

Chapter 5

Comfort Energetics: Thermal Comfort Under Energy Constraints

This chapter represents an energy-conscious approach to understanding thermal comfort. With the environmental variable of T, MRT, RH and wind v as its structure, it begins with a psychrometric analysis where the roles of temperature and humidity are denominated in PMV votes and the power needed to improve those votes. The role of humidity is critically reviewed with a case made for challenging the accepted norms an optimum RH range. Finally the efficacy of moving air is analysed in the different ways it can be employed.

5.1 Case-Study: The Windup Girl

The PMV is the most popular standard for air-conditioned buildings, underpinning the ISO 7730 and ASHRAE 55 standards. It is used as a tool to assist mechanical engineers in specifying the correct plant sizes to meet the anticipated space cooling requirements. It is common that the engineering tools and rules of thumbs already have a safety margin, in addition to which engineers typically load another margin (Houghton 1998). Assumed in this approach is that there will not be any energy constraints placed in the life of the building and its equipment. It assumes that it is acceptable to over-engineer the mechanical ventilation system for that worst case scenario with the consolation that it can always be operated at partial load, even if that requires far more power than running a correctly sized plant at full load.

In his award winning science-fiction novel *The Windup Girl*, Bacigalupi (2009) paints the world post global-warming, where the oil economy has collapsed and the primary sources of energy are from methane composters and food calories. Electricity is so scarce it only exists to run government-owned computers, and personal computers are pedal operated like an old-fashioned sewing machine. Energy is produced by the muscles of people or beasts of burden, then stored kinetically in either flywheels or springs to power factories, transport and weapons. What is realistic and sobering about Bacigalupi's future is the challenge he makes to many assumptions we take for granted: unlimited resource, continued global

collaboration, genetic diversity. What if the only energy available was what we could muster from our muscles in a world with limited food?

Overhead, the towers of Bangkok's old Expansion loom, robed in vines and mold, windows long ago blown out, great bones picked clean. Without air conditioning or elevators to make them habitable, they stand and blister in the sun. (Bacigalupi 2009, p. 10).

The windup girl, the protagonist of the story, is a genetically modified woman with porcelain-perfect skin and greatly reduced pores, once a secretary for the rich who could afford air-conditioning, she has now been abandoned in hot-humid Bangkok (Fig. 5.1).

Even with her augmented vision she barely spies the pores of her flesh. So small. So delicate. So optimal. But made for Nippon and a rich man's climate control, not for here. Here, she is too hot and sweats too little. She wonders if she were a different kind of animal, some mindless furry cheshire, say, if she would feel cooler. Not because her pores would be larger and more efficient and her skin not so painfully impermeable, but simply because she wouldn't have to think. She wouldn't have to know that she had been trapped in this suffocating perfect skin by some irritating scientist with his test tubes and DNA confetti mixes who made her flesh so so smooth, and her insides too too hot. (Bacigalupi 2009, p. 51)

5.2 PMV, Psychrometrically Analysed

If presented with the severe energy constraints of Bacigalupi's Bangkok, how would one reapply the principles underpinning the PMV model so that our windup girl does not overheat? For argument's sake, assume she votes on the comfort scale the same way normal people do. The aim here is to provide improvement to comfort with the smallest energy input. Starting with a reference condition (Table 5.1), one can determine the impact on PMV with each variable changed, and the corresponding energy needed to drive the change.

The windup girl, for this exercise, will be housed in a small and heavily insulated cubicle ($1 \times 1 \times 2.4$ m, see Fig. 5.2). Fresh air is supplied at low level by displacement ventilation at a rate of 10 l/s in a small space (c.f. Fanger 2000), the air change will be high at 15 ach (air-changes per hour). Under steady-state conditions, air is exhausted at the same rate as it is being supplied, and there is no recirculation. It is assumed that the supply air temperature will be the same as room air temperature and MRT.

The following calculations determine how much energy is required for each process, and the extent to which the PMV is improved. As the heat loads (external and internal) are independent of process, they have been omitted to simplify the calculations.

Four methods are plotted psychrometrically with about 7 steps in each process, in which the PMV is compared to the energy required for each step shown below and tabulated in Table 5.2.



Fig. 5.1 Book cover of *The Windup Girl* (Bacigalupi 2009) illustrated by Raphael Lacoste. Image credit: Night Shade Books © 2009, reproduced with permission

Table 5.1 Reference conditions for study in comfort energetics

Reference occupant		Notes
Ventilation rate	10 l/s/pax	Typical rate in practice (Melikov 2002)
T, outside	30 °C	
RH, outside	70 %	
Air speed	0.05 m/s	
Clothing	0.7 clo	Between 0.5 and 1.0 clo used to generate comfort envelope in ASHRAE 55 (2010) pp. 6, 24
Metabolic rate	1.1 met	As used to generate comfort envelope in ASHRAE 55 (2010) pp. 6, 24
PMV	+2.0	
Air-conditioning	C.O.P. = 3	One watt of electricity transporting 3 w of heating/cooling

Omitted items: envelope thermal load, internal heat load, fan power

1. Cooling without dehumidification: air is cooled at constant moisture ratio up to dew point (30 to 24 °C, 1 °C/step).
2. Cooling with dehumidification: air is cooled past dew point (70 to 40 %RH, 5 %RH/step).

Fig. 5.2 Climate booth to determine energy required to improve PMV by various cooling processes

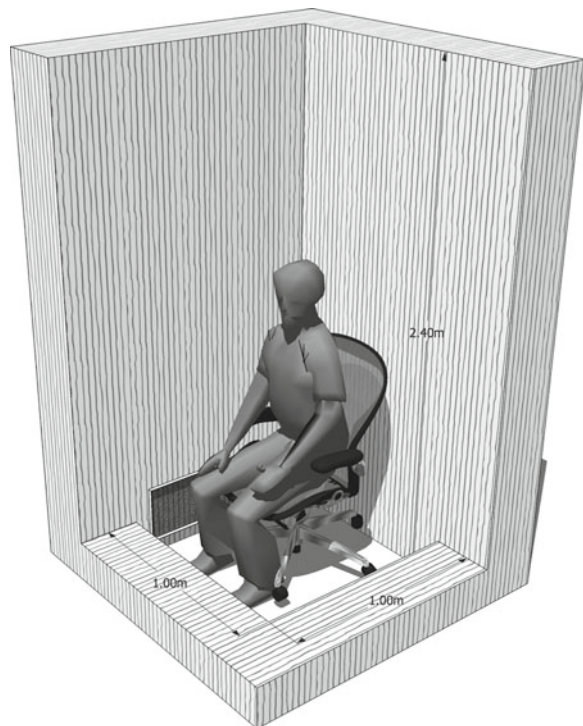


Table 5.2 Four methods analysed psychrometrically

Cooling, no dehumidification										
T (°C)	RH (%)	PMV	H (kJ/kg)	dH (kJ/kg)	Spec vol (m ³ /kg)	Power (W)	Thermal W/vote	Electrical W/vote		
30	70	1.994	78.37							
29	74	1.728	77.33	1.04	0.882	11.8	44.3	14.8		
28	79	1.454	76.29	2.08	0.879	23.7	43.8	14.6		
27	83	1.181	75.24	3.13	0.8761	35.7	43.9	14.6		
26	88	0.907	74.2	4.17	0.8732	47.8	43.9	14.6		
25	94	0.632	73.16	5.21	0.8703	59.9	44.0	14.7		
24	100	0.356	72.12	6.25	0.8674	72.1	44.0	14.7		
Cooling + dehumidification										
RH equiv. (if at 30 °C) (%)	Dew Pt	PMV @dp	H- dp	dH (kJ/kg)	Spec vol (m ³ /kg)	Power (W)	Thermal W/vote	Electrical W/vote		
70	23.93	0.327	72.04							
65	22.7	-0.056	67.22	11.15	0.862	136	66.5	22.2		
60	21.39	-0.467	62.36	16.01	0.856	194	78.8	26.3		
55	19.98	-0.908	57.41	20.96	0.850	254	87.4	29.1		
50	18.45	-1.376	52.36	26.01	0.844	315	93.5	31.2		
45	16.78	-1.903	47.19	31.18	0.837	379	97.4	32.5		
40	14.94	-2.487	41.86	36.51	0.830	447	99.7	33.2		
Direct evaporative cooling										
RH (%)	T (°C)	PMV	HR (g/kg)	Vapour density (g/m ³)	Humidification (g/sec)	Power (W)	Electrical W/vote			
70	30	1.994	18.89	21.33						
75	29.17	1.796	19.23	21.76	0.0043	0.030	0.154			
80	28.33	1.561	19.57	22.2	0.0087	0.062	0.142			
85	27.56	1.385	19.89	22.6	0.0127	0.090	0.148			
90	26.83	1.175	20.18	22.98	0.0165	0.117	0.143			

(continued)

Table 5.2. (continued)

Direct evaporative cooling									
RH (%)	T (°C)	PMV	HR (g/kg)	Vapour density (g/m ³)	Humidification (g/sec)	Power (W)	Electrical W/vote		
95	26.11	0.993	20.48	23.37	0.0204	0.145	0.145		
100	25.5	0.84	20.73	23.69	0.0236	0.167	0.167		
Desiccant dehumidification									
RH (%)	PMV	H (kJ/kg)	HR (g/kg)	dH (kJ/kg)	Spec vol (m ³ /kg)	Power (W)	Thermal W/vote	Electrical W/vote	
70	1.994	78.37	18.88		0.8849				
65	1.943	74.83	17.49	3.54	0.883	40.1	786.1	262.0	
60	1.892	71.3	16.11	7.07	0.8811	80.2	786.7	262.2	
55	1.841	67.79	14.74	10.58	0.8792	120.3	786.5	262.2	
50	1.79	64.29	13.37	14.08	0.8773	160.5	786.7	262.2	
45	1.738	60.8	12.01	17.57	0.8754	200.7	784.0	261.3	
40	1.687	57.33	10.65	21.04	0.8735	240.9	784.6	261.5	
35	1.636	53.88	9.3	24.49	0.8717	280.9	784.8	261.6	
30	1.585	50.44	7.95	27.93	0.8696	321.2	785.3	261.8	

Software PMV determined by PMV_Calc_V2 Modified by Håkan Nilsson, Department of Technology and Built Environment, Laboratory of Ventilation and Air Quality, University of Gävle. Psychrometry based on PsyCalc8 ver 3.1, 2008

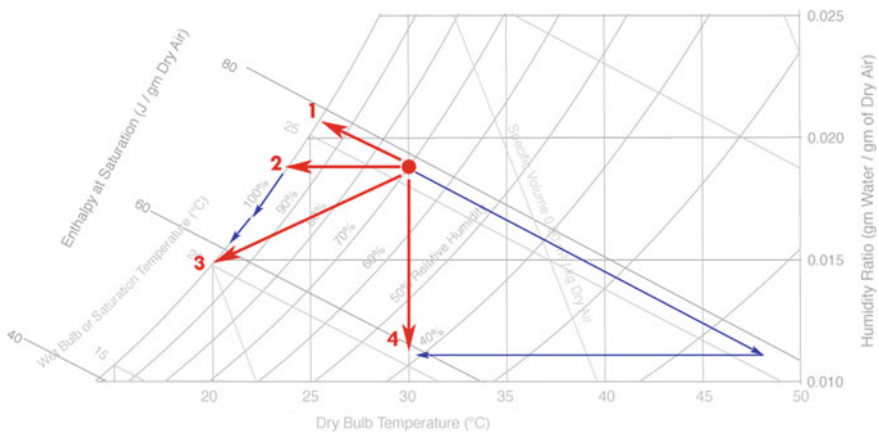
3. Direct evaporative cooling: air is humidified at constant enthalpy (70 to 100 %RH, 5 %RH/step). This process is actually passive so the only energy input has been the embodied energy of water (for sourcing, conveyance, treatment, distribution, end use and water treatment, Cohen et al. 2004).
4. Desiccant dehumidification: air is dehumidified by desiccant (constant enthalpy) and allowed to cool to ambient (constant T) (70 to 30 %RH, 5 %RH/step).

When the energy input for each step change is tallied with comfort outcome, it was observed that there is a good linear fit (with the exception of ‘cooling with dehumidification’) where each step change of equal magnitude in T or RH were matched by both ‘comfort’ and ‘electrical W/vote’.

5.3 Comfort Energetics

In recognising that there can be multiple ways of achieving a better comfort vote, how do we determine which method has the most effective use of the energy input? A new unit for comfort energetics, watts per vote, is proposed to express this rating. By comparing different cooling techniques and expressing the energy needed to improve PMV by one vote we have an index to rate the efficacy of each approach to thermal comfort.

The comfort energetics, in watts/vote are summarised and the cooling technique ranked in terms of PMV efficacy (Table 5.3).



With the exception of “Refrigeration Past Dew Point” the comfort energetics described in watts/vote show good linearity and minimal deviation (s.d. < 3 %), indicative that the comfort energetics are constant and independent of the starting PMV. In other words, an improvement in comfort from +3 to +2 will require the same amount of energy as +2.5 to +1.5, +1 to 0 and so forth.

Table 5.3 Cooling methods analysed in terms of energetics

Rank	Comfort approach	Process	Comfort energetics (W/vote)	S.D. (%)
1	Direct evaporative cooling	Adiabatic humidification	0.15	2.9
2	Refrigeration up to dew point	Constant humidity ratio (isodrosothermic)	15	0.4
3	Refrigeration past dew point	Cooling with dehumidification	28	15.7
4	Desiccant dehumidification	Adiabatic + isothermic dehumidification	262	0.1

Whilst the linear fit was unexpected it is convenient that the ordinal scale of the comfort vote could relate so well with the numerical scale of the psychrometric chart.

5.3.1 Comfort in High Humidity

We can also see that a change in temperature has a far more profound impact on comfort than a change in humidity. This is not new, the ISO 7730 standard mentions that “typically a 10 % higher relative humidity is felt to be as warm as a 0.3 °C rise in the operative temperature.” There is however little meaning to compare %RH to T°C unless it can be denominated in terms of the energy embodied to drive the change. Based on the psychrometric analysis of a PMV response, reducing air temperature will drive a response far more significant than the reduction in humidity. So in terms of comfort energetics we can say that reducing temperature (15 W/vote) is 17 times more effective than reducing relative humidity (262 W/vote), and that an evaporative cooler is 100 times more effective in achieving comfort than an air-conditioner, even if the air-conditioner does not perform any dehumidifying function.

This marginal impact of humidity is consistent with de Dear’s findings that in climate chambers held at 35 and 70 %RH respectively there were no significant differences in preferred temperature (de Dear et al. 1991).

If energy was the only constraint, and thermal comfort the only consideration, then misting fans with their saturated supply air offer unmatched comfort energetics. These are used in outdoor and semi-outdoor settings with the common notion that they are unsuitable for indoor use because they increase the moisture ratio of indoor air. From the human thermal comfort point of view, this logic is flawed. If the indoors and outdoors were at the same temperature and the misting fans were merely saturating the air surrounding an occupant, then there is no psychrometric difference. If all environmental (T°C, RH%, MRT°C and v m/s) and personal variables (clo and met) were made equal to both indoor and outdoor conditions, the PMV will be identical. Likewise, the adaptive model will show that if the daily, or mean monthly T, were the same in both the in and outdoor, the ideal T will be the same. So strictly speaking, from the thermal comfort perspective,

there should be no objection that would prejudice against the indoor operation of misting fans, an example being provided by gadget blogger Christen Costa (2012).

Low humidity is associated with higher evaporation rates of the skin. The operative parameter should be that of absolute humidity rather than relative humidity. In the earlier calculation, air at 30 °C and 70 %RH had a dew point of 24 °C. It should not be assumed that skin cannot dry in air at 100 %RH as the saturated vapour pressure of moist skin of 5.3 kPa (at 34 °C) is still significantly higher than that of the air of 3.0 kPa (at 24 °C). The vapour pressure difference will drive moisture from perspiration to evaporate and re-condense in the saturated air as steam, a phenomena similar to breathing out mist in cold weather.

In an excellent review on the effect of high humidity levels on comfort (Fountain et al. 1999) research has consistently showed that the impact of humidity on comfort is minor (Table 5.4).

Dehumidification improves comfort very marginally and comes at considerable energetic cost. Dehumidification by cooling air past its dew point involved double the comfort energetics as compared to cooling without passing the dew point. In typical air-conditioning applications the dehumidified air is chilled so far below

Table 5.4 Literature on comfort in high RH (Fountain et al. 1999)

Years	First author	T range tested	RH range tested	Effects noted
1923	Houghten	26.7–69.4 °C	22.8–45 °C wet bulb	In comfort zone, comfort depends equally on wet- and dry-bulb temperature
1950	Glickman	Unspecified	20–80 %RH	ET* overemphasizes RH for long exposures
1960	Koch	20–34 °C	20–90 %RH	Small over entire range
1966	Nevins	18.9–27.8 °C	15–85 %RH	Increase of 0.3 °C temperature = decrease of 10 %RH for sensation
1967	McNall	15.6–40.6 °C	20–90 %RH	No effect at low met, some effect at higher met
1973	Anderson	23 °C	10–70 %RH	None
1987	Tanabe	27.8–31.3 °C	50–80 %RH	PMV predicts TS too high
1989	Berglund	21–27.2 °C	20–95 %RH	Temperature order of magnitude more important for determining comfort
1989	Hayakawa	30–35 °C	30–90 %RH	No effect on thermal sensation up to 70 %—discomfort increased significantly with elevated metabolism
1994	Tanabe	25–30 °C	30–70 %RH	SET* good relation with TS
1994	Tanabe	26–28 °C	35–75 %RH	TS vote at 1.2 met similar for all RH
1995	Berglund	Reanalysis of 1989 data		Increased discomfort above 18 °C WB
1996	Bauman	26.1–26.6 °C ET*/ 26.2–28.1 °C SET*	50–80 %RH	No change in TS, no difference in acceptability except under increased met
1999	Fountain	20–25 °C (ET*)	60–90 %RH (9-15 g/kg)	Acceptable humidity limit can be quite high when subjects are sedentary

comfort it has to be reheated. Though this can be done with passive devices (e.g. Dinoh heat pipes) it is common for the reheat cycle to be performed by electrical heaters, bringing the comfort energetics to 3–4 times that of cooling without dehumidification.

High humidity is not contrary to thermal comfort. The ISO 7730 (2005) standard repeatedly states that whilst humidity affects comfort, its impact is minor.

The influence of humidity on thermal sensation is small at moderate temperatures close to comfort and may usually be disregarded when determining the PMV value. (p. 4)

If humidity limits are based on the maintenance of acceptable thermal conditions based solely on comfort considerations—including thermal sensation, skin wetness, skin dryness, and eye irritation—a wide range of humidity is acceptable. (p. 44)

Why then all this trouble to dehumidify?

5.3.2 *Understanding Humidity*

It is a common perception that humidity affects comfort and IAQ. In ASHRAE's "Top Ten Things Consumers Should Know About Air Conditioning" (<http://www.ashrae.org/education/page/1455#10>) it is mentioned:

Humidity control was the problem that originally spurred the need for air conditioning. Lack of humidity control in hot, humid climates, in particular, can lead to mold growth and other moisture-related problems. High indoor humidities can lead to health and comfort problems. Modern air conditioners dehumidify as they cool; you can see that by the water that drains away, but this dehumidification is incidental to their main job of controlling temperature.

Humidity, to the technical minded, can often be used too loosely.

Humidity ratio (HR) expresses the masses of moisture to dry air, as a ratio in g/kg. Relative humidity (RH) expresses the vapour pressure in air as a percentage of the maximum it can potentially be at the same temperature. Until dew point is reached, HR is unaffected by temperature, whereas RH is always temperature-dependent.

Another common misconception is that when air is saturated it cannot contain more moisture. This is incorrect. At high altitudes and in the absence of condensing nuclei, air can be supersaturated to 'several hundred per cent' (Battan 2003, p. 8). Where condensing nuclei are present it will be correct to say that air cannot contain more moisture in the vapour state, yet still more moisture can still be accommodated as suspended droplets. For instance if one were to breathe outdoors on a cold morning when the air is saturated, one can observe a breath mist. In a similar condition if one were to go for a run, the profuse perspiration will evaporate, condense into suspended droplets, and appear as steam emerging from the hot skin. It is thus incorrect to think of saturated air being incapable of accommodating more moisture. The determining function is that as long as the

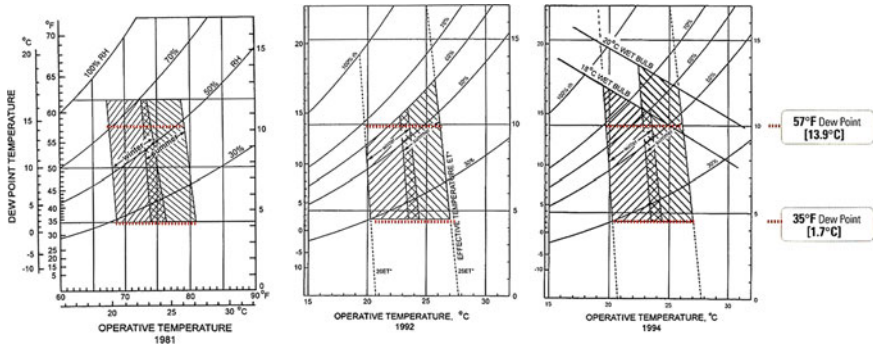


Fig. 5.3 Uncertainty in the RH stipulation in the ASHRAE 55 standard in years 1981, 1992 and 1994 (Harriman et al. 2008)

saturated vapour pressure of skin was higher than the vapour pressure of air, perspiration can still evaporate.

Currently the most complete treatise on this topic will be ASHRAE 90421 *Humidity Control Design Guide for Commercial and Institutional Buildings* (Harriman et al. 2008) which begins with a preface ‘It appears to be the first time in the 101 year history of the Society that ASHRAE has focused a book on this topic.’ The guide admits the uncertainty in defining a measure of RH and shows the variation over three editions of the same standard (Fig. 5.3). However there is a range of HR (corresponding to dew point temperature) that satisfies the three editions. The guide states, ‘What does seem consistent is the sedentary people are likely to find comfort in all seasons when the dew points between 1.7 and 13.9 °C’ (p. 73).

As this issue is of fundamental interest to the development of comfort energetics, we shall investigate it at length.

5.3.3 Optimum RH and ASHRAE 55 Misrepresented

The ASHRAE 55 (2004) revision maintained the upper 12 g/kg (17 °C dew point) limit and removed the lower limit. The 2010 revision maintains the 2004 limits for the graphical (i.e. psychrometric) approach but also allows the software (PMV) approach, for which the RH has no stated limit. With the PMV approach, as long as the range is within ± 0.5 (corresponding to PPD < 10 %), the conditions are deemed to be within the comfort zone (ASHRAE 2010, p. 7).

When optimum RH values are recommended, the ASHRAE 55 and other standards have been serially misquoted by major references and texts (see Table 5.5). An example is:

The earlier accepted relative humidity range, 20–80 %, was based primarily on comfort concerns; the narrower range today, 30–60 %, incorporates health considerations (Burroughs and Hansen 2011, p. 148).

Table 5.5 Recommended RH in common reference and textbooks

1st author, ref	Year	Recom. RH	Rationale, sources	Comments
Burroughs and Hansen	2011	30–60 %RH	Health	ASHRAE, Proetz, Andrews, Ritzel, Sale, Gelperin and Green Only ASHRAE and Green appear in bibliography. Still cites ASHRAE 55 (1992) though humidity standards have been superseded in 2004, and again in 2011
Nufert	2002	50–60 %RH 40–70 %RH	Comfort Health	–
Adler	1999	40–70 %RH	Comfort	–
Avallone	2007	Min. 0.0045 HR, max. 60 %RH	Comfort	ASHRAE handbook (1993)
Kreith	1999	<30 % or >60 %RH	Energy economy	ASHRAE 90.1 (1989)
Stein	2006	None, inconclusive	Comfort	ASHRAE 55 (1995, 2004), ASHRAE handbook (1997, 2001), Egan (1975), Givoni (1998), Heschong (1979), Humphreys and Nicol (1998), Kwok (1998), Lechner (2001), Milne (1979), Olgay (1963)
Bas	2003	30–60 %RH 50 %RH ideal	Comfort	ASHRAE 55 (1981)

Standard for economy to not dehumidify below 60 %RH or humidify above 30 %RH
 Recognises that ASHRAE 55 only stipulates control of HR, not %RH.
 Also recognises adaptive model has no RH recommendations

Once the erroneous referencing enters legislative controls, there are more serious implications. Under Australian Occupational Health and Safety (OH&S) guidelines (Jul 2011), the recommended range is 30–60 %RH. This range was adopted from the Canadian CSA Standard CAN/CSA Z412-00 (R2005)—“Office Ergonomics” which in turn has been sourced, it is claimed, from ASHRAE 55 (2010).

However, a review of the most current standards (Table 5.6) within my access indicate no stipulation for the control of RH.

Could it be possible that the ASHRAE 55 standard referred to have been those other than the 1981 and 2004 editions I have acquired? A summary of the standards preceding ASHRAE 55 (2004) has been documented (Fountain et al. 1999), from where we see that this humidity ratio of 0.012 has been specified since 1974 and is here updated with the 2010 edition (Table 5.7).

In my investigation, the only official ASHRAE document recommending an optimum RH range is to be found in the ASHRAE Handbook, *HVAC Systems and Equipment* (2008), with a lone reference to Sterling et al. (1985). The optimum RH chart shows a graphical indication of air quality concerns being directly correlated to RH and a sweet-spot of 40–60 %RH which can minimise all negative effects (Fig. 5.4).

Table 5.6 Humidity criteria from American and Australian standards

Standard	Years	Title	Humidity criteria
ASHRAE 55	2004	Thermal environmental conditions for human occupancy	w(i) under 0.012
ASHRAE 55	1981	Thermal environmental conditions for human occupancy	w(i) between 0.0045–0.012
ASHRAE 62	2010	Ventilation for acceptable indoor air quality	RH 65 % or less
ASHRAE 90.1	2004	Energy standard for buildings except low-rise residential buildings	Not stipulated
ASHRAE 160	2009	Criteria for moisture-control design analysis in buildings	w(i) = 0.004 + 0.4 w(o)
ASHRAE 189	2009	Standard for the design of high-performance green buildings	RH under 60 % during post construction, pre-occupancy flush out
AIRAH DA20	2000	Humid tropical air conditioning	55–60 %RH, based on upper limit of Sterling et al. (1985)
AIRAH DA09	1998	Air conditioning load estimation	Not stipulated
AS 1668.2	1991	The use of mechanical ventilation and air-conditioning in buildings—mechanical ventilation for acceptable indoor-air quality	“Out of scope”
AS 4254	2002	Ductwork for air-handling systems in buildings	Not stipulated

w(i) indoor design humidity ratio, w(o) mean coincident design outdoor humidity ratio for cooling, 1 % annual basis

Table 5.7 Humidity control in ASHRAE 55 and preceding standards

Years	Issuer	Maximum humidity	Document
1915	ASHVE	50 % relative humidity (recommended)	Code of Minimum Requirements for Ventilation
1920	ASHVE	64 °F wet bulb (17.8 °C)	Synthetic Air Chart
1932	ASHVE	70 % relative humidity	ASHVE Ventilation Standard
1938	ASHVE	75 % relative humidity	Code of Minimum Requirements for Comfort
1950–1965	ASHRAE	No explicit limit	ASHRAE Comfort Chart
1966	ASHRAE	60 % relative humidity	Standard 55-66
1974	ASHRAE	14 mm Hg (12 g/kg humidity ratio)	Standard 55-74
1981	ASHRAE	12 g/kg humidity ratio	Standard 55-81
1992	ASHRAE	60 % relative humidity	Standard 55-92
1992a	ASHRAE	18 °C (64.5 °F) wet bulb, winter 20 °C (68 °F) wet bulb, summer	Addendum to Standard 55-92
2004	ASHRAE	12 g/kg humidity ratio	Standard 55-2004
2010	ASHRAE	12 g/kg humidity ratio, or software	Standard 55-2010

Over the years the chart has remained untouched in its basic form and is widely cited by manufacturers of humidity control equipment. This chart still underpins

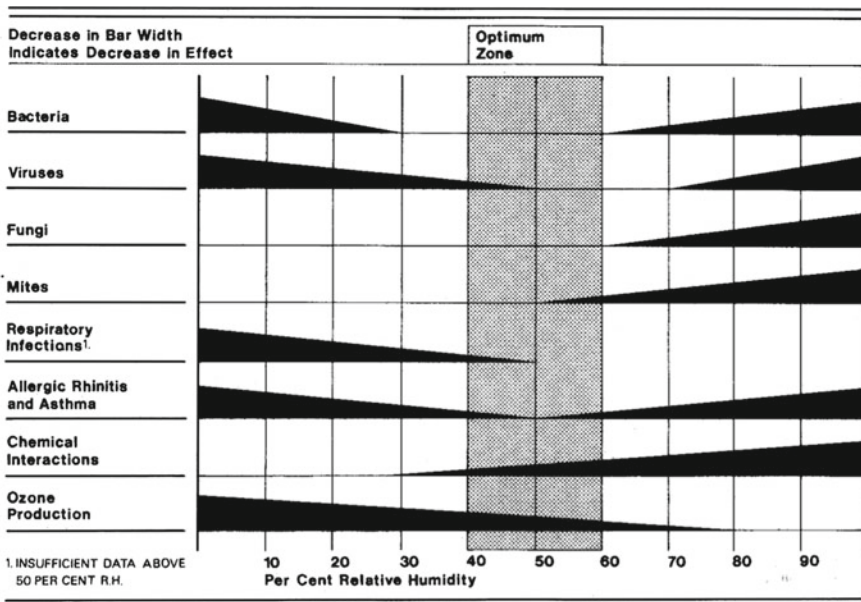


Fig. 5.4 The widely reproduced premise for recommending the optimum RH range, graphically presented (Sterling et al. 1985)

the section on “Humidifiers” in the current ASHRAE Handbook—HVAC Systems and Equipment (2009) ch 21.

Extremes of humidity are the most detrimental to human comfort, productivity, and health. Figure 1 referring to Sterling et al. (1985) shows that the range between 30 and 60 %RH (at normal room temperatures) provides the best conditions for human occupancy (Sterling et al. 1985). In this range, both the growth of bacteria and biological organisms and the speed at which chemical interactions occur are minimized.

In the most recent and extensive study commissioned by ASHRAE into humidity control, Brundrett [Chapter 9 in “Humidity Control Design Guide” (Harriman et al. 2008)] reviewed the behaviour of three viruses (*influenza A*, *Vaccinia*, *Poliomyelitis*) and ten bacteria (*Escherichia coli*, *Legionella pneumophila*, *pneumococcus*, *Serratia marcescens*, *Staphylococcus aureus*, *streptococci*, *pseudomonas vulgaris*, *proteus morgani*, *Salmonella derby*, *Ps aeruginosa*), based on old papers, stating that such research was “rare and valuable”. This literature reviewed averaged 1966 ± 12 y (s.d.), of which 5 of the 20 references could be found in Sterling et al. (1985). Brundrett also recommends the same 30–60 %RH range even though 2 viruses and 5 bacteria in the review could be most limited by high RH.

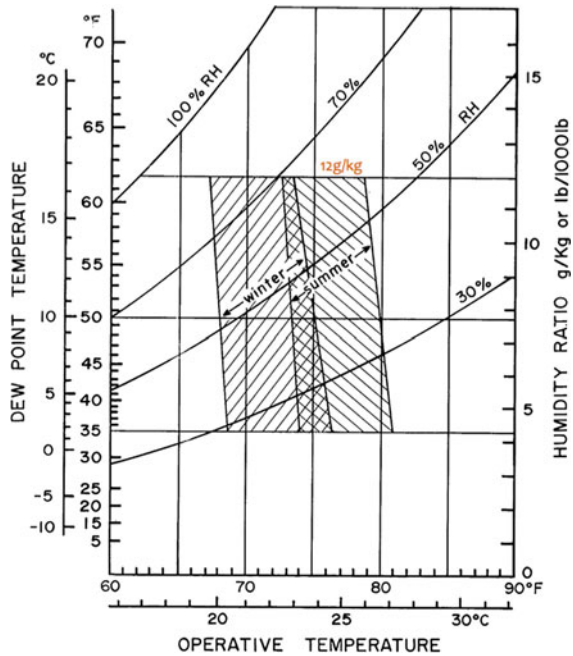
After a half-century one wonders if new microbes have become more commonplace, or if these old strains of microbes have not long mutated to behave differently since they were tested. The dated research is also indicative that there is no modern evidence (in the last 20 years, based on the most recent of Brundrett’s references) for RH as control mechanism for bacteria and viruses. It appears that the whole premise for the optimum RH range has not changed from trying to limit “the life span of the broadest variety of bacteria and viruses” (Brundrett 2008), an approach first proposed by Sterling.

This paper (Sterling et al. 1985) is here investigated in greater detail, not as a young researcher trying to be captious, but because it stands as such a significant document, and the only one offering a rationale behind the optimum RH range.

5.3.4 A Critique of Sterling

Sterling’s et al. (1985) paper “Criteria for Human Exposure to Humidity in Occupied Buildings” was a starting point to recommend a narrower range of 40–60 %RH. The then current ASHRAE Standard 55 (1981) did not actually stipulate an acceptable range of 20–90 %RH as Sterling suggested, but a 4.25–12 g/kg ratio, which within the limits of acceptable operative temperature could potentially vary from 20 to 90 %RH (Fig. 5.5). Graphically, the acceptable range is bound by two (horizontal) limits in terms of *humidity ratio* lines of 4.25 and 12 g/kg, and so it is erroneous to interpret the standard as thinking the acceptable range is bound by (the curved) *relative humidity* lines of 20 and 90 %. The failure to distinguish the humidity units has, in my opinion, cascaded in a series of potentially flawed analyses.

Fig. 5.5 Acceptable T and humidity range (ASHRAE 55 1981)



Sterling’s paper is not based on original experiments but a “review of the relevant health literature”, a total of 75 separate studies, manuals and literature reviews with a mean date 1975 ± 7.8 years (s.d.).

The paper looks at the premise for recommending 40–60 %RH as optimum “at normal room temperature” (p. 615). However, instead of indoor temperatures, it begins with the introduction:

High relative humidity prevents effective evaporative cooling of the body during exposure to high temperatures and may lead to heat exhaustion or heat stroke and possible death.

Evaporation rate is affected by absolute humidity (or humidity ratio, HR) not relative humidity (RH). To impede evaporative cooling takes the combination of both high RH and, more importantly, high T, a T that well exceeds normal room temperatures. We labour this point again because this paper presents a common misconception: that high RH is a health hazard. The condition that causes heat related injuries is primarily that of high temperatures. Whilst high RH can exacerbate the condition, it is not the primary cause of these injuries, and one can as easily, if not more easily, succumb to heat stroke in the hot-arid desert as in a hot-humid forest. It is temperature, radiant temperature in particular, that is the key factor in “heat exhaustion or heat stroke and possible death” and not RH.

On closer inspection of Sterling’s et al. (1985) graphical presentation, a number of concerns can be raised:

1. The term “humidity” is used very loosely. Of interest to the issue of RH should be that of gaseous atmospheric humidity. However references are made to “humidifying equipment” where it is water in liquid state that becomes a breeding site for bacteria. Control of RH is immaterial to the subsequent delivery of these contaminants in the misting aerosol.
2. The literature review is based on work done with airborne pathogens but the selection does not necessarily reflect a representative sample of the most common variants in the indoor environment. Salmonella and rhinovirus are, respectively, the most striking bacterium and virus missing. Uncommon viruses like polio and cowpox are included in the mix and their relevance is somewhat stretched.
3. It is unclear what is meant by “effect” which for microbes could possibly refer to the increase in colony forming units per cubic metre of air (i.e. cfu/m³). Alternatively, it could even refer to absenteeism, occupants who complain of at least two SBS symptoms, perceived air quality, and an unending list of criteria. Without understanding the actual mechanics of each parameter it is impossible to plot wedges to indicate linear relationships between effect and RH%.
4. In many instances the relationships are non-linear to RH%. Different species of viruses also behave differently: *influenza* and *vaccinia* decrease in viability at increasing RH whereas results for *poliomyelitis* and *Venezuelan equine encephalitis* showed the opposite tendency (Harper, 1961 in Kowalski 2005). One can as easily argue that 50 %RH is the ideal condition for limiting, as it is for promoting, respective strains of virus.
5. The graphical presentation gives the impression that at certain point in the RH scale, there are no deleterious effects. A casual observer will not realise that the author of the paper admits, ‘...there is probably no level of humidity at which some biological or chemical factor that affects health negatively does not flourish’ (p. 615).
6. Many other parameters come into play to influence the ‘effect’. For example the termination of microbes from desiccation is affected by “the initial moisture content, growth phase, rate of drying, relative humidity or composition of the surrounding medium, temperature, the coincident presence of sunlight, and other factors” (Kowalski 2005 p. 113). No mention has been made if these conditions have been kept constant to isolate RH % as the determining factor in the review of literature.
7. An impression is given that all the listed contaminants should be controlled for good IAQ. However, whilst ‘viruses are almost exclusively communicable’, fungal spores, on the other hand, ‘are almost exclusively noncommunicable’ and come ‘mainly from the outdoor environment’ (Kowalski 2005 p. 4).

Thus the optimum RH chart remains an oversimplified approach for controlling air contaminants by limiting RH to a range, though this is not to say RH does not have any effect on air quality.

What is needed is to ascertain firstly, if RH has a direct correlation to contaminant proliferation and human susceptibility; and secondly, if regulating RH is

the most effective way of controlling contamination. In a hot-humid context, the discussion is focussed on the following areas of concern:

1. Bacteria
2. Viruses
3. Moulds
4. Mites

5.3.5 Effects of RH on Bacteria and Viruses

Airborne Legionella are capable of surviving “over greater distances when the relative humidity is 65 % or greater” (Kowalski 2005, p. 60). “Viruses may not behave the same as bacteria in terms of their susceptibility to variations in relative humidity. Low humidities appear to have a limited effect on virus viability while high humidities appear to decrease viability.” (Kowalski 2005, p. 113). Figure 5.6 shows a graph where the survival of influenza is plotted against relative humidity at several sample times. “The pattern is clearly a decrease in viability at increasing relative humidity.” (Kowalski 2005, p. 113)

Besides direct contact, the main mode of transmission of communicable diseases is through large droplets $>10\ \mu\text{m}$ and droplet nuclei $<10\ \mu\text{m}$ (ASHRAE 2009). ‘The residue of the droplet after evaporation, which contains any organism originally present, is called a droplet nucleus. It is typically $5\ \mu\text{m}$ or smaller in size’ (Rural Infection Control Practice Group, 2008, Sect. 2.1). At 50 %RH, a droplet of distilled water of $3\ \mu\text{m}$ can be reduced to a tenth of its original size in a fraction of a second (Dunklin and Puck 1948). Figure 5.7 shows that RH has an impact on the initial evaporation rates, with droplets evaporating about four times faster at 0 %RH than at 70 %RH. However, droplets which contain bacteria do not evaporate completely, and despite similar initial evaporation rates, droplets with bacteria retain the nuclei for much longer than those without. The authors of this graph observed:

Fig. 5.6 Influenza viability under different exposure of RH% and exposure (Clarke et al. 1999, in Kowalski 2005)

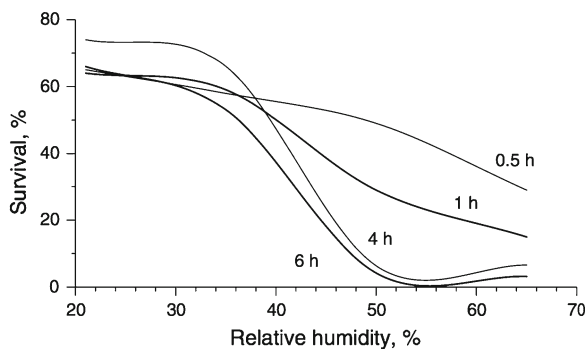
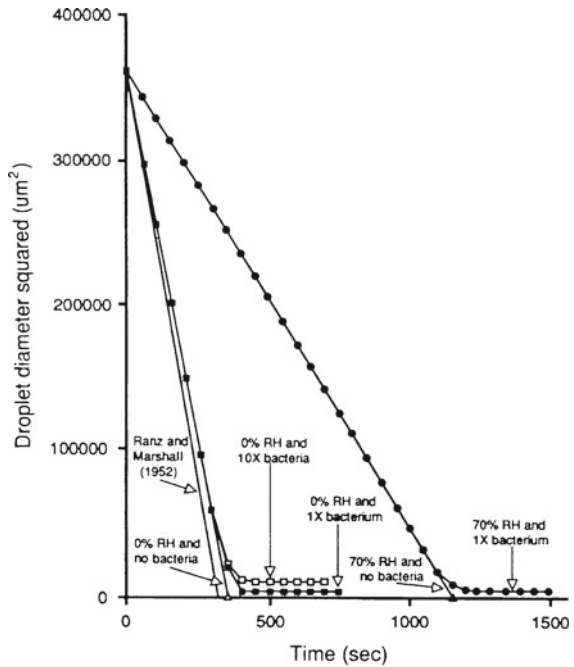


Fig. 5.7 Simulated water evaporation from bacteria containing 600 μm diameter water droplets in an atmosphere of 0 %RH or 70 %RH and 25 $^{\circ}\text{C}$ (Lighthart and Kim 1989)



the graph shows how droplet-contained bacteria might not appreciably affect droplet evaporation until almost all of the droplet water had evaporated and the agglomerated bacteria or droplet nuclei were left. At this point the survival of the agglomerated bacteria might depend on such unknowns as the water evaporation rate, the position in the agglomerate, and the life-sustaining water content of the bacteria. (Lighthart and Kim 1989, p. 2351)

They suggest that the following mechanisms are involved with this phenomena:

...it is hypothesized that as droplet water evaporates, cells, which are thought to have a lower water vapor pressure (e.g., due to lipid-containing cell membranes), may be present in the droplet air–water film, lowering the overall droplet evaporation rate. Carrying the argument further, it is hypothesized that as the cells agglomerate or form a protective shell configuration during evaporation, the outermost cells would dry to a critical (life-sustaining) water content before those cells deeper in the agglomerate do. From this, it might be expected that those relatively large droplets that originate with the most microbes would contain at least one deeply embedded viable microorganism for the longest period of transport time downwind. Thus, the larger droplet nuclei presumably will have more surviving microbes over a longer time. (Lighthart and Kim 1989, p. 2354)

Evaporation of these agglomerated droplets under typical room conditions is not immediate enough to control disease spread and the most effective way of removing these droplets or droplet nuclei is by filtration. Filtration, however, is significantly affected by RH. Below 80 %RH, air filters led to a marked reduction of airborne bacteria concentrations by approximately 70 %. Above 80 %RH, a proliferation of bacteria on air filters resulted in a subsequent release back into the

filtered air (Kowalski 2005, p. 225). Hence where air filters are used, care must be exercised to control the subsequent re-entry of bacteria into the air stream at >80 %RH.

For bacteria and viruses, air sanitisation has been the most reliable control. For instance, Langmuir (1948, in Fisk 1999) reported 23 % decrease in respiratory illness with UV irradiation during an epidemic in U.S. Army barracks.

5.3.6 Effects of RH on Moulds

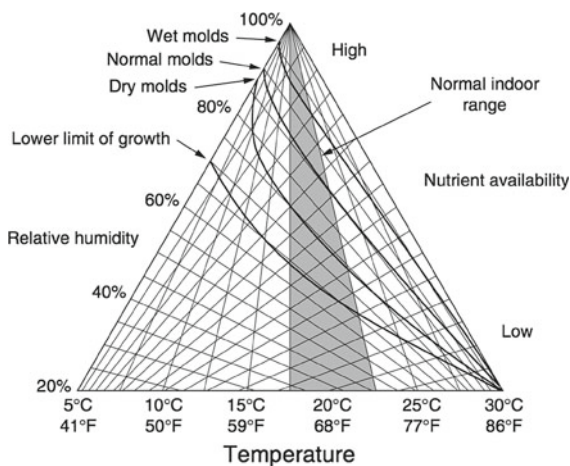
Though moulds are non-communicable, the spores of some species, in high concentrations, can be toxic to humans. Mould growth occurs at the confluence of humidity, temperature and the availability of nutrients (Fig. 5.8). In referring to building enclosures, Kowalski (2005, pp. 175–177) notes,

Evidence exists to suggest that many of the building materials like gypsum and wallpaper that favor the growth of fungi also influence the generation of potentially hazardous mycotoxins. Bacteria can influence the growth of fungi. Environmental bacteria can grow biofilms, and thereby provide fungal spores a nutrient base. Biofilms generally grow where there is excessive moisture or where water collects, such as drain pans, dehumidification cooling coils, and sump type humidifiers.

The indoor relative humidity itself is less an indicator of mould growth than is the water activity (A_w) of the building materials (Kowalski 2005, p. 174), determined as follows:

$$A_w = \frac{P}{p_o} \quad (5.1)$$

Fig. 5.8 Growth conditions for mould (Clark 1998, in Kowalski 2005, p. 176)



 Equation notes

P	Vapour pressure of water in the substance
P_o	Vapour pressure of pure water at the same temperature

Kowalski further observes, ‘Fungal growth is likely if the water activity exceeds 0.76 to 0.96, depending on fungal species, temperature, time, and composition of the material (Pasanen et al. 1992, in Kowalski [2005]).’

Thus the way to control mould growth is thus by selecting the least hygroscopic materials and finishing them with hydrophilic paints. Alternatively, mould can be controlled by air filters, where it was found that the lack of available nutrients on clean filters generally precluded most mould growth even under high RH (Kowalski 2005, p. 223).

5.3.7 *Effects of RH on Mites*

In a review on ‘Controlling Dust Mites Psychrometrically’, M. J. Cunningham of the Building Research Association New Zealand (BRANZ) explains his rationale for using humidity control on mites management, ‘Higher humidities provide conditions suitable for mite population survival and growth. It also results in greater faecal production per mite, resulting in higher allergen levels (Cunningham 1996).’ However his research shows that low temperatures drastically impeded mite population growth rate, and at a rate far more significant than RH. He further notes that regulating the humidity in a room had little effect on the microclimate where mites were typically found (e.g. pillows, carpets).

It would thus be ineffective to regulate ambient T, and even less effective to regulate ambient RH, in order to control mite populations and their allergenic effects. To control mites it would be far more effective to deal with them in their habitats by vacuuming, chemical sanitisation, extreme heat or cold, or simply regular replacement of items like pillows.

5.3.8 *Summary of Findings Compared to Sterling et al. (1985)*

Sterling’s work was for its time a significant contribution, and the optimum RH recommendation was based on tenable ideas. We must however recognise that this was a literature review and not basic research involving testing the contaminants, particularly the biological ones, with the specific intent of determining their viability under varying ambient RH conditions. After a quarter of a century, huge advancements have been made in microbiology, with potential epidemics like SARS and avian flu fuelling very active research into the area. It is needful that this work be

updated, if possible. As the sole reference in the ASHRAE Handbook and AIRAH Guide, and on which other standards and legal requirements are based, the limitations to our knowledge of such an optimum RH should be openly acknowledged.

The problem is that microbiologists themselves recognise that ‘the effects of relative humidity on airborne microbes are complex and involve phase changes at the molecular level’ (Kowalski 2005, p. 157). With so many aspects of aerobiological behaviour not fully understood it is no wonder that in the *ASHRAE Position Document on Airborne Infectious Diseases*, the only mention of humidity is that

humidity affects survival of the infectious agent although not always in predictable ways. (ASHRAE 2009, p. 5)

Controlling RH% in today’s context is, without doubt, a blunt and energetically costly way of controlling microbes, and for some viruses it is of questionable efficacy. There can be many other ways of controlling each factor that affects IAQ as tabulated in Table 5.8.

If indoor air quality (IAQ) were the real issue, than there are other approaches to improving IAQ without dehumidification. Instead of controlling humidity, designers could look at paints and finishes that inhibited microbial growth on walls and ceilings. They could also minimise the amount of soft finishes (carpets, drapery, fabric, etc.) used in interiors. The key is to look at materials with low water activity and low nutrient content for moulds. As an air quality issue, filtration and sanitisation devices could be employed to reduce, remove or destroy microscopic organisms. All these could be done, with less operational energy than to dehumidify air, an energy demand particularly strenuous in tropical cities.

However, when humidity is made a thermal comfort issue, the need for compliance does not leave designers any room to explore alternative solutions. In an energy-constrained world we need to evaluate the issues separately: lowering of temperatures for improved comfort which is distinct from removing of microbes for improved air quality—of which lowering humidity is but one of many viable strategies.

In as much as we can demonstrate energy efficient alternatives to improving IAQ, one reason why the control of indoor climates is so energetically inefficient is from the inclusion of humidity control as a comfort issue. Other than instances where air quality is crucial (like in hospitals), the inclusion of humidity control as a comfort consideration nullifies the need to adopt any of these alternative strategies.

Table 5.8 Alternatives to RH control in managing various microbes

Virus and Bacteria	Air filtration (Fisk 1999) and sanitisation (UV, ozone), control RH of filter (can be achieved by heating air passing through filter)
Mould	Select building materials with low water activity and low nutrient content, maintain clean filters, control bacteria biofilms
Mites	Control amount and type of soft finishes, reducing temperature, change pillows, steam cleaning, vacuuming with fine filter

5.3.9 Humidity and Disease

I am New People. Your sicknesses do not frighten me. (Bacigalupi 2009, p. 488)

If microbes had no pathogenic effects on the windup girl, there would be no necessity for humidity control. Adopting a purist approach to thermal comfort, the most efficient route would be firstly by evaporative cooling and, if necessary, further refrigeration to reach thermal neutrality. There will be no need to dehumidify (to a humidity ratio of 12 g/kg) and reheat (till RH is at 65 %). Designing an energy efficient climate chamber for the windup girl has uncovered a new index for evaluating the effectiveness of climate control processes: comfort energetics. This hypothetical client, with her genetically engineered resistance to disease, also helps to distil the thermal comfort considerations from indoor air quality ones.

Nevertheless, we are not “New People”, and high RH is a potential sick building syndrome (SBS) problem. Besides SBS, there are also other ways in which RH affects perceived air quality, odours and eye irritation, though these are not covered here as they have not been cited in the standards as reasons for controlling humidity. Recognising that the standards cannot be easily amended, compliance will still be obligatory for the immediate future. If we intended to improve thermal comfort with less energy and without increasing the RH of a space, what viable options remain?

5.4 Energetics of Fans

Yellow card coolies crank at wide-bore fans, driving air through the club. Sweat drips from their faces and runs in gleaming rivulets down their backs. They burn calories as quickly as they consume them and yet still the club bakes with the memory of the afternoon sun. (Bacigalupi 2009, pp. 55–56)

The cooling effect of moving air can be analysed in three methods:

1. Altering the wind speed parameter in the PMV
2. Determining the effective dT of moving air and adjusting the operative T in PMV to account for this effect.
3. Determining the wind chill index (WCI) through wind velocity and air temperature ASHRAE Handbook: Fundamentals (ASHRAE 2005, 8.21). The methods discussed for methods 1 and 2 could also be applied to WCI , using the energy required to increase v to evaluate the lower vote from WCI . However, as the wind chill index has been developed for outdoor use, it is not considered in our discussion on comfort energetics.

Table 5.9 Comfort energetics of increasing ventilation rates

Increased ventilation				
Ventilation (l/s)	Velocity (m/s)	PMV	Power (W)	W/vote
10	0.05	1.994	5	
20	0.1	1.937	10	175.4
30	0.15	1.873	15	124.0
40	0.2	1.835	20	125.8
50	0.25	1.804	25	131.6
60	0.3	1.778	30	138.9
70	0.35	1.755	35	146.4
80	0.4	1.735	40	154.4
90	0.45	1.718	45	163.0
100	0.5	1.702	50	171.2

5.4.1 Wind Velocity in the PMV Model

Under this setup, the flow rate can be assumed to have a linear relationship with power consumption: a ventilation fan is installed in the ceiling with a fixed maximum power (5 W) and exhaust rate (10 l/s). Additional fans are installed, up to 10 units, all running at the same speed and power, giving a linear relationship between power and exhaust rates. The fan-only cooling under this calculation averages 148 W/vote (s.d. = 11 %, refer to Table 5.9) and does not improve comfort by much.

5.4.2 dT as a Result of Increased Cooling with Higher Wind Speed

Instead of looking at the fans as performing exhaust ventilation, we can see them as circulating the air, as in the case of ceiling fans. In this scenario it is assumed that the occupants will sit in the region where the fan speed is about 1 m/s (Rohles 1983, figure). The moving air across a lightly clothed occupant will cause a difference in temperature (dT) from the forced convection. Various models exist (Emmanuel 2005, p. 69) to determine the magnitude of the dT . At 1 m/s the average of these eight models is $dT = -3.89$ °C (s.d. = 1.69 m/s) in agreement with the difference in equivalent temperature of 3.3 °C when ceiling fans are compared to still air (Rohles 1983). This difference is applied to determine the operative T for the PMV model (ASHRAE 55-2004).

$$T_{operative} = A \cdot T_{air} + (1 - A) \cdot MRT \quad (5.2)$$

Equation notes

A	Ratio of air temperature	0.7 (at vel = 1 m/s)
T air	Air temperature, corrected for wind at 1 m/s	(30–3.89) °C
MRT	Mean Radiant Temperature	30 °C

Table 5.10 Comfort energetics of a ceiling fan

Ceiling fan reference	
Motor power (Martec 1.3 m)	69 W
Free air displacement	3,770 l/s
Assumed wind velocity	1.0 m/s
T (operative)	27.28 °C
Reference PMV	1.994
PMV with wind cooling	1.159
Comfort energetics for 1 pax	83 W/vote
Comfort energetics for 4 pax	21 W/vote

Based on a 69 W ceiling fan (Table 5.10), the comfort energetics works out as 83 W/vote for a single occupant. However, recognising that a ceiling fan can support a whole room (see Fig. 5.9), the comfort energetics could be as low as 21 W/vote if 4 occupants could be positioned within the 1 m/s range of the ceiling fan.

In the ASHRAE standards, draft rates (DR) are used to predict the percentage of people dissatisfied with a draft condition based on the work of Fanger and Christensen (1986) and Fanger et al. (1988) with the following expression:

$$DR = (34 - T_a \times [v - 0.05]^{0.62}) \times (0.37 \cdot v \cdot Tu + 3.14) \tag{5.3}$$

Equation notes

v	Mean velocity
t	Air temperature
Tu	Turbulence intensity

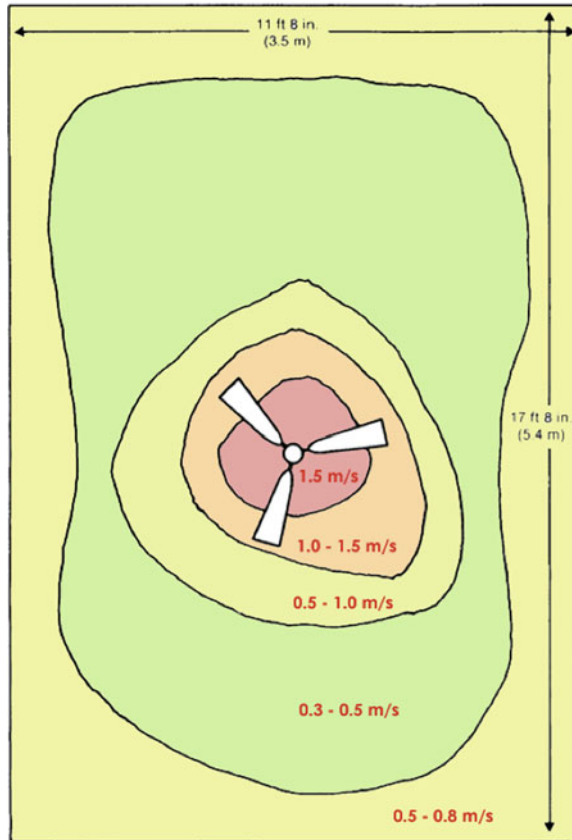
The turbulence intensity (Tu) for ceiling fans were measured to be 7.1 % under the middle of blades and 3.5 % at 300 mm from the outer edge of blades (Chiang et al. 2007). Based on a DR <20 % stipulated under ASHRAE 55, the air speed cannot exceed 0.9 m/s. However other researchers (Zhang et al. 2007) present a case to raise the draft limit. It was found that even when the draft standard had been exceeded, 41 % of occupants feeling neutral and 68 % for occupants feeling warm still wanted more air motion.

5.4.3 How Much Air Movement is Considered a Draft?

The maximum permissible air speed has varied at different times between researchers:

1. ‘0.2 m/s is the de facto draft limit in the current ASHRAE and ISO thermal environment standards’ (Zhang et al. 2007, when referring to ASHRAE 55-2004).

Fig. 5.9 Wind speeds under a ceiling fan (Cook 1989, p.46)



- The ASHRAE Standard also states that 'loose paper, hair and other light objects may start to be blown about at air movements of 160 fpm (0.8 m/s).' (Rohles 1983 when referring to ASHRAE 55-1981)

It should be emphasized that the 1.0 m/s (200 ft/min) air speed was not demonstrated to be a maximum allowed speed. ASHRAE defines 0.8 m/s (160 ft/min) as an upper limit on uniform air speed in occupied spaces based on motion of hair, paper, and other light objects. Such annoyance was not observed at 1.0 m/s (200 ft/min) in the Kansas State experiments. In an earlier experiment with ceiling fans mounted above a flow straightening louver, McIntyre concluded that 2 m/s (400 ft/min) was the upper limit on acceptable air speed (McIntyre 1976). ...However, because the rate of convective heat removal increases as only the square root of the air speed, higher air speeds provide diminishing comfort returns at increased power requirements (McIntyre 1976). Clark, "Passive Cooling Systems" in Cook (1989), p. 366.

High air velocities of up to 1.6 m/s were preferred by Japanese subjects exposed to a modified temperature of 31 °C (Tanabe et al. 1987). Modified temperature was defined as the ambient temperature connected to the equivalent

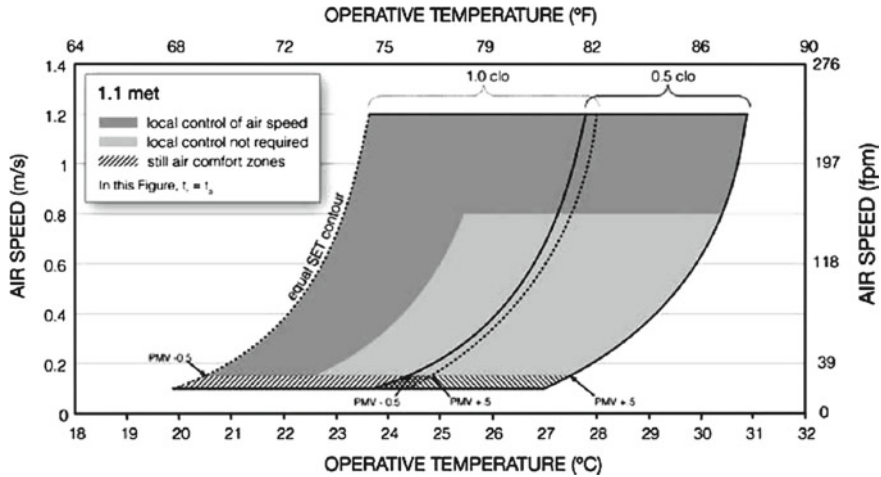


Fig. 5.10 Acceptable operative temperature and air speeds for the comfort zone at humidity ratio of 0.010 (ASHRAE 55, 2010, in Turner 2011)

condition at 50 %RH and 0.1 m/s air velocity, according to the PMV-model, that would be felt equally warm by a subject wearing 0.6 clo.

Toftum (2004) when referring to the highest category (in terms of measuring instrumentation and rigour) in the ASHRAE RP884 Project “*The World Database of Thermal Comfort Field Experiments*” commented:

The distribution of air velocities measured during these field studies was skewed towards rather low values. Thus, out of a total number of 5653 observations, only 8 % resulted in air velocities higher than 0.2 m/s, 3 % higher than 0.25 m/s and 2 % higher than 0.3 m/s. This alone may explain why occupants under warm conditions frequently expressed a desire for more air movement rather than complained of draught.

The latest ASHRAE 55 (2010) standard recognises this research and has increased the maximum permissible operative temperatures dramatically from traditional still-air comfort zones (Turner 2011) as seen in Fig. 5.10.

5.4.4 Air Movement Summarised

Relating air movement to PMV is heavily dependent on the method in which airflow is generated. In an air-conditioned office, the increased air speed is achieved by increased supply-air flow rate, and in a naturally ventilated office by the recirculation with a ceiling fan. With the acceptance of higher air velocities in the ASHRAE 55 standard, it might even be possible to have a hybrid system of fans within an air-conditioned office space. These methods are concerned with general flow rates across an occupant and do not consider the different sensitivities of body parts to air movement. To understand this, we turn our discussion to a specialised form of air movement known as personalised ventilation.

5.5 Personalised Ventilation

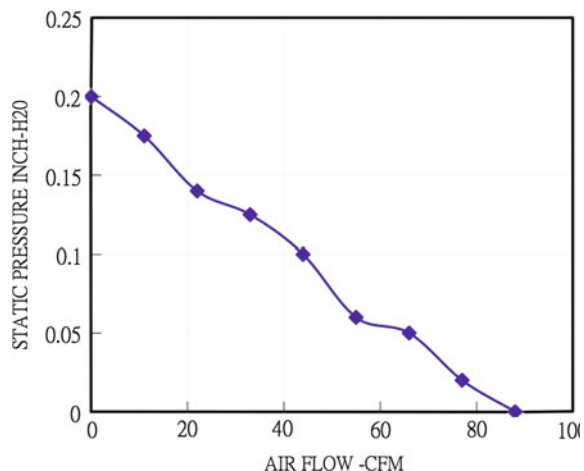
An alternative approach to air movement would be to apply personalised ventilation (PV). PV works on the principle each occupant should have direct control over the immediate thermal environment in order to adjust it to their personal preference. PV is delivered locally at each workstation through air terminal devices (ATDs) that typically cool the upper part of a seated occupant.

10 l/s is negligible flow rate under PMV model. In the reference climate booth the air speed would average 0.01 m/s and could potentially reach 0.05 m/s in the constricted areas where the body is close to the wall. In a 120 mm diameter fan this same flow rate will produce a face velocity of 0.7 m/s. When applied frontally to the face, air speeds of up to 0.9 m/s (Gong et al. 2006) were found to be acceptable to 96 % of occupants at an ambient and task air supply temperature of 26 °C. This gives another indicator of the energetics of personal ventilation. The PMV at the experiment conditions would be +0.596 without air movement, and the actual vote is assumed as 0 with air flow directed to the face (96 % satisfaction will surpass the 5 % PPD at PMV = 0).

To achieve 0.9 m/s air speed, a flow rate exceeding 23 l/s (49 cfm) is required and this be easily achieved by a 120 mm axial fan (used in computer cases, see Fig. 5.11 for a sample specification) running at 7 W, yielding 11.7 W/vote in comfort energetics.

Schiavon (2009) developed a cooling fan efficiency (CFE) index comparing the AT equivalent of whole body cooling effect with the power input for the fan, expressed in °C/W. He determined that the most effective cooling device was the desk fan (16–20 W) with an efficiency of 0.123 °C/W. We also see from Fig. 5.12 the AT has a practical limit of −3 °C. Within this range, personal ventilation

Fig. 5.11 Flow rate of 120 mm diameter fan at 3,000 rpm (Specifications from manufacturer: Commonwealth Industrial Corporation)



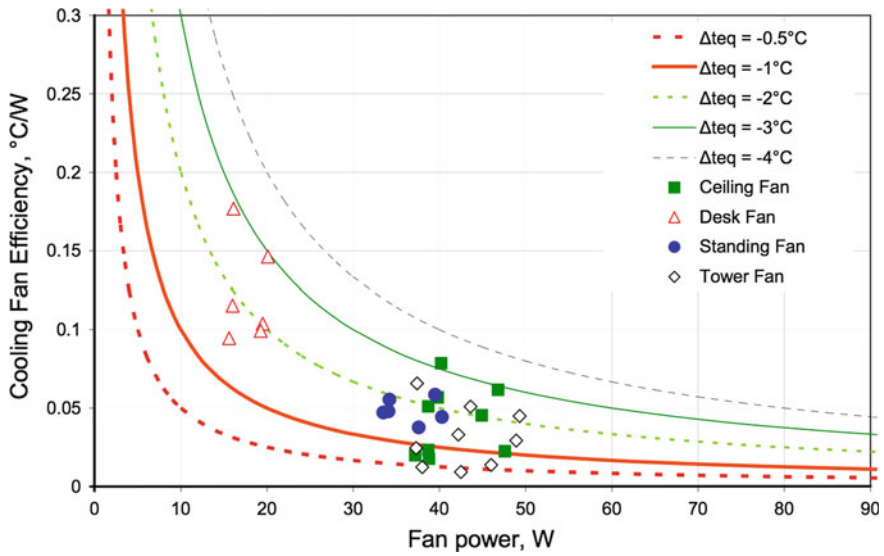


Fig. 5.12 Fan power versus cooling fan efficiency index for the ceiling fan (CF) the desk fan (DF), the standing fan (SF), and the tower fan (TF). Lines with constant cooling effect ($\Delta T >eq$) are plotted (Schiavon 2009)

yields 26.3 W/vote (Table 5.11). Most significant to the difference between the comfort energetics of desk fans (26 W/vote) and computer fans (12 W/vote) is the power input. At a fan power of 10 W, the CFE is double that at 20 W and places the earlier estimation in the same ball-park as Schiavon’s detailed model.

...the required power input of the fan is a critical factor for achieving energy saving at elevated room temperature... The power of a fan used to cool people by increasing air movement should be in general less than 20 W. ...Computer fans can be used as cooling fans because their power input is extremely low while the generated flow rate is quite high. (Schiavon 2009, pp. 43–45)

Table 5.11 Comfort energetics of a desk fan based on Schiavon’s CFE index

Cooling fan efficiency					
T	Fan power (W)	CFE = 0.123 °C/W	PMV	dPMV	Elect W/vote
30.0			1.994		
29.5	4.1		1.840	-0.154	26.4
29.0	8.1		1.685	-0.309	26.3
28.5	12.2		1.530	-0.464	26.3
28.0	16.3		1.375	-0.619	26.3
27.5	20.3		1.221	-0.773	26.3
27.0	24.4		1.066	-0.928	26.3

Table 5.12 Summary of comfort energetics for cooling

	Comfort approach	Process	Comfort energetics (W/vote)
1	Computer fan	Wind directed only on face	12
2	Direct evaporative cooling	Adiabatic humidification	14
3	Refrigeration up to dew point	Constant humidity ratio (isodrosothermic)	18
4	Ceiling fan	Recirculation (shared 4 pax)	21
5	Desk fan	Wind directed to upper body	26
6	Refrigeration past dew point	Cooling with dehumidification	29
7	Ceiling fan	Recirculation (1 pax)	83
8	Refrigeration past dew point	To ASHRAE standards	76–124 (depending on reheat efficiency)
9	Extraction fan	Exhaust ventilation	148
10	Desiccant dehumidification	Adiabatic + isothermic dehumidification	277

5.6 Comfort Energetics, in Summary

In summary, the various processes can be ranked in the power each process consumes to effect an improvement of one vote. A fan power consumption of 7 W is here added to the computation where it had been earlier omitted to give a common baseline against fan-based cooling and tabulated in (Table 5.12).

5.7 The Limit of Comfort Energetics

Comfort energetics offers a basis to compare the efficiency of different approaches to comfort. As a theoretical exercise it separates peripheral issues like RH, which is more an air quality issue, from the central components of thermal comfort: air temperature and air movement. Whilst the human sensitivity to MRT is acknowledged, it is often not significantly different from air temperature for office occupants sitting away from the windows.

It needs to be noted that these tabulations are based on PMV scenarios. Even for processes that show linearity it may not be feasible to use one process alone to improve comfort by 2 to 3 votes. Based on the science of thermal comfort, for the best comfort energetics in a hot-humid climate, it is noted that:

1. PMV is most effectively improved by reducing temperature
2. increased air movement is generally welcome
3. high RH not a comfort issue, and arguably not even an IAQ issue
4. IAQ is better improved with filtration and sanitisation than with dehumidification

5.8 Conclusion to Comfort Energetics

With the wet season, Emiko's life becomes bearable. The flooded metropolis means that there is always water nearby, even if it is a stagnant bathtub stinking with the refuse of millions. ...She eats well and sleeps easily, and with water all around, she does not so greatly fear the heat that burns within her. If it is not the place for New People that she once imagined, it is still a niche. (Bacigalupi 2009, p. 501)

In the tropics the options for cooling are greatly limited with high humidity. The difficulty with cooling in the tropics is that the air is already near saturation, limiting the options for evaporative cooling. The high humidity and cloud cover also limit the availability of radiant cooling to the night sky. Passive cooling in the tropics is often limited simply to increased cross-ventilation, which requires adequate spacing between blocks to encourage airflow.

In developing tropical cities confronted with poor air quality, massive demand for prime commercial real estate and the need for a productive workforce, the ubiquity of offices that are fully-glazed, deep-planned and air-conditioned is inevitable. Where increasing the humidity ratio is not an option, the best improvements in terms of comfort energetics will be, in order:

1. Low-power computer fan blowing at the face
2. Refrigeration of air, up to and not below dew point
3. Ceiling fan covering multiple occupants

However, the long-term acceptance of air-conditioning marketing, the acclimatised addiction to air-conditioning standards and the social expectations for air-conditioning all indicate that there will be massive resistance from the general public to accept the proposition that moving air will give them just as much comfort as air-conditioning. Whilst low-power fans can use far less energy than air-conditioning to ASHRAE standards, it is unlikely, at this stage, to provide an acceptable solution in Singaporean office buildings. It is acknowledged that this may be a disheartening conclusion unless there are mechanisms to change behaviour, an endeavour not within the scope of this thesis.

It would appear that to solve the problem of over reliance on energy-intensive air-conditioning, the solution is not to avoid air-conditioning altogether, but rather to make it less energy-intensive. To do this we ask the following questions:

1. Without resorting to evaporative cooling, increased air movement or radiant sky cooling, can cooling still be achieved passively?
2. If it cannot be done passively, what are the limits to which the energy inputs for cooling can be minimised?

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