COMFORT, SUSTAINABILITY AND ENERGY SAVINGS USING MOVING AIR

ABSTRACT

ASHRAE Standard 55 provides for the use of moving air instead of cold air for comfort. One of the criteria of Standard 55 for using moving air is personal control. With moving air room temperatures can be set much higher. This results in substantial energy savings because the cooling system does less work and extends the hours of economizer operation. Tests by Bauman, et al., at the University of California at Berkeley and Khedari in Thailand show that moving air can reduce the perceived temperature by as much as 15°F (9°C). This permits setting the room temperature at 78° to 80°F (26° to 27°C) instead of 72° to 74°F (22° to 23°C). The results are compressor energy savings of 20 to 25% (Fig. 3a-b). When incorporating an economizer cycle, very large additional savings of compressor running time are available. Of course, this varies with geographic location. In addition, personal control will reduce the need for outside air for more energy savings. Further, along with the very substantial energy savings, personal control will eliminate the number one complaint in virtually every office by giving each person the ability to adjust the temperature each one feels from 80°F down to 65°F (27° down to 18°C), cooling to his or her metabolic need at all times and with no time delay.

INTRODUCTION

This paper will show a very effective way to make a building more economical, ergonomic and sustainable by the use of moving air to provide comfort cooling.

ASHRAE Standard 55 provides for doing so, provided the air is under direct personal control of the occupant. It is known as Task/Ambient Conditioning, or TAC. The principal has been used effectively in automobiles and airplanes for many years. The following will show that not only will this save a substantial amount of energy, but it will also allow individuals to control their environment and select the perceived temperature to meet their varying needs. This will virtually eliminate the number one complaint in most offices. By combining the TAC system with Displacement Ventilation, additional energy can be saved and the indoor air quality (IAQ) maximized.

ENERGY SAVINGS

Tests performed at the Center for Environmental Design Research at the University of California at Berkeley by K. Tsuzuki, Fred Bauman, Edward Arens and D. Wyon[1] show that temperature perceived by occupants can be as much as 15°F (8°C) below room temperature. This allows room temperatures as high as 80°F (27°C).

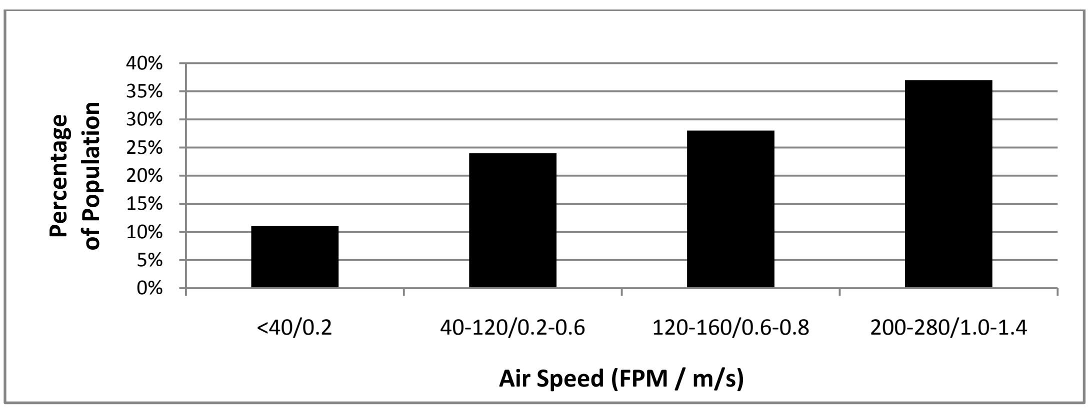


Figure 1. Chosen air velocities.

Depending on where the occupant sets the air velocity (normally people will set it between 40 and 200 FPM (0.2 – 1.0 m/s), see Fig. 1[2]), the perceived temperature can vary between 80°F (27°C), the room temperature, and 65°F (18°C), the temperature perceived because of the additional heat removed by the moving air.

This effect is confirmed by curves prepared by Richard Aynsley[3] based on work by Khedari[4] in Thailand (Fig. 2). It makes possible operation at a much higher room temperature than is customary today. By so doing the amount of work performed by the refrigeration plant is materially reduced. For example, an estimate of the savings on a 50,000 SF (4,645 SM) office building in Atlanta can be up to 61%, or as much as $9,000 per year (see Addendum B).

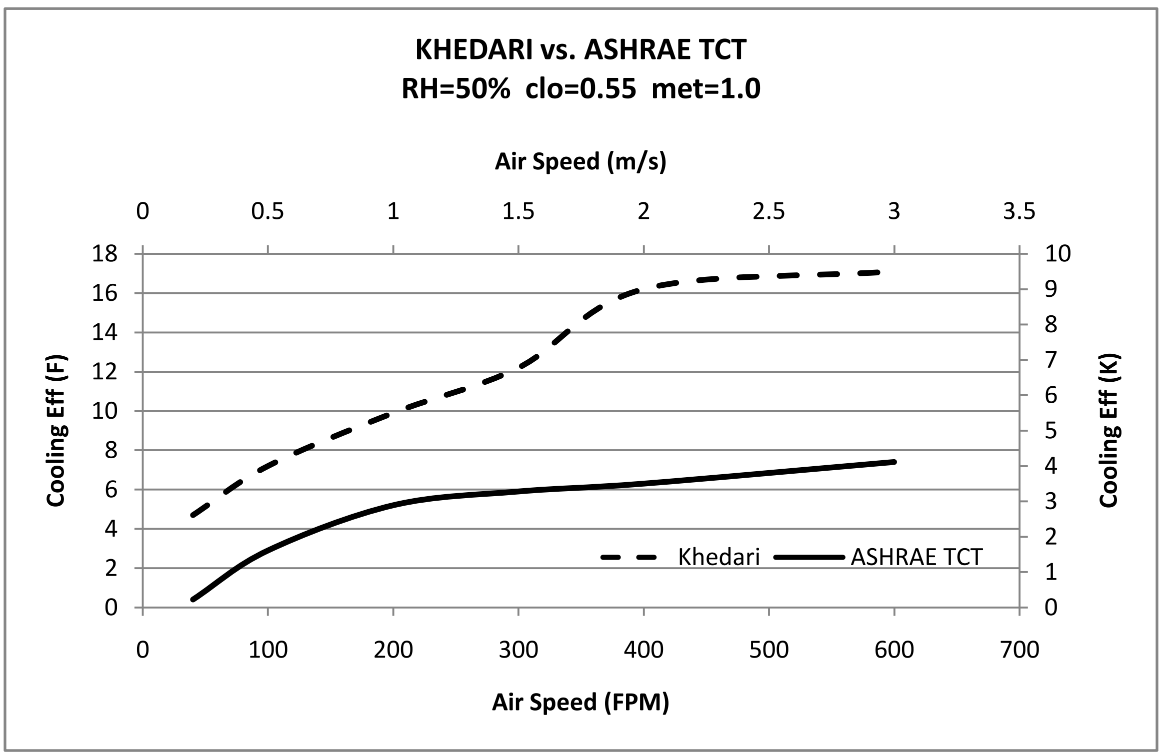


Figure 2. Cooling effect of elevated air speed.

Assuming a design temperature of 95°F (35°C), the required temperature reduction is decreased from the usual 21°F (12°C) to 15°F (8°C), or approximately 30% (Table 1).

Table 1. Energy Savings

	Stand HVAC	Task/Ambient	+/-
Design Condition	95°F (35°C)	95°F (35°C)	
Room Setpoint	74°F (23°C)	80°F (27°C)	
▲T Differential	21°F (12°C)	15°F (8°C)	6°F (4°C)

Since the compressor power requirements are roughly proportional to the amount of temperature reduction, this results in a savings of nearly 20-25% of the compressor energy. (If the compressor is not modulating, as it is in most package equipment, this shows up as a reduction in running time.)

Further energy savings are achieved by a reduction in the ventilation air required, and the reduced energy need to cool to the higher temperature.

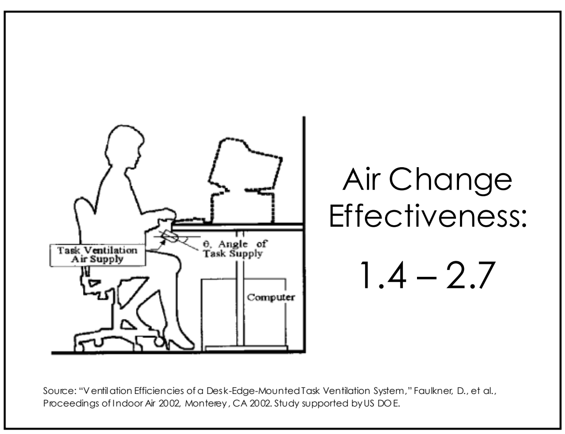


Figure 3. Ventilation effectiveness.

The ventilation effectiveness of task/ambient systems (see Fig. 3) is substantially increased, since air travels a very short distance before being inhaled, and therefore will not mix significantly with contaminated room air. The amount of energy saved depends on the enthalpy of the outside air, which varies a great deal from area to area.

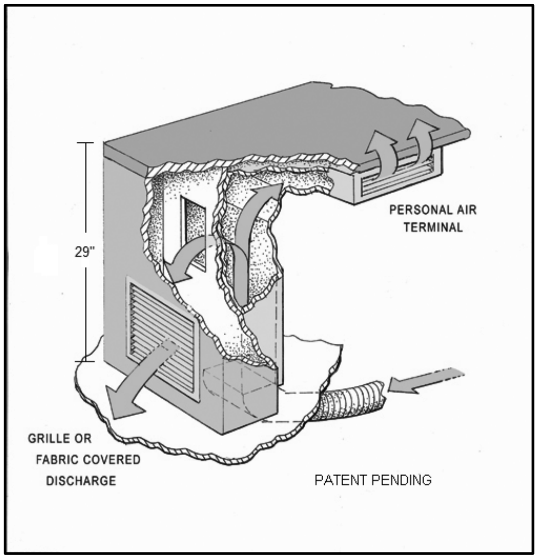
Last, but certainly not least, a great amount of energy can be saved with an economizer cycle much greater than with conventional systems. While conventional systems can operate an economizer cycle up to 55°F (13°C), the task/ambient system can use outside air up to 65°F (18°C), or even higher when enthalpy control is used. For example, in the Washington, DC, area this represents a decrease of 1397 hours of compressor operating time annually. Assuming cooling requirements for 4814 hours a year, this saves approximately 29% of the compressor hours (see Addendum A) in addition to the savings due to the higher operating temperature. This will also make the economizer cycle useful in many areas where it normally is not cost effective. We estimate total compressor energy savings to be 40%. This figure is an average for the Washington, DC, area.

Examples of increased economizer range based on DOE WYEC weather data (see Addendum A) and higher operating temperature with TAC are shown in Table 2.

Table 2. Additional Economizer Hours

	Hours of mechanical cooling with CAC (>55°F (13°C))	Additional hours of economizer with TAC (55°F (13°C) to 65°F (18°C))	Per cent reduction in mechanical cooling†
Atlanta, GA	5844	1952	33.4%
Washington, DC	4814	1397	29.0%
Chicago, IL	3922	1374	35.0%

† Based on continuous system operation



Using underfloor air distribution (UFAD) is probably the most cost effective and practical way to install a task/ambient system. This will further save energy by reducing the required fan horsepower because of the absence of most ductwork. A precise measure will depend on specific project fan power requirements.

Figure 4. Task/ambient air distribution.

Task/ambient air distribution by means of office furniture or similar distribution equipment creates space to distribute the additional air required to cool the surrounding area by means of displacement ventilation (Fig. 4). Tests conducted by others[5] show that this will result in the best interior air quality (IAQ) obtainable.

In addition to energy savings numerous tests indicate that providing personal control saves human energy in the form of higher productivity. While conventional energy savings can be readily calculated, the increase in productivity is difficult to ascertain.

Three questions have been raised regarding the system described above.

1) Does the higher temperature present a problem in controlling humidity?

> In a properly designed UFAD system the refrigeration circuit is the same as in a conventional system and operates at around 55°F (12°C). The

higher leaving air temperature is achieved by mixing return air with supply air. Using face and bypass dampers will also provide humidity control. (See the psychrometric chart, Fig. 5.)

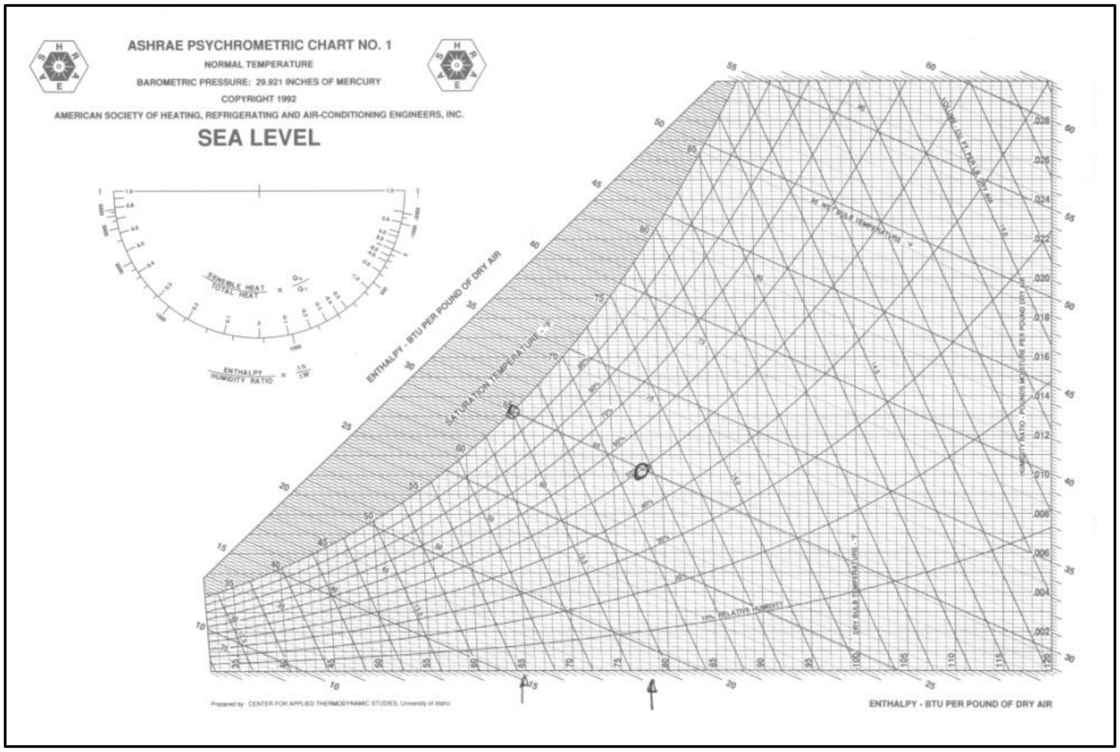


Figure 5. Psychrometric chart.

2) Does distributing air under the floor present a problem with contamination?

Underfloor air distribution was first used in computer rooms at a time when IBM insisted on careful testing of the air to make sure it would not affect their equipment. Provided the floors were cleaned before they were closed in, and with reasonable housekeeping, contamination never posed a problem. Furthermore, should contamination occur, it is easier to clean below an access floor than inside ductwork.

3) Do underfloor systems with pressurized floors show substantial leakage and resulting losses of energy?

> To prevent leakage from the plenum, the simplest solution is to draw the air out of the plenum with fans. This is already the practice in many installations at the perimeter for the purpose of better control. By carefully choosing efficient fans, any power consumed will be saved in the fans of the air handling equipment. This design was effectively used by Hank Spoormaker in many buildings in Johannesburg, South Africa.[6]

SUMMARY

Moving air has been used effectively for cooling in automobiles and aircraft for many years. Because of limited space, velocities tend to be somewhat high. This is easily overcome in the office environment where more space is available.

By judiciously using moving air as suggested in ASHRAE Standard 55-2004, a substantial savings in energy can be accomplished. In combination with task/ambient room air distribution, a reduction in compressor power consumption can exceed 50%. By being able to use higher room temperatures, savings are achieved in several ways.

First of all, less compressor power is needed because the compressor does less work. Figure 3a shows energy savings of 20 to 25%. Greater ventilation effectiveness and less cooling requirement save an additional 10 to 15%, for example, in the Atlanta area (see Addendum B). Last, and not least, a substantial increase in economizer cooling hours that can be used to save 33% of compressor running time.

Addendum B shows calculations for an assumed 50,000 SF (4,645 SM) office building in Atlanta estimating savings of approximately $9,000.

In addition to saving energy, task/ambient air distribution empowers people to control their environment, thus eliminating the number one complaint in office buildings.

REFERENCES

[1] Tsuzuki, K., E.A. Arens, F.S. Bauman, and D.P. Wyon. 1999. "Individual thermal comfort control with desk-mounted and floor-mounted task/ambient conditioning (TAC) systems." *Proceedings: Indoor Air 99.* Edinburgh, August 8-13.

[2] 1998. "A Study of Occupant Cooling by Personally Controlled Air Movement." *Energy and Buildings*, Vol. 27, No. 1, pp 45-59.

[3] Aynsley, R. 2006. "Cooling effect of elevated air speed." *Proceedings: ASHRAE Annual Meeting.* Quebec City, June 24-28.

[4] Khedari, J., N. Yamtraipat, N. Pratintong, and J. Hinrunlabbh. 2000. Thailand ventilation comfort chart. *Energy and Buildings*, Vol. 32, pp 245-249.

[5] Chen, C., and L. Glicksman. 2003. *System Performance Evaluation and Design Guidelines for Displacement Ventilation*. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

[6] Spoormaker, H.J. 1990. "Low-pressure underfloor HVAC system." *ASHRAE Transactions*, Vol. 96, Pt.2.

BIBLIOGRAPHY

Bauman, F.S., E.A. Arens, S. Tanabe, H. Zhang, and A. Baharlo. 1995. "Testing and optimizing the performance of a floor-based task conditioning system." *Energy and Buildings*, Vol. 22, No. 3, pp. 173-186.

Cornell University. 1999. "Case study: 901 Cherry – Gap Headquarters." http://dea.human.cornell.edu/Ecotecture/Case%20Studies/Gap/gap_home.htm. Ecotecture site, Department of Design and Environmental Analysis, Cornell University, Ithaca, NY.

Drake, P., P. Mill, V. Hartkopf, V. Loftness, F. Dubin, G. Zigara, and J. Posner. 1991. "Strategies for health promotion through user-based environmental control: a select international perspective." *IAQ 91, Healthy Buildings*. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Engineering Interface Limited. 1993. "Personal control and 100% outside-air ventilation for office buildings." Report prepared for Efficiency and Alternative Energy Technology Branch, CANMET, Canada.

Faulkner, D., W.J. Fisk, D.P. Sullivan, and S.M. Lee. 2002. "Ventilation efficiencies of a desk-edge-mounted task ventilation system." *Proceedings of Indoor Air 2002, Monterey, CA, 30 June – 5 July 2002.*

Houghton, D. 1995. "Turning air conditioning on its head: Underfloor air distribution offers flexibility, comfort, and efficiency." E Source Tech Update TU-95-8, E Source, Inc. Boulder, CO, August, 16 pp.

Kroner, W., J. Stark-Martin, and T. Willemain. 1992. "Using advanced office technology to increase productivity: The impact of environmentally responsive workstations (ERWs) on productivity and worker attitude." The Center for Architectural Research, Rennsalaer, Troy, NY.

Levy, H. 2002. "Individual control by individual VAV." *Proceedings, ROOMVENT 2002, Copenhagen, Denmark, 8-11 September 2002.*

Loftness, V., R. Brahme, M. Mondazzi, E. Vineyard, and M. MacDonald. 2002. "Energy savings potential of flexible and adaptive HVAC distribution systems for office buildings – Final Report." Air Conditioning and Refrigeration Technology Institute 21-CR Research Project 605-30030, June. Full report available at http://www.arti21cr.org/research/completed/index.html.

McQuillen, D. 2001. "3 case studies for improved IAQ." *Environmental Design + Construction*, posted 1/24/2001, http://www.edcmag.com.

Nielsen, P.V. 1996. *Displacement Ventilation – Theory and Design*. Department of Building Technology and Structural Engineering, Aalborg University, Aalborg, Denmark.

ADDENDUM A

Table 2

ANNUAL TEMPERATURE DISTRIBUTION

WASHINGTON, D.C. 38-51N 77-02W 14 FT. ELEV. ANNUAL

TEMP BIN	1 - 4	5 - 8	9 - 12	13 - 16	17 - 20	21 - 24	TOTAL		HORIZ. SOLAR (BTUH/SQ.FT.)
DRY-BULB TEMPERATURES :	HOURS OF OCCURRENCE (MCWB)								
90 / 94	0 (0)	0 (0)	11 (76)	60 (77)	15 (77)	0 (0)	86	(77)	190
85 / 89	0 (0)	0 (0)	54 (75)	142 (72)	84 (74)	0 (0)	280	(73)	166
80 / 84	2 (74)	11 (75)	140 (72)	172 (70)	155 (71)	51 (74)	531	(71)	155
75 / 79	89 (73)	88 (73)	166 (68)	168 (64)	169 (67)	145 (71)	825	(69)	128
70 / 74	186 (69)	188 (68)	164 (64)	135 (61)	173 (63)	198 (67)	1044	(66)	106
65 / 69	178 (64)	156 (63)	119 (59)	81 (56)	102 (59)	178 (62)	814	(61)	101
60 / 64	133 (58)	119 (58)	86 (54)	103 (51)	73 (53)	102 (57)	616	(56)	83
55 / 59	106 (53)	137 (53)	113 (50)	97 (47)	90 (48)	75 (51)	618	(51)	74
50 / 54	99 (48)	90 (48)	95 (46)	99 (44)	129 (45)	111 (47)	623	(46)	70
45 / 49	128 (43)	110 (43)	128 (41)	108 (40)	102 (41)	135 (42)	711	(41)	79
40 / 44	137 (38)	128 (38)	99 (36)	113 (36)	130 (36)	135 (38)	742	(37)	67
35 / 39	148 (33)	142 (34)	134 (32)	107 (31)	113 (32)	137 (33)	781	(33)	71
30 / 34	138 (29)	137 (29)	97 (28)	47 (29)	84 (28)	114 (28)	617	(28)	51
25 / 29	62 (25)	96 (25)	35 (24)	19 (24)	25 (24)	52 (24)	289	(24)	48
20 / 24	39 (20)	41 (20)	8 (18)	5 (19)	10 (19)	13 (19)	116	(20)	63
15 / 19	6 (15)	5 (14)	10 (13)	4 (15)	4 (15)	6 (16)	35	(15)	77
10 / 14	9 (10)	12 (11)	1 (12)	0 (0)	2 (11)	8 (10)	32	(10)	21

163
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ATLANTA, GEORGIA 33-39N 84-26W 1010 FT. ELEV. ANNUAL

TEMP BIN	1 - 4	5 - 8	9 - 12	13 - 16	17 - 20	21 - 24	TOTAL		HORIZ. SOLAR (BTUH/SQ.FT.)
DRY-BULB TEMPERATURES :	HOURS OF OCCURRENCE (MCWB)								
90 / 94	0 (0)	0 (0)	1 (77)	29 (75)	6 (75)	0 (0)	36	(75)	197
85 / 89	0 (0)	0 (0)	65 (75)	179 (73)	76 (73)	0 (0)	320	(74)	189
80 / 84	0 (0)	0 (0)	174 (73)	231 (71)	186 (72)	7 (72)	598	(72)	169
75 / 79	19 (72)	36 (72)	192 (70)	144 (67)	202 (69)	174 (72)	767	(70)	140
70 / 74	308 (70)	291 (70)	144 (66)	161 (61)	183 (64)	274 (69)	1361	(67)	103
65 / 69	203 (65)	180 (65)	145 (60)	195 (57)	148 (58)	142 (63)	1013	(61)	105
60 / 64	125 (59)	128 (61)	158 (56)	128 (54)	186 (55)	160 (57)	885	(57)	90
55 / 59	173 (54)	152 (55)	123 (52)	114 (49)	122 (50)	180 (52)	864	(52)	85
50 / 54	149 (48)	132 (49)	110 (47)	82 (44)	105 (46)	165 (47)	743	(47)	91
45 / 49	143 (44)	161 (44)	102 (43)	55 (40)	74 (41)	101 (43)	636	(43)	71
40 / 44	109 (39)	107 (40)	82 (38)	49 (36)	65 (36)	87 (38)	499	(38)	68
35 / 39	80 (33)	91 (34)	77 (33)	47 (31)	38 (31)	64 (33)	397	(33)	72
30 / 34	69 (29)	83 (29)	31 (28)	27 (28)	42 (28)	51 (28)	303	(29)	72
25 / 29	44 (24)	54 (24)	28 (23)	11 (22)	15 (23)	32 (24)	184	(24)	80
20 / 24	18 (19)	17 (20)	16 (20)	8 (20)	9 (19)	13 (19)	81	(20)	84
15 / 19	13 (14)	12 (14)	10 (15)	0 (0)	3 (15)	10 (14)	48	(15)	94
10 / 14	7 (12)	16 (12)	2 (11)	0 (0)	0 (0)	0 (0)	25	(12)	36

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CHICAGO,ILLINOIS 41-47N 87-45W 607 FT. ELEV. ANNUAL

DRY-BULB TEMPERATURES : HOURS OF OCCURRENCE (MCWB)

TEMP BIN	TIME GROUP 1 - 4	5 - 8	9 - 12	13 - 16	17 - 20	21 - 24	TOTAL		HORIZ. SOLAR (BTUH/SQ.FT.)
90 / 94	0 (0)	0 (0)	15 (74)	64 (73)	18 (71)	0 (0)	97	(73)	187
85 / 89	0 (0)	0 (0)	64 (71)	113 (68)	45 (70)	0 (0)	222	(69)	186
80 / 84	0 (0)	7 (71)	131 (67)	113 (65)	90 (67)	21 (70)	362	(67)	179
75 / 79	25 (70)	51 (68)	114 (64)	116 (64)	125 (65)	81 (68)	512	(65)	150
70 / 74	120 (66)	125 (65)	131 (62)	111 (62)	151 (63)	167 (65)	805	(64)	124
65 / 69	156 (62)	155 (62)	88 (60)	65 (58)	80 (59)	143 (60)	687	(61)	97
60 / 64	159 (57)	130 (57)	78 (54)	73 (53)	72 (55)	103 (57)	615	(56)	103
55 / 59	111 (52)	112 (52)	93 (49)	109 (49)	96 (51)	101 (52)	622	(51)	103
50 / 54	112 (48)	92 (48)	88 (46)	89 (47)	100 (47)	104 (47)	585	(47)	82
45 / 49	92 (44)	102 (43)	97 (42)	96 (40)	83 (42)	107 (43)	577	(42)	94
40 / 44	120 (38)	104 (39)	72 (37)	94 (37)	131 (37)	115 (38)	636	(38)	89
35 / 39	106 (34)	105 (34)	133 (33)	134 (33)	131 (33)	111 (34)	720	(33)	83
30 / 34	182 (30)	182 (30)	156 (29)	129 (29)	135 (30)	173 (30)	957	(30)	64
25 / 29	93 (25)	98 (25)	66 (25)	68 (24)	94 (25)	92 (25)	511	(25)	79
20 / 24	75 (21)	75 (21)	60 (20)	40 (20)	44 (21)	60 (21)	354	(21)	70
15 / 19	63 (16)	56 (16)	30 (16)	21 (15)	33 (15)	40 (16)	243	(16)	73
10 / 14	25 (11)	32 (11)	25 (10)	13 (10)	11 (11)	19 (11)	125	(11)	104
5 / 9	5 (7)	14 (6)	11 (5)	8 (6)	16 (6)	12 (5)	66	(6)	94
0 / 4	15 (1)	16 (0)	7 (0)	4 (1)	5 (0)	11 (1)	58	(0)	100
-5 /-1	1 (-4)	4 (-7)	1 (-6)	0 (0)	0 (0)	0 (0)	6	(-6)	39

137
615
622
1374

Source: DOE WYEC data from ASHRAE BINDATA program.

ADDENDUM B

This addendum describes examples of energy savings in three areas based on task/ambient conditioning (TAC): increased economizer time, reduced compressor load, and reduced ventilation load.

Assumptions are as follows:

Location	Atlanta, GA
Type of use	Open plan office
Occupied area	50,000 SF (4,645 SM)
Occupancy	350 Persons (7 Persons/1,000 SF (93 SM))
Total cooling load	400 SF/Ton (37 SM/Ton)
Compressor energy	0.6KW/Ton
Energy cost	$0.10/KWH
Equivalent full load hrs (EFLH)	2,000 Hrs

Total annual compressor operating cost	=	(Area x cooling load) x compressor energy x energy cost x EFLH
	=	(50,000/400) x 0.6 x 0.10 x 2,000
	=	$15,000 per year

<u>Economizer Savings</u>

Task/ambient conditioning takes advantage of economizer operation up to 65°F (18°C), versus 55°F (13° C) for conventional air conditioning (CAC). Total hours in Atlanta from 55°F (13°C) to 65°F (18°C) are 1,952 (See DOE WYEC data, Addendum A).

Per cent economizer savings	=	Economizer hours / Total CAC compressor hours per year
	=	1,952 / 5,844
	=	33.4%
Economizer savings per year	=	$15,000 x 33.4% = $5,010

Reduced Compressor Load

Raising the temperature in the occupied space reduces compressor energy consumption by 25% (see Figs. 2a-b).

Lower compressor load savings per year	=	(CAC compressor energy cost – economizer savings) x 25%
	=	($15,000 - $5,010) x 25%
	=	$2,498

Reduced Ventilation Load

Savings in cooling costs for outside air (OA) are based on higher ventilation effectiveness and higher operating temperature with TAC.

OA ventilation per person with CAC	20 CFM (10 l/s)(ASHRAE Standard 62)
Average OA cooling load	25 BTUH/CFM (3.8 W/l/s)
Ventilation effectiveness with TAC	2X

OA cooling load with CAC	=	OA ventilation per person x Cooling load per CFM x No. Persons
	=	20 x 25 x 350
	=	175,000 BTUH

@ 12,000 BTUH/Ton	=	14.6 Tons
@ 0.6 KW/Ton	=	8.8 KW

Cost savings from reduced ventilation load	=	OA cooling load with CAC x (1-1/Ventilation effectiveness with TAC) x Hours above 65°F (18°C) x Energy cost
	=	8.8 x (1-1/2) x 3,895 x 0.10
	=	$1,714

Total Energy Cost Savings	
Increased Economizer Time	$5,010
Reduced Compressor Load	2,498
Reduced Ventilation Load	1,714
	$9,222
Cost with CAC	$15,000
Energy Cost Savings %	61%