.91	27	3	0.117
Air cooled screw chiller	3.17(3)	1.1(5)	3.49	36	4	0.140
Air cooled recip chiller	2.71(3)	1.1(5)	2.98	144	16	0.477
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
Packaged AC	2.13(4)	1.3(6)	2.9	351	39	1.13
Heat pump	2.13(4)	1.3(6)	2.9	62	7	0.203
Absorption chiller	0.98	1.0	0.98	9	1	0.001
TOTAL				899	100	3.22

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

The supply and return fans constitute a large energy consumer and add a significant heat load to the system as well. The fan laws state that a 25% reduction in air flow reduces the energy consumption to 0.75^3 i.e. a reduction in energy consumption of $1 – 0.75^3 = 57.8\%$. There is also a corresponding decrease in compressor power requirement equivalent to the fan power reduction divided by the COP.

In existing buildings, a reduction in air flow needs to be compensated for by a lower air temperature which in turn requires a lower chilled water and/or refrigerant evaporating temperature To. A lower To reduces the compressor COP, which would tend to increase compressor specific energy consumption. In new buildings, it would be possible to design ductwork for lower velocities, thereby saving considerable energy.

Initially, the reduction in fan and resulting compressor energy consumption is greater than the increase in compressor energy consumption due to a reduction in the compressor COP. This constitutes an optimization problem identical to that found by Visser (1976) when blast freezing export meat cartons. See Figure 3. This shows that it is frequently more energy efficient to select a compressor with a lower COP which needs to handle a much lower heat load. This principle is not widely understood, but universally applicable in cold air chilling and blast freezing systems, including air conditioning cooling, where the energy consumed by auxiliary equipment such as fans, chilled water pumps and refrigerant pumps is converted to heat to be removed by the compressor.

3. CARBON DIOXIDE (CO_2) REFRIGERATION

As will be shown, CO_2 is the ideal refrigerant for the cooling and heating of buildings.

The properties of CO_2 relevant to this paper are its low critical temperature of 31.1^0C, a high volumetric refrigeration capacity due to the high vapour densities and a large pressure differential per degree C. CO_2 is non-toxic, non-flammable and has a GWP = 1. It is a totally natural substance with the average adult human producing about 1 kg/day.

Figure 4 shows the COP of sub critical CO_2 compressor performance as extracted from a CO_2 compressor manufacturer's rating tables. Dorin (2007). Figure 5 shows the COP of transcritical CO_2 compressor performance again from the Dorin (2007) compressor ratings.

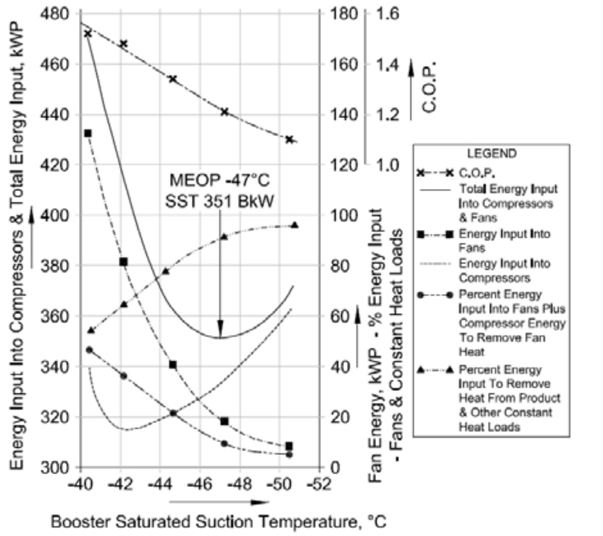


Figure 3 Total Blast Freezer Energy Demand Variation with Saturated Suction Temperature due to Reducing air temp and Air Velocity

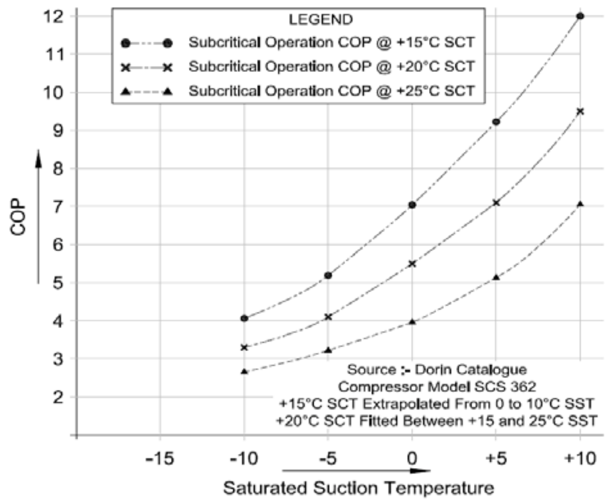


Figure 4 CO_2 Compressor COP Values Variation with Sat Suction and Condensing Temp

4. AUSTRALIAN CAPITAL CITY AMBIENT DRY AND WET BULB TEMPERATURES

Dry and Wet Bulb Temperature (DWBT) data was obtained form the Australian Bureau of Meteorology (2008) for all Australian Capital cities. See Table 2.

In Figure 6, the ambient DWBT occurrence has been plotted from a total of 199,520 data readings taken at Sydney International Airport between 6am and 10pm for the past 10 years. We have selected a Sydney design condition of $+35^0$C dry bulb, $+23^0$C wet bulb, which are exceeded only 0.2% and 0.3% of the time respectively. After Brisbane, Sydney has the second most humid climate and is Australia's largest city. Sydney was therefore selected as the city for which to project potential energy and water savings to be had from CO_2 refrigerated cooling in both existing and new buildings.

In Table 3, we have evaluated the Seasonally Weighted COP (SWCOP) based on the Sydney climate. An advantage of CO_2 is its high pressure, which allows close temperature approaches between condensing temperature and cooling air or water. The lower the ambient temperature, the higher the COP and the higher the heating load will be in addition to hot water requirements, but the lower the cooling load will be. It has been assumed that on average 25% of the plant capacity would run in transcritical mode to provide heat for all purposes.

Table 2 Ambient AC Design Dry and Wet Bulb Conditions – Australian Capital Cities 6am – 10pm

City	DB, ^{0}C	WB, ^{0}C	City	DB, ^{0}C	WB, ^{0}C
Adelaide	37	22	Hobart	30	19
Brisbane	33	25	Melbourne	35	21
Canberra	34	20	Perth	37	22
Darwin	35	29	Sydney	35	23

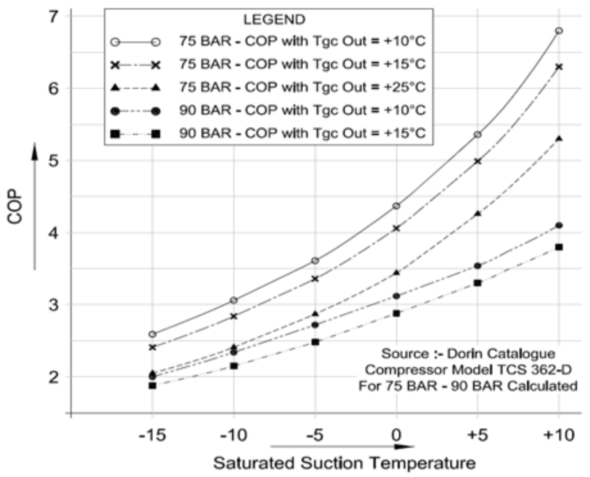


Figure 5 CO_2 Compressor COP Value Variation with Sat Suction Temp, Transcritical Discharge Pressure and Gas Cooler Outlet Temperatures

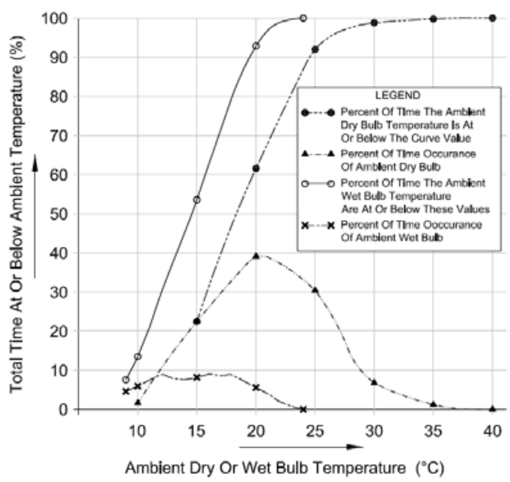


Figure 6 Ambient Dry & Wet Bulb Temperature Profile – Sydney Airport, 6am – 10pm

Table 3 Evaluation of Weighted COP with Ambient Temp Conditions for 12 Months Running of CO_2 Cooling in Sydney in:

a. Existing Buildings with +5^{0}C Evaporating Temp, Water Chiller Retrofitted to Existing Chilled Water Systems. See Figure 7

Ambient Temp ^{0}C	Sat Cond Temp., ^{0}C Avg	COP from Figs 4 & 5	Refrig load % of full load	Weighted % of time from Fig 6	Seasonally Weighted COP
Up to +15	+15	9.2	85	12.2	0.95
16 – 20	+20	7.1	90	21.3	1.36
21 – 25	+25	5.1	95	16.5	0.80
26+trans@75bar,+25^{0}C gas cooler out	–	4.3	100	50	2.15
Totals				**100**	**5.36**

b. New Buildings with 10^{0}C pumped CO_2 to Air Handling Units. See Figure 8

Up to +15	+15	12	85	12.2	1.24
16 – 20	+20	9.5	90	21.3	1.81
21 – 25	+25	7.0	95	16.5	1.1
26+trans@75bar,+25^{0}C gas cooler out	–	5.3	100	50	2.65
Totals				**100**	**6.80**

5. EVALUATION OF EXISTING SYSTEMS PERFORMANCE

Using primary energy data from Westphalen et al (1999) and (2001), and the weighted COP from Table 1, the total electrical energy consumption heat removed during cooling and heat rejected by the cooling system were calculated for the entire USA office building stock as shown in Table 4.

Table 4 Evaluation of Power Consumption of, Cooling Heat Removal from And System Heat Rejected by Existing Cooling and Heating Systems in USA Office Buildings

Parameter	Electric Energy Consumer			Total
	Comp-res-sors	Parasitics		
		Supply, Return Extract fans	Other Fans, Pumps, etc	
Primary energy use, Quads (1) (2)	0.378	0.297	0.076	0.751
Primary energy, GJ x 10^9 (2)	0.4	0.31	0.08	0.79
Multiplier GJ to kWhrs @ 31% of energy delivered to the consumer (1) (3) (4)	86.12	86.12	86.12	86.12
Power consumption, kWhrs x 10^9	34.44	26.7	6.9	68.04
Weighted COP from Table 1	3.22	–	–	3.22
Total heat removed during cooling, GJ x 10^9	0.399	0.096	–	0.399
Total heat rejected, GJ x 10^9	0.523	–	–	0.523

1. Source: Westphalen et al (2001)
2. 1 Quad = 10^{15} BTU's = 1.0551 x 10^9 GJ
3. 1 kWhr = 11,055 BTU's/hr = 11.611 MJ Primary Energy
4. 1 GJ = 277.77 kWhrs

Table 5 Evaluation of Beneficial Impact of Transcritical CO_2 Cooling on Annual Energy Consumption

Parameter	System Type Number			
	1 (1)	2 (2)	3 (3)	4 (4)
Cooling load excl. parasitics, GJ x 10^9	0.303	0.303	0.303	0.303
Cooling load from Fans, GJ x 10^9, variable	0.096 (5)	0.096 (5)	0.041 (6)	0.041 (6)
Other parasitics, GJ x 10^9 – Constant	0.025	0.025	0.025	0.025
Total cooling load, GJ x 10^9	0.399 (5)	0.399 (5)	0.344	0.344
Evap. Temp, ^{0}C	Existing	+5	+2	+10
COP Compressor	3.22 (5)	5.36	4.5	6.8
Comp. Power consumption, kWhr x 10^9	34.42 (5)	20.68	21.23	14.05
Add parasitics, kWhrs x 10^9	33.61 (5)	33.61 (5)	18.33	18.33
Total energy con, kWhrs x 10^9	**68.03 (5)**	**54.29**	**39.56**	**32.38**
Primary energy, GJ x 10^9	0.79	0.63	0.46	0.38
Add heating primary energy, GJ x 10^9	0.25	–	–	–
Total annual primary energy, GJ / 10^9	1.04	0.63	0.46	0.38
Primary energy saving, GJ x 10^9 / %	–	**0.41/39.4**	**0.58/55.8**	**0.66/63.5**
Total heat rejected, GJ / 10^9	0.523	0.223	0.17	0.145
Building area, m^2 x 10^9	0.899	0.899	0.899	0.899
Primary energy use intensity, GJ /m^2	1.157	0.701	0.51	0.423

(1) System type number 1 = Existing systems
(2) System type number 2 = Existing systems with transcritical CO_2 retrofit TC CO_2R
(3) System type number 3 = Existing systems with TC CO_2 R & 25% less air
(4) System type number 4 = New buildings with liq recirc. TC CO_2 & 25% reduced air duct velocity
(5) From Table 4
(6) Reduced parasitic fan heat load

6. EVALUATION OF ANNUAL REDUCTIONS IN ENERGY CONSUMPTION, TOTAL HEAT REJECTED AND PRIMARY ENERGY USE AND USE INTENSITY

Using the value of parameters evaluated in Table 4 permitted computation of the performance of systems using transcritical CO_2 cooling. We know the cooling load and weighted compressor COP for the existing systems. When slowing the supply, return and exhaust fans down by 25% allowed us to calculated the new parasitic fan heat load, which gave us the heat load for the CO_2 refrigerated systems. Using the weighted COP's evaluated in Tables 3a and 3b permitted calculation of the total energy consumption for CO_2 refrigerated systems in the buildings with the same air flow and with reduced air flow and, finally, installed in a new building with purpose designed and built transcritical CO_2 cooling and heating systems. See Table 5.

7. EVALUATION OF ENERGY COST SAVINGS AND CORRESPONDING REDUCTIONS IN CO2 EMISSIONS, AND REDUCTIONS IN COOLING WATER CONSUMPTION

In Table 6, the total annual energy costs have been evaluated for the different CO_2 systems. And compared with existing systems. Savings have been estimated in dollars / m^2 and percentage and the total energy intensity in annual consumption per unit area. Similarly, the resulting reduction in global warming CO_2 production have been estimated and finally, the reduction in cooling water consumption due to the use of air cooled CO_2 gas coolers has also been estimated.

Table 6 Evaluation of Operating Cost Savings in Office Building

Parameters	System Type Number			
	1	2	3	4
Electrical energy consumption, kWhrs x 10^9 (1)	68.03	54.28	39.66	32.38
Gas consumption, GJ / 10^9 (1)	0.25	–	–	–
Electrical energy @ $0.08kWhr, $ x 10^9	5.44	4.34	3.17	2.58
Gas @ $8.00/GJ, $ x 10^9	2.00	–	–	–
Total energy cost, $ x 10^9	7.44	4.34	3.17	2.58
Energy cost savings, $ x 10^9	–	3.10	4.27	4.86
Building area, m^2 x 10^9	0.899	0.899	0.899	0.899
Annual energy cost, $/$m^2$	8.28	4.83	3.53	2.87
Annual energy cost savings, $/$m^2$	**–**	**3.45**	**4.75**	**5.41**
Annual energy cost savings /m^2, %	**–**	**41.7**	**57.4**	**65.30**
Electrical use intensity, kWhrs/m^2	75.7	60.4	44.1	36.0
Gas use intensity, mJ / m^2	278.1	–	–	–
Estimated USA emissions, kg/kWhr (2)	0.85	0.85	0.85	0.85
Total Electricity CO_2 emissions, tonnes x 10^6	58.06	46.14	33.71	27.52
Add gas emissions @ 55kg/GJ, tonnes x 10^6	13.75	–	–	–
Total CO_2 emissions, tonnes x 10^6	71.81	46.14	33.71	27.52
Annual reduction in emission, tonnes x 10^6	**–**	**25.67**	**38.1**	**44.29**
Annual reduction in emission, %	**–**	**35.8**	**53.1**	**61.8**
Total heat rejected to atmosphere, GJ x 10^9 (1)	0.523	0.223	0.17	0.145
% water cooled plant or % of time water on	20 (3)	15 (4)	15 (4)	15 (4)
Heat rejected to water, GJ 10^9	0.105	0.034	0.026	0.022
Cooling Water use including 10% bleed, ltrs/GJ	460	460	460	460
AC plant cooling water use, Gl	48.3	15.6	12.0	10.1
Reduction in water consumption, Gl	**–**	**32.7**	**36.3**	**38.2**
Reduction in water consumption, %	**–**	**67.7**	**75.2**	**79.1**

(1) From Table 5 (3) % water cooled plant from Table 1 (4) % of time water on
(2) Assumed USA power generation: 15% nuclear, 15% gas and 70% black coal

Table 7 Summary of Benefits from Transcritical CO_2 Cooling and Heating of American Office Buildings

Parameter			Qty and % Reduction in System Type No					
			2		3		4	
No	Description	Unit	Qty	%	Qty	%	Qty	%
1	Primary energy	GJ x 10^9	0.4	39.9	0.58	55.8	0.66	63.5
2	Electrical energy	kWhrs x 10^9	13.7	20.2	28.4	41.8	35.7	52.4.
3	CO_2 gas emissions	Tonne x 10^6	2.57	35.8	38.1	53.1	44.3	61.8
4	Energy cost reduction	$/m^2	3.45	41.7	4.75	57.4	5.41	65.3
5	Cooling water	Gl	32.7	67.7	36.3	75.2	38.2	79.1
6	Energy use intensity	kWhrs/m^2	15.3	20.2	31.6	41.7	39.7	52.4

8. IMPLEMENTATION

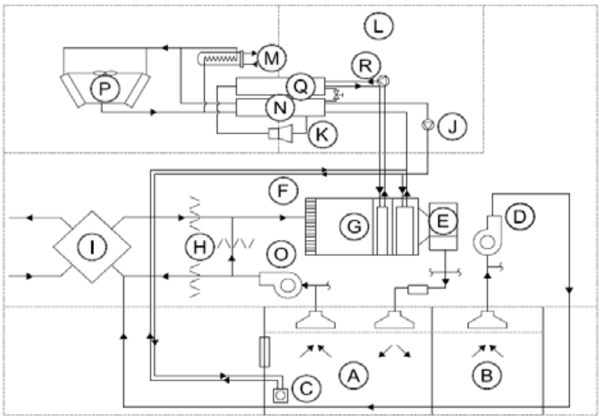
8.1 Applied to Existing Buildings (See Figure 7)

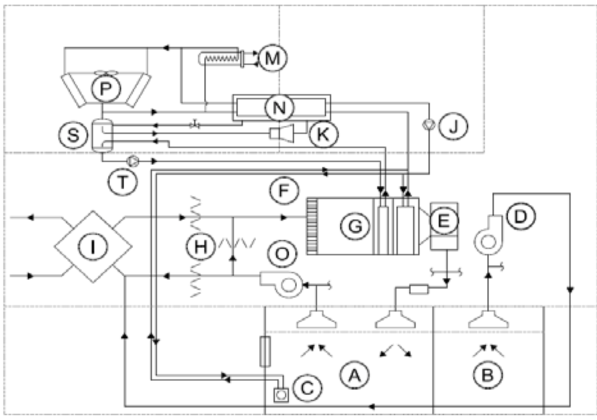
In an existing building the chiller set, cooling tower and boiler would be removed. These would be replaced by CO_2 compressors, water chillers, a water cooled gas cooler for heat recovery and air or water cooled gas coolers to remove the balance of the heat. Existing pumps would be used to circulate chilled and hot water through the building to existing air handling units. The supply and return fans would be slowed down to 75% of their current speed.

8.2 New Buildings (See Figure 8)

A total system would be designed to circulate CO_2 refrigerant or chilled water to air handling units throughout a building. Re-circulated CO_2 would be the most efficient and this would require custom design of the refrigeration systems similar to industrial refrigeration design practices.

All components are available in the market, but the required compressors would be expensive and on a long lead time. The entire system such as ducting and fans, hot water systems, etc would be designed using well established energy efficient design practices.





COMMON LEGEND

A Office Space
B Bathroom
C Tap Hot Water
D Central Exhaust Fan
E Supply Fan
F Air Delivery System
G Central Air Handling Unit
H Air Dampers
I Air to Air Heat Exchanger
J Hot Water Pump
K CO_2 Compressor
L Cold Water System
M River Water Gas Cooling
N 2 Stage Water Heater
O Return Air Fan
P Gas Cooler
Q Evaporator
R Chilled Water Supply & Return
S CO_2 Suction Separator
T CO_2 Pump

Figure 7 Schematic of a Central System with CO_2 Cooled Water Chiller, Exhaust Air Energy Recovery and Economising Cycle and Two Stage Water Heating

Figure 8 Schematic of a Central System with Liquid Recirculation CO_2 Evaporators, Exhaust Air Energy Recovery and Economising Cycle and Two Stage Water Heating

9. CONCLUSIONS

The retrofitting of CO_2 refrigerated cooling applications to the USA office building stock existing in 1999 would result in estimated reductions of 53%, 57%, 55% and 75% in CO_2 emissions, energy cost/m^2, primary energy consumption and water consumption respectively when applied to climates like that in Sydney, Australia.

When applied to new buildings with liquid recirculation CO_2 cooling, the estimated reductions would increase to about 61%, 65%, 63% and 79% in CO_2 emission, energy cost/m^2, primary energy consumption and water consumption respectively.

Where liquid recirculation CO_2 is frowned upon because of the high CO_2 pressures, CO_2 refrigerated chilled water systems would result in estimated reduction of 56%, 60%, 58% and 76% in CO_2 emissions, energy cost /m^2, primary energy consumption and water consumption respectively. This is based on a weighted COP of 5.36 and a 25% reduction in duct velocities.

Further AC cooling load reductions will result from reduced heat loads generated by more efficient lighting and more use of natural lighting . Thus electrical energy is saved again two ways, i.e. energy consumed by the lights and the compressor energy to remove the resulting heat.

Still greater savings would be possible if ambient air economizing cycles and exhaust air energy recovery systems would be implemented as well.

It is worth noting, that 79% of office floor space was cooled by low COP packaged air cooled air conditioning equipment (52%), air cooled screw and reciprocating compressor chillers (20%) and heat pumps (7%). This phenomenon is driven by low capital cost, but it comes at the price of high energy costs. In Australia this same phenomenon is also driven by an almost hysterical fear of legionella disease coupled with serious water supply problems.

However, reduced cooling water consumption at a building with high electrical energy consumption results in more cooling tower water use at the power station which is usually a greater quantity than that saved at the building.

Furthermore, the threat of legionella is entirely eliminated, as are the dangers of HFC fugitive gases with high GWP escaping into the environment.

Having regard to the outstanding environmental benefits of CO_2 refrigeration evaluated in this paper, it is abundantly clear, that the revival of CO_2 refrigeration for air conditioning cooling is a sustainable process far superior to current practices.

There is no other refrigerant, natural or chemical, which can deliver such superior and sustainable results in the cooling and heating of buildings of any type.

10. ACKNOWLEDGEMENT

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