



JAMES COOK UNIVERSITY

TOWNSVILLE Queensland 4811 Australia Telephone: (07) 4781 4111

Australian Institute of Tropical Architecture

Director: Professor R M Aynsley

Telephone: (07) 4781 4147

International: 61 7 4781 4147

Facsimile: (07) 4781 5766

E-mail: Richard.Aynsley@jcu.edu.au

J. SHIEL

OPTIONS FOR ASSESSMENT OF THERMAL COMFORT/DISCOMFORT

for

AGGREGATION INTO NatHERS STAR RATINGS

prepared by

R.M. Aynsley and S.V. Szokolay

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Campuses at –

TOWNSVILLE
(07) 4781 4111

CAIRNS
(07) 4042 1111

MACKAY
(07) 4957 6048

Tasks set for the consultant:

Part 1

1. Briefly review the available measures of thermal comfort/discomfort, including indices of thermal stress. Evaluate the practical application of each in the context of the nationwide HERS and the NatHERS software. Any which are clearly inappropriate for use in this context, such as those dealing with stress in outdoor workers, should be merely noted as such, while others which may appear to be more appropriate, such as effective temperature, should be explored in greater depth.
2. Recommend the most appropriate measure for adoption into the scheme, or if two or more are deemed equally suitable, select those and discuss in more detail their advantages and disadvantages. It is imperative that the report clearly recommends at least one appropriate measure.
3. Discuss and recommend the actual values to be used as the basis for measurement of discomfort in the domestic context in each of the 28 locations used in NatHERS. If an index is used, provide examples for Darwin and Alice Springs of some combinations of factors which give relevant values.
4. Specify the means by which the measurements of discomfort can be aggregated (or otherwise manipulated) into star ratings that reflect the sum of discomfort over the entire year, and are compatible with energy-based star ratings.

MEASURES OF THERMAL COMFORT

Physical measures

Simple	DBT	dry bulb (air) temperature
	WBT	wet bulb temperature
	GT	globe temperature (measured with a 150 mm diameter black globe)
Composite	MRT	mean radiant temperature: solid angle-weighted temperature of surrounding surfaces
	DRT	dry resultant temperature: $\frac{1}{2} \text{ DBT} + \frac{1}{2} \text{ MRT}$
	EnvT	environmental temperature: $\frac{1}{3} \text{ DBT} + \frac{2}{3} \text{ MRT}$
	WGBT	wet bulb globe temperature for indoors: $0.7 \text{ WBT} + 0.3 \text{ GT}$ for outdoors: $0.7 \text{ WBT} + 0.2 \text{ GT} + 0.1 \text{ DBT}$ (using naturally ventilated WBT)

The following are described in detail:

Empirical measures: comfort indices

ET	effective temperature	page 2
CET	corrected effective temperature	3
OT	operative temperature	4
EqT	equivalent temperature	5
EqW	equivalent warmth	5
RT	resultant temperature	6
ECI	equatorial comfort index	7
Tsi	tropical summer index	8

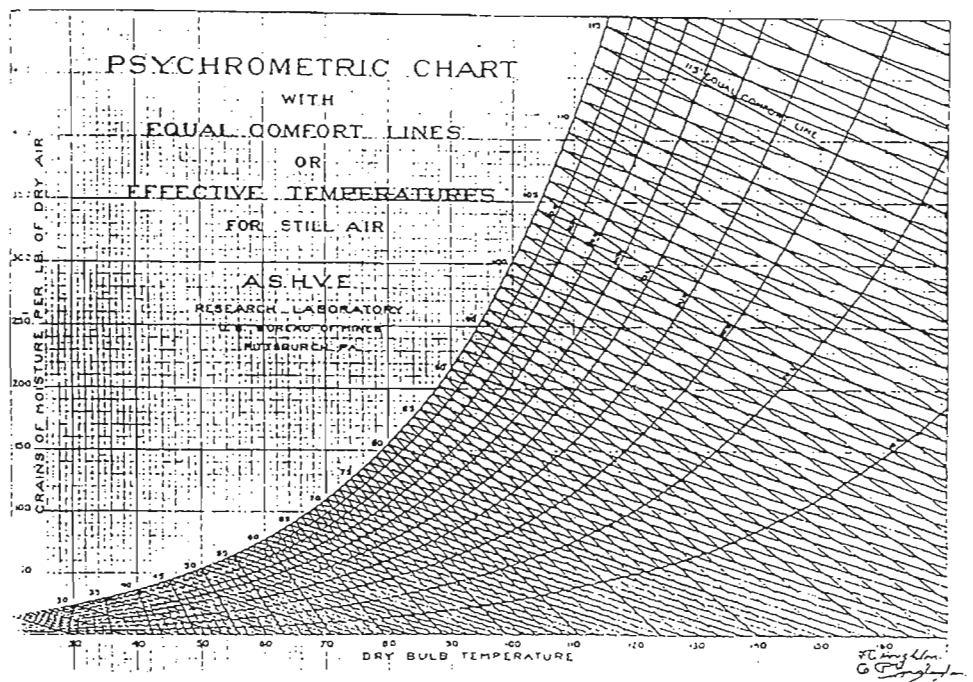
Analytical measures: stress indices

TSI	thermal strain index	9
TAR	thermal acceptance ratio	9
P4SR	predicted 4-hour sweat rate	10
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RSI	relative strain index	12
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SET	standard effective temperature	16
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Effective temperature (ET)

Developed by Houghten and Yagloglou at the ASHVE Pittsburgh research laboratories in 1923; represented by a set of equal comfort lines drawn on the psychrometric chart. It is defined as the temperature of a still, saturated atmosphere, which would, in the absence of radiation, produce the same effect as the atmosphere in question. It thus combines the effect of dry air temperature and humidity.



Yagloglou's original representation of the effective temperature.

Yaglou in 1947 (who shortened his name by then) already noted that the ET overestimates the effect of humidity. Smith (1955) found that the relationship is not linear and that the P4SR index gives a better correlation with comfort votes.

Glickman et al. (1950) also found that ET overestimates the effect of humidity under both cool and comfortable conditions.

Glickman, N, T Inouye, Keeton, R W and Fahnestock, M K (1950): Physiological examination of the effective temperature index, *ASHVE Trans.* 56:51

Houghten, F C & Yagloglou, C P (1923): Determination of comfort zone. *Trans Am Soc Heat Vent Engrs.* 29:361

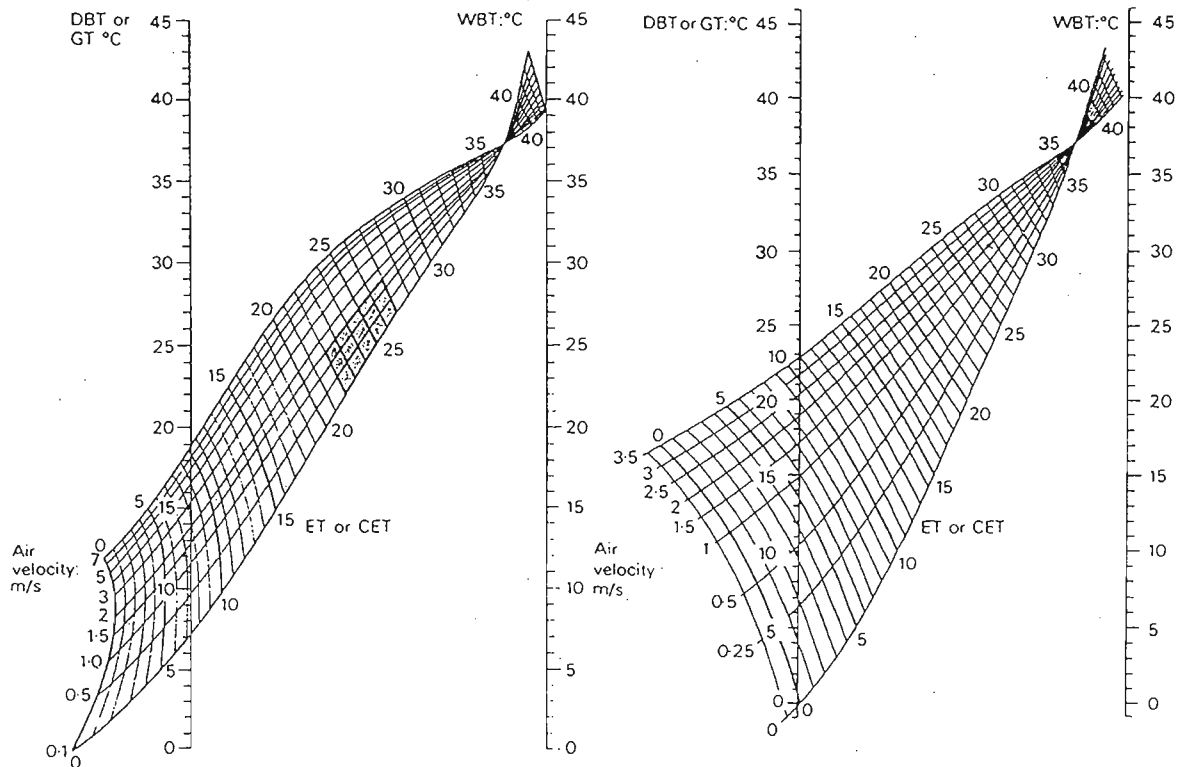
Houghten, F C & Yagloglou, C P (1923/a): Determining equal comfort lines. *J Am Soc Heat Vent Engrs.* 29:165

Yagloglou, C P (1927): The comfort zone for men. *J of Industrial Hygiene.* 9:251

Yaglou, C P (1947): A method for improving the effective temperature index. *Trans ASHVE*, 53:307

Corrected effective temperature (CET)

ASHVE (1932) published a nomogram representation of the ET index, which included air velocity effects and showed that over about 100°F (37.8°C) air movement increases the thermal load (hence the reversal of the air velocity lines). Bedford (1940) included the effect of radiation by substituting globe temperature values for the dry bulb temperature scale. This became known as the CET nomogram. As clothing has a large influence on radiation and wind effects, he produced two nomograms: for people wearing 1 clo clothing and for people stripped to the waist:



CET nomograms for people wearing 1 clo clothing and stripped to the waist

Smith (1955) found that in hot environments the effect of humidity is underestimated and that the adverse effect of 0.5 – 1.5 m/s air velocities at high temperatures is overestimated. Givoni (1963) however established that above 32°C air movements produced a greater heating effect than that suggested by the ET.

ASHVE (1932): *Guide*. Am Soc Heat Vent Engrs

Bedford, T (1936): *Warmth factor in comfort at work*. Med Res Council, Industr Health Res Board, Report No 76. HMSO

Bedford, T (1940): *Environmental warmth and its measurement*. Med Res Council, War Memorandum No 17. HMSO

Givoni, B (1963): *Estimation of the effect of climate on man*. Research Report to UNESCO. BRS Technion, Haifa

Operative temperature (OT)

This index was produced by Winslow, Herrington & Gagge, as a result of work similar to Bedford's. It is defined as the temperature of a uniform, isothermal "black" enclosure in which man would exchange heat by radiation and convection at the same rate as in the given non-uniform environment; or by the following expression:

$$OT = \frac{h_r MRT + h_c DBT}{h_r + h_c}$$

where h_r and h_c are radiation and convection coefficients

This index integrates the effect of air temperature and radiation, but ignores humidity and air movement. The study was carried out under cool conditions, where the effect of humidity was small and indoor air movement negligible.

There is a subsequent correction for air movement:

$$OT = \frac{h_r MRT + h_c \left[DBT * \sqrt{\frac{v}{v_o}} - t_s \left(\sqrt{\frac{v}{v_o}} - 1 \right) \right]}{h_r + h_c}$$

where v air velocity (ft/min)
 v_o reference velocity (15 ft/min \approx 0.075 m/s)
 t_s skin temperature

If $MRT = DBT$ and air movement is negligible, then $OT = DBT$

The index is thought to be unsuitable above 27°C (Givoni, 1962)

Winslow, C E A, Herrington, L P & Gagge, A P (1937): Physiological reactions to environmental temperature. *Am J of Physiology*, 120:1-22

Givoni, B (1962): *The nature and application of thermal indices*. Bulletin No 73-74. Israel Inst of Technology

Equivalent temperature (EqT)

This scale has been introduced by Duffon (1932 & 1933) and its use is described by Bedford (1951). Its conceptual definition is: the temperature of a uniform enclosure, with still air, in which a sizeable black body at 24°C (75°F) would lose heat at the same rate as that observed. Mathematically the definition is (quoted in original units):

$$\text{EqT} = 0.522t_a + 0.478t_w - 0.0147v^{0.5}(100 - t_a)$$

where t_a = air temperature (°F)

t_w = mean radiant temperature (°F)

v = air velocity (ft/min)

or

$$\text{EqT} = 0.522t_a + 0.478t_g + v^{0.5}(0.0808t_g - 0.0661t_a - 1.474)$$

where t_g = globe temperature (°F)

Bedford (1936) devised a nomogram for determining the EqT from measured individual thermal factors.

This index does not take into account humidity, thus it is unsuitable for temperature higher than about 24°C, as at such levels humidity becomes increasingly important.

Bedford, T (1951): Equivalent temperature, what it is, how it's measured. Heating, Piping, Air conditioning, Aug. p.87-91

Duffon, A F (1932): *Equivalent temperature and its measurement*, B R Technical Paper 13. HMSO

Duffon, A F (1933): The use of kata thermometers for the measurement of equivalent temperature. *J Hygiene, Camb.* 33:349

Equivalent warmth (EqW)

This index was developed by Bedford, based on experiments with over 2000 factory workers, engaged in light work, under varying indoor conditions. Air temperature, humidity and mean radiant temperature were measured and recorded and correlated with subjective responses of the subjects. Clothing and skin temperatures were also recorded. This correlation produced the EqW scale.

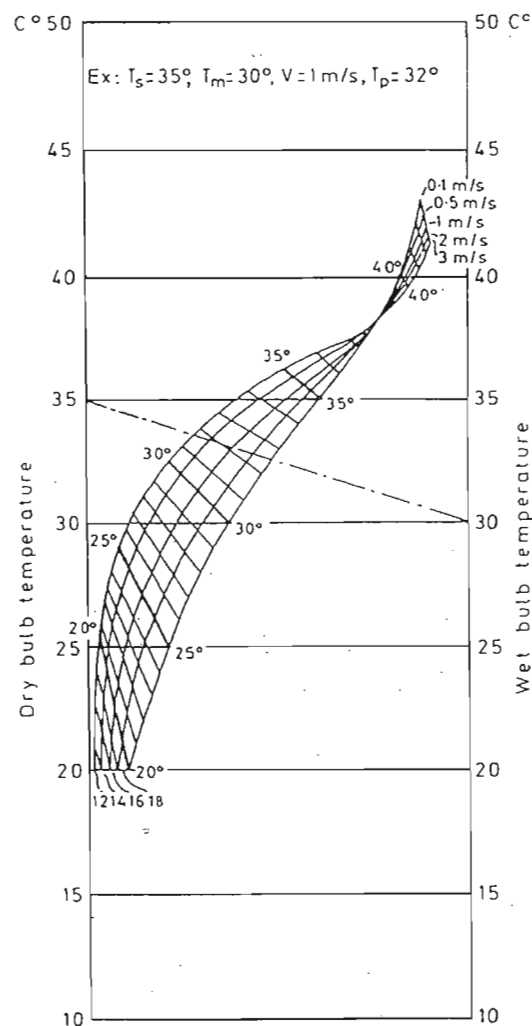
It is now thought to be reliable within the comfort range and up to 35°C with low RH or up to 30°C with high RH. It underestimates the cooling effect of air movement with high humidities.

Bedford, T (1936): *Warmth factor in comfort at work*, Med Res Council, War Memorandum No 17. HMSO

Resultant temperature (RT)

This index was developed by Missénard in France. It is based on measurements and votes in a test room, after 0.5 hour of adjustment (as opposed to the ET scale, also based on test room measurements, but on instantaneous reactions).

It is a slight improvement on the ET scale, but only for rest or low activity conditions. The nomogram defining it is almost identical with the ET nomogram. It is thought to be reliable for moderate climates, but not for tropical conditions, as it underestimates the cooling effect of air movements at temperatures above 35°C and over RH 80%, while at lower values of the RT the effect of air movement is overestimated.



Resultant temperature nomogram by Missénard

Givoni (1969) found that the RT is in better agreement with observed physiological responses than the ET, although below 30°C there is a slight overestimation of humidity effects. The cooling effect of air movement is underestimated at higher levels and overestimated at the lower range.

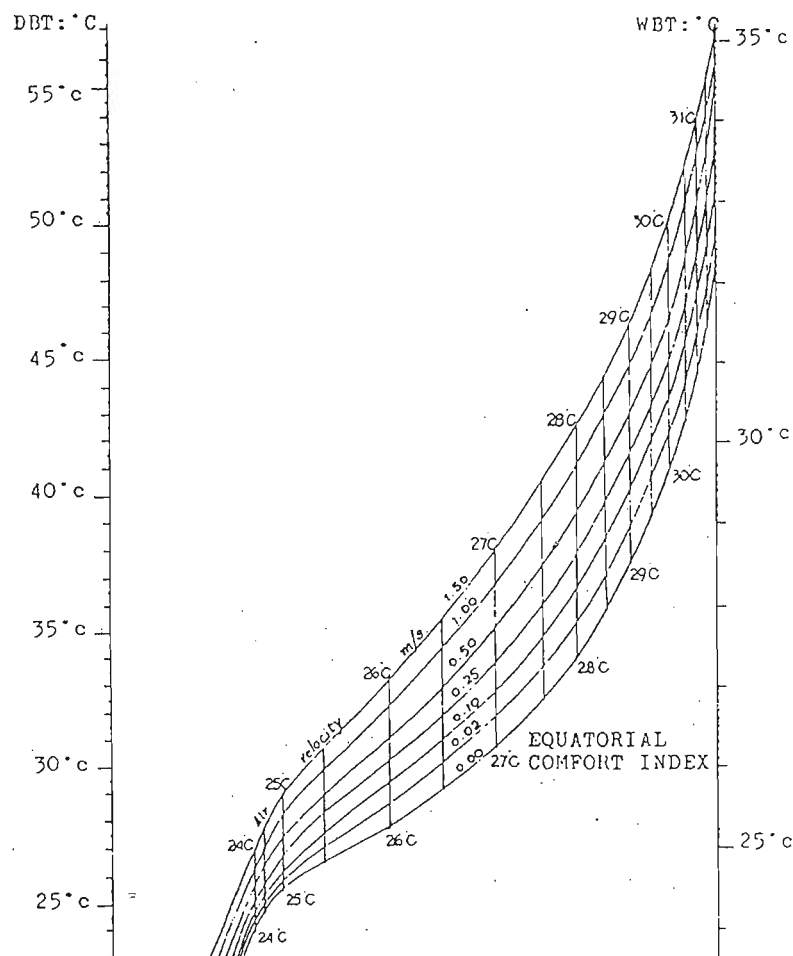
Missénard, A (1935): Teorie simplifié du thermomètre resultant. *Chauffage et Ventilation*, 12:347

Missénard, A (1959): On thermally equivalent environment. *IHVE J*, 27:231

Equatorial comfort index (ECI)

Developed by Webb, working in Malaysia and Singapore. Subjective responses of fully acclimatised subjects were recorded, together with measurements of air temperature, humidity and air movement. The relationships were organised to produce a formula and expressed in a graph rather similar to the ET nomogram.

The index is defined as the temperature of a still, saturated atmosphere which is physiologically equivalent to the climate in question.



Equatorial comfort index nomogram, by Webb

Webb, C G (1959): An analysis of some observations of thermal comfort in an equatorial climate. *Brit J of Industrial Medicine*, 16(3):297

Webb, C G (1960): Thermal discomfort in an equatorial climate. *IHVE J* 27:297

Tropical summer index (Tsi)

It is somewhat confusing that this index has the same abbreviation as the analytical Thermal Strain Index. To differentiate, we will refer to this one as Tsi (with lower case 'si').

The tropical summer index has been developed in the mid-1980s at the Central Building Research Institute, Roorkee (India), for the climatic conditions prevalent in that country and to suit the living habits of its people. It is defined as the temperature of still air, at 50% relative humidity, which causes the same thermal sensation as the given environmental conditions. Its mathematical expression is somewhat similar to that of the WBGT, but it includes the air velocity cooling effect and the empirical constants are different:

$$Tsi = 0.308WBT + 0.745GT - 2.06\sqrt{v} + 0.841$$

where WBT = wet bulb temperature (°C)

GT = globe temperature (°C)

v = air velocity (m/s)

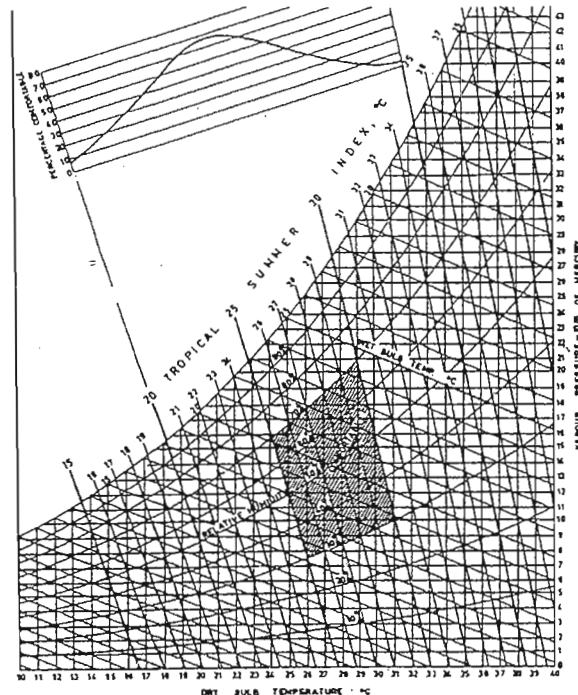
For a quick assessment this is simplified to:

$$Tsi = \frac{1}{3}WBT + \frac{3}{4}GT - 2\sqrt{v}$$

It is suggested that if GT readings are not available, GT can be taken as equal to the DBT. If (known) strong directional radiation is present, the GT value can be approximated as 1 K higher for each 90 W/m² irradiance.

The merit of Tsi lies in the fact that it is simple to compute and it is based in the local climatic and social conditions (habits, clothing, etc.) It has not been tested outside that country.

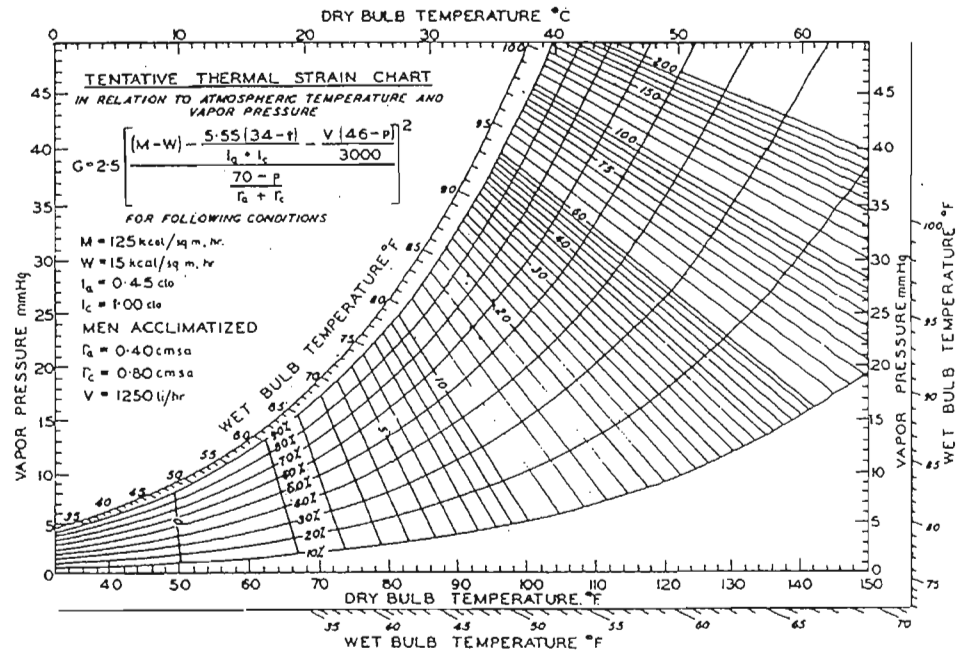
Lines of equal Tsi are drawn on a psychrometric chart for still air conditions. Tsi values can then be reduced for air velocity as shown in the table below.



Bureau of Indian Standards (1987): *Handbook of functional requirements of buildings (other than industrial buildings)* SP:41

Thermal strain index (TSI)

D H K Lee developed this index partly on the basis of observation, partly by analysing the heat transfer mechanisms. He plotted a set of *equal strain lines* on the psychrometric chart. At high levels of strain these are almost parallel with the WBT lines, whilst at low levels they are vertical, coinciding with the DBT lines.



Lee's proposed thermal strain chart (in psychrometric format)

Lee, D H K (1958): Proprioclimates of man and domestic animals. *Climatology: reviews of research*. UNESCO Conf. Paris, 1956.

Thermal Acceptance Ratio

This is the ratio of the heat acceptance potential of an environment from a nude person to the metabolic heat output of that person.

$$TAR = Ha / M$$

$$Ha = Ek(44.8-VP) + Ck(97-DBT) + Rk(97-MRT)$$

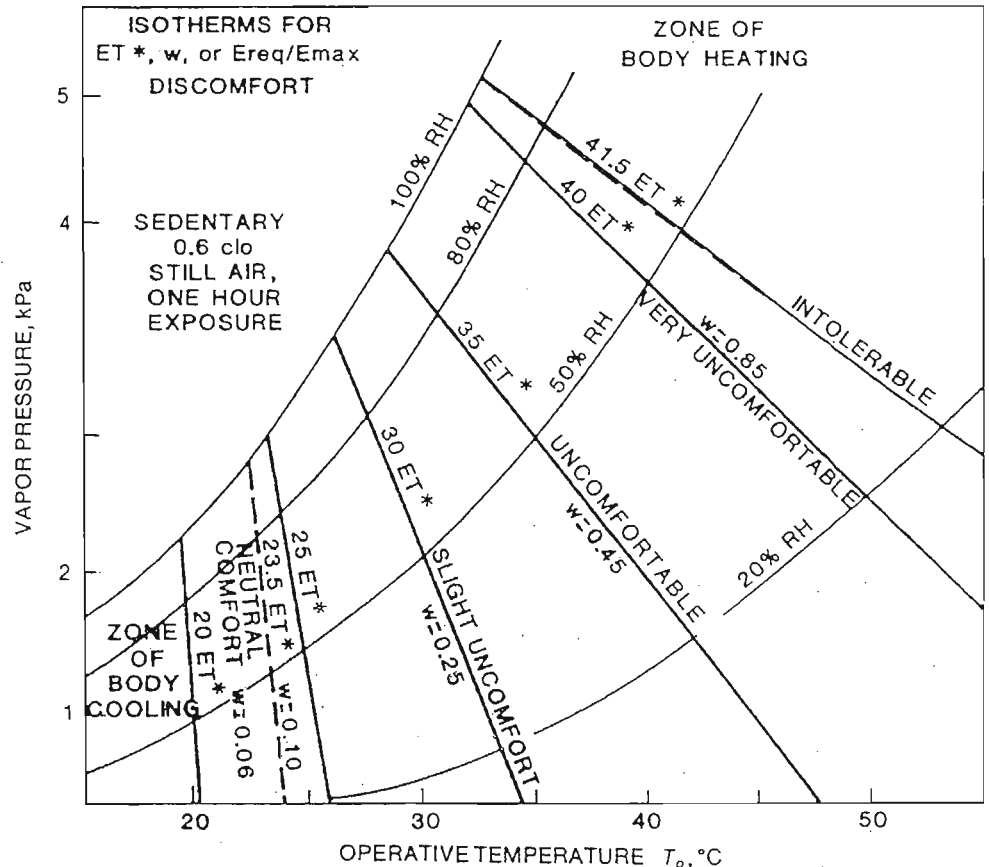
where Ek, Ck and Rk are evaporation, convection and radiation constants
VP = vapour pressure

This may be applicable to hot environments, but its main significance is that it is the precursor of the later work by Belding and Hatch.

Ionides, M, Plummer, J and Siple, P A (1945): Report from the Climatology and Environment Protection Section, Office of the Quartermaster General (US). September 17.

New effective temperature (ET*)

ET* is defined as the temperature (DBT) of a uniform enclosure at 50% relative humidity, which would produce the same net heat exchange by radiation, convection and evaporation as the environment in question. ET* lines coincide with DBT values at the 50% RH curve. Radiation is taken into account by using OT on the horizontal scale instead of DBT. The ET* lines are shown on the psychrometric chart for the following conditions: clothing: 0.6 clo, activity: 1 met, air movement = < 0.2 m/s, exposure time: 1 hour



Psychrometric chart showing constant ET* lines

The old ET lines are parallel with the 30°C ET* line and for high air movements the WBGT lines are parallel with the 35°C ET* line. The ET* lines show good correspondence with isotherms for skin wettedness, skin temperature, discomfort votes and heart rate.

Summer upper comfort limit, for subjects wearing 0.5 clo is set as 26°C ET* with 0.25 m/s air movement. This can be extended by 1 K for each 0.275 m/s increase in air velocity, up to 28°C ET* with 0.8 m/s air velocity.

Gagge, A P, Stolwijk, J A J & Nishi, Y (1971): An effective temperature scale based on a simple model of human physiological regulatory response. *ASHRAE Trans.* 77(pt.1):247-262

Gagge, A P, Gonzales, R R & Nishi, Y (1974): Physiological and physical factors governing man's thermal comfort, discomfort and heat tolerance. *Build International*, 7:305-331

Rohles, F H, Hayter, R B & Milliken, G (1975): Effective temperature (ET*) as a predictor of thermal comfort. *ASHRAE Trans.* 81(pt.2):148-156

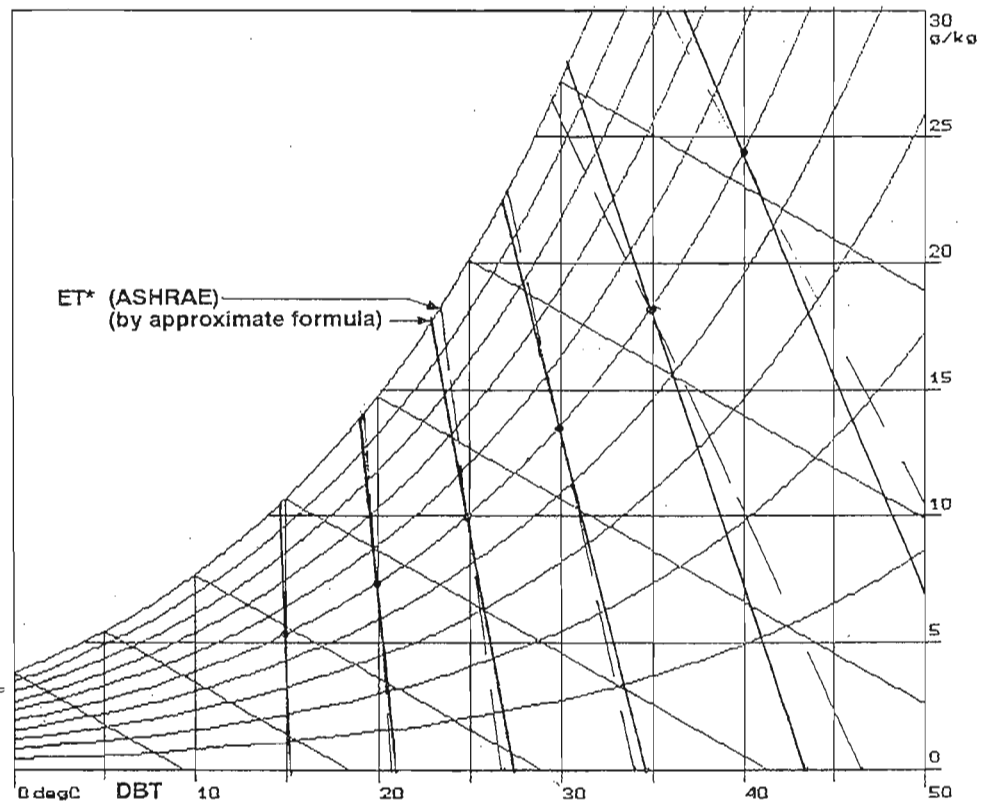
Approximation of ET*

The slope of ET* lines shows an irregularity (they are constructed locating the skin temperature points by a complicated geometric process, above the saturation curve) both in the original publication of Gagge, Stolwijk and Nishi (1971) and the diagram presented in the ASHRAE Fundamentals (shown on the preceding page). In order to facilitate the use of ET* in a computer algorithm a simple function has been produced which approximates the ET* lines.

The temperature in question (T) is plotted on the 50% RH curve and the corresponding absolute humidity is read (AH_T). The base-line intercept (DBT_b) of the ET* line will be

$$\text{DBT}_b = T + 0.025 (T - 14) \text{ AH}_T \quad \dots 1)$$

The diagram below compares the ET* lines with their approximation by this function.



The following algorithm can be used to find the AH_T value:

First the saturation vapour pressure (ps) is found for temperature T

$$ps = 0.133322 * \exp[18.686 - 4030.187 / (T + 235)] \quad \text{in kPa}$$

the corresponding saturation humidity is

$$Y_s = 622 * ps / (pt - ps) \quad \text{in g/kg}$$

where pt = 101.325 kPa, the total standard atmospheric pressure.

The AH_T used in expression (1) will be half of this value.

Standard effective temperature (SET)

Nishi and Gagge (1977) reported the further development of the ET* index and proposed the SET index, which allows for the variation of atmospheric pressure.

This is defined as " the temperature of a uniform enclosure at 50% RH, in which the mean body temperature of a sedentary subject (1.1 met), wearing 0.6 clo, in still air (< 0.15 m/s) at sea level is the same as the actual environment, described by

Pb barometric pressure
OT operative temperature
v air velocity
VP vapour pressure
clo clothing."

At sea level, under the above standard environmental conditions $SET \equiv ET^*$
At higher levels of ET* the difference between the two scales increases; with greater skin wettedness the influence of barometric pressure is increasing.

In thermal equilibrium (storage component $S = 0$) between 23°C and 41°C
SET is linearly related to average body temperature:
 $SET = 34.95 T_b - 1247.6$

The body temperature itself is calculated for a wide range of metabolic rates, clothing levels, air movement and atmospheric pressure, as well as the air temperature, mean radiant temperature and humidity catered for by the ET*.

SET gives a rational basis for measuring the equivalence for any combination of environmental factors

Nishi, Y and Gagge, A P (1977): Effective temperature scale useful for hypo- and hyperbaric environments. *Aviation, Space and Environmental Medicine*, February, p.97-107

Note on the wet-bulb globe temperature (WBGT)

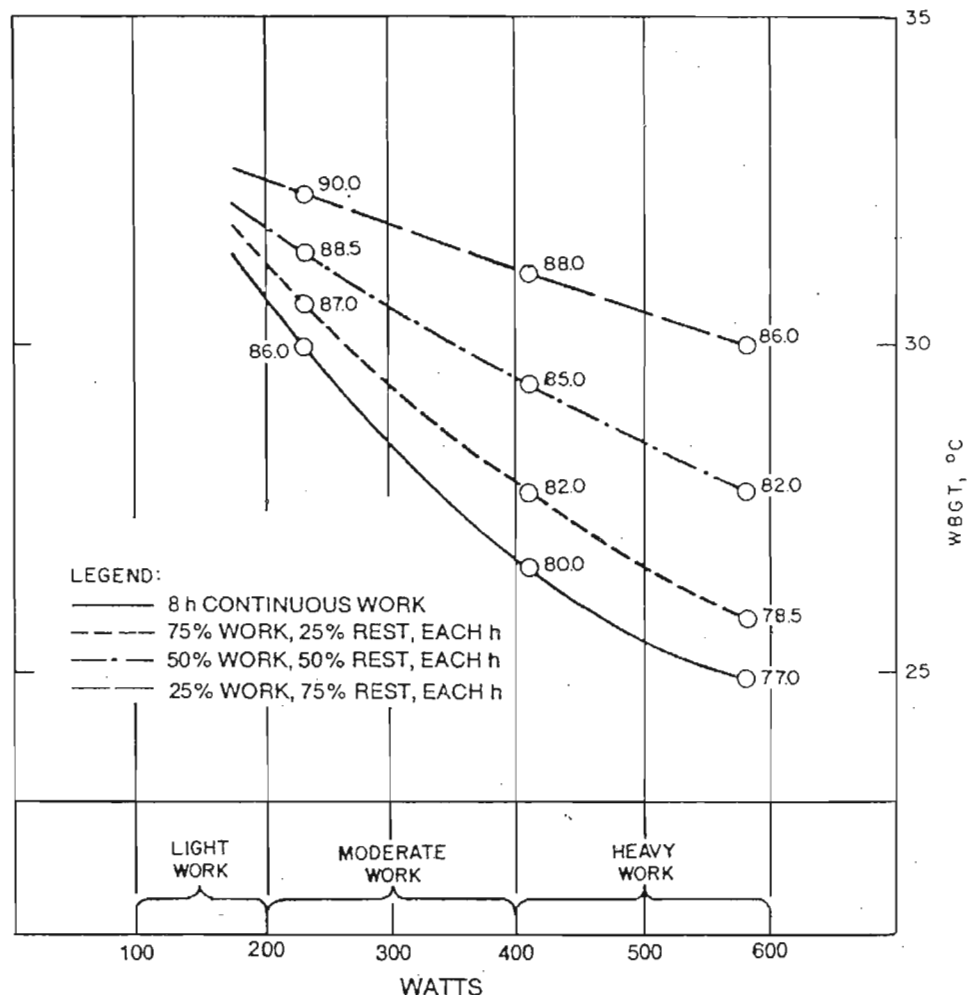
The WBGT has been developed by Yaglou and Minard for a simple field measurement of the old ET. It indicates the combined effect of air temperature, low temperature radiant heat, solar radiation and air movement. It is the weighted average of DBT, naturally ventilated WBT and globe temperature, for outdoor use (including the presence of solar radiation):

$$\text{WBGT} = 0.7 \text{ WBT} + 0.2 \text{ GT} + 0.1 \text{ DBT}$$

For indoor use the DBT term is dropped (and not the GT, as the ASHRAE Fundamentals suggests):

$$\text{WBGT} = 0.7 \text{ WBT} + 0.3 \text{ GT}$$

The relation between WBGT and permissible heat exposure limits is shown by the following graph (numbers alongside the curves are in °F):



Permissible heat exposure limits predicted by the WBGT index

Yaglou, C P & Minard, D (1957): Control of heat casualties at military centers. *AMA Archives of Industrial Health*, 16: 302

Dukes-Dobos, F & Henschel, A (1971): *The modification of the WBGT index for establishing permissible heat exposure limits in occupational work*. HEW, USPHS, ROSH, TR-69

Note on the prediction of thermal sensation

The Kansas State University research group published a set of regression equations which would predict people's thermal sensation (comfort vote) on a 7-point scale, as a function of DBT, VP (vapour pressure) and duration of exposure. The meaning of the vote, the 'Y' term is:

3	hot
2	warm
1	slightly warm
0	comfortable
-1	slightly cool
-2	cool
-3	cold

and the equations are

exposure	gender	DBT in °C	VP in kPa
1 hour	males	$Y = 0.220 \text{ DBT} + 0.233 \text{ VP} - 5.673$	
	females	$Y = 0.272 \text{ DBT} + 0.248 \text{ VP} - 7.245$	
	combined	$Y = 0.245 \text{ DBT} + 0.248 \text{ VP} - 6.475$	
2 hours	males	$Y = 0.221 \text{ DBT} + 0.270 \text{ VP} - 6.024$	
	females	$Y = 0.283 \text{ DBT} + 0.210 \text{ VP} - 7.694$	
	combined	$Y = 0.252 \text{ DBT} + 0.240 \text{ VP} - 6.859$	
3 hours	males	$Y = 0.212 \text{ DBT} + 0.293 \text{ VP} - 5.949$	
	females	$Y = 0.275 \text{ DBT} + 0.255 \text{ VP} - 8.622$	
	combined	$Y = 0.243 \text{ DBT} + 0.278 \text{ VP} - 6.802$	

(for young adult subjects at sedentary activity, wearing 0.5 clo when MRT = DBT and $v < 0.2$ m/s)

This is said to correspond to Fanger's PMV (predicted mean vote), except that this scale corresponds to PMV - 4.

Rohles, F H & Nevins, R G (1971): The nature of thermal comfort for sedentary man, *ASHRAE Trans.* 77(I):239

Rohles, F H (1973): The revised modal comfort envelope. *ASHRAE Trans.* 79(II):52

The choice of an index

A total of 20 measures and indices have been reviewed and in most cases some evaluative comments have been attached. One, that was deliberately omitted is Fanger's (1970) comfort equation, which is probably the most meticulous and detailed analysis of human thermal relationship with the proximal environment. His analytical index, the PMV (predicted mean vote) with the PPD (predicted percentage dissatisfied) form the basis of ISO 7730:1994, as well as several national standards.

However, his approach, nowadays labelled as the "constancy theory" is falling into disrepute, challenged by the "adaptation hypothesis" of Humphreys (1978), Auliciems (1981) and Nicol & Roaf (1996). Their "adaptive model" recognises that people's thermal preferences depend on the thermal conditions prevailing at their location and it varies with the seasons. They produced the following correlation functions between thermal comfort (or neutrality) and the monthly mean temperature:

$$\begin{array}{ll} T_n = 11.9 + 0.534 * T_m & \text{(Humphreys)} \\ T_n = 17.6 + 0.31 * T_m & \text{(Auliciems)} \\ T_n = 17 + 0.38 * T_m & \text{(Nicol \& Roaf)} \end{array}$$

These are neutrality temperatures for people at sedentary work, in their normal environment, wearing the clothing of their choice and are valid between 18 and 30 °C. The comfort limits are then taken as $T_n \pm 2^\circ\text{C}$.

Of these the expression of Auliciems is likely to be most relevant, as his sample of some half a million votes included a significant Australian contingent.

In a recent paper Williamson et al. (1995) showed that the PMV strongly overestimates warm discomfort, especially in warm climates. Karyono (1996) also found (in Indonesia) that people in warm-humid climates prefer up to 6 K higher temperatures than those predicted by ISO 7730. Humphreys and Nicol (1996) conclusively showed the errors inherent in Fanger's approach.

Much earlier Macpherson (1962) suggested that there are many factors not recognised by the various indices *the most important of these is acclimatisation*. The PMV approach denies the role of acclimatisation.

Consequently it is not thought to be applicable for the purposes of NatHERS.

It should be noted that all these workers used the DBT as an index of thermal comfort or neutrality. This trend can be traced back to Drysdale (1950), who demonstrated that at or near comfort level the best measure of thermal conditions is the dry bulb temperature. Macpherson (1962) agreed: *The simpler the index chosen, the more likely it is to prove satisfactory and the simplest index of all is the DBT*; and further: *under ordinary conditions in still air the DBT in itself is a better index of warmth than is effective temperature and any other composite index*.

It can be concluded that DBT is the most useful measure for the specification of comfort, but for the measurement of the magnitude of discomfort some other measure must be found, recognising the other environmental factors: humidity, radiation and air movement.

Comparison of some thermal comfort indices
for 0.5 clo, DBT = GT, v < 0.2 m/s

DBT	16				20				24				28				32°C			
	wbgt	ET	RT	ET*	RSI	wbgt	ET	RT	ET*	RSI	wbgt	ET	RT	ET*	RSI	wbgt	ET	RT	ET*	RSI
RH																				
	10.4					13.7					16.9					19.9				
		14.4					17.2					20					22.8			
	30%		-					-					22.2				25.2		28	
				15.9					19.8					23.6			26.8			30.2
				-					-					0.028			0.112			0.21
45%	11.8					15.1					18.8					22			25.4	
		14.8					17.8					20.9					23.8		26.6	
			-					-					22.6				25.8		28.7	
				16					20					23.8			27.7			31.6
				-					-					0.029			0.128			0.251
60%	13.1					16.6					20.2					23.8			27.6	
		15					18.5					21.6					24.8		28	
			-					-					23				26.4		29.5	
				16.1					20.2					24.4			28.7			33.1
				-					-					0.034			0.149			0.314
75%	14.2					17.9					21.8					25.5			29.3	
		15.4					19					22.5					26		29.5	
			-					-					23.2				27		30.5	
				16.2					20.6					25			29.8			34.7
				-					-					0.039			0.178			0.424
90%	15.3					19.3					23.2					27.2			31.1	
		15.9					19.5					23.4					27.2		31	
			-					-					23.5				27.5		31.4	
				16.3					21.1					25.8			31			36.2
				-					-					0.046			0.225			0.669

wbgt = wet bulb globe temperature
ET = effective temperature
RT = resultant temperature
ET* = new effective temperature
RSI = relative strain index

The RT nomogram does not extend below 20°C

The RSI expression below 20°C gives a negative value

A scan of the indexes reviewed shows the following results:

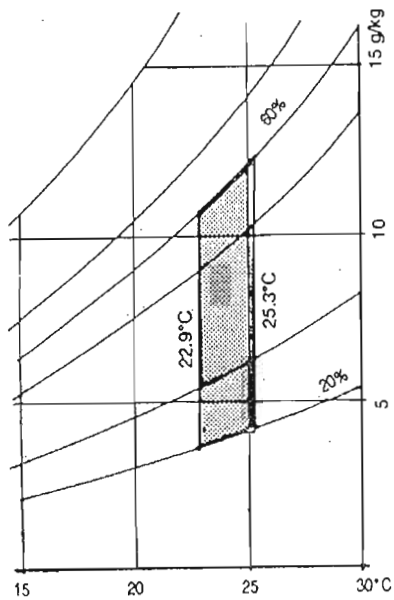
ET	p.2	Superseded by ET*
CET	p.3	Superseded by ET*
OT	p.4	Does not consider humidity, judged as unsuitable above 27°C
EqT	p.5	Does not consider humidity, unsuitable above 24°C
EqW	p.5	Not suited to high temperatures, underrates air movement cooling
RT	p.6	Underrates air movement cooling at high temperatures
ECI	p.7	An improvement on ET, but does not recognise radiation effects
Tsi	p.8	
TSI	p.9	A predecessor of RSI, its use is now discontinued
P4SR	p.10	Reliable for high temperatures but not below 28°C
HSI	p.11	Later developed into RSI
RSI	p.12	A very significant theoretical development, Hounam (1969) used it; calculating the frequency of occurrence of RSI 0.2 and 0.3 values and plotting these contours on maps of Australia. However, its use is more appropriate in research than in practice.
ITS	p.13	An index far too complicated for practical use
ET*	p.14	It embodies the state-of-the-art knowledge, widely recognised and used. With the proposed approximation it would be easy to use
SET	p.16	Probably the best index, applicable to the widest range of conditions, but it is complicated and for the limited set of conditions encountered in NatHERS for the tropics its use is not warranted
WBGT	p.17	Attractive for its simplicity, but it has been designed principally for strenuous activity (military) or outdoor manual labour

Such indices may be employed for several different purposes.

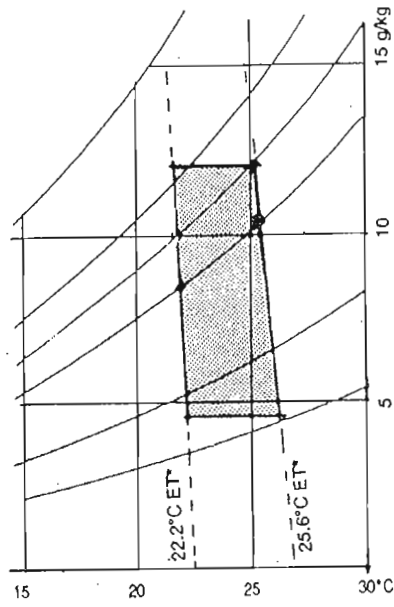
1. Setting exposure limits or "thresholds". As Lee suggests, these may be (a) limits not to be exceeded or (b) precautionary limits. For these purposes the WBGT is quite adequate.
2. Defining comfort, the limits of comfort, i.e. the "comfort zone", which is applicable to residential or office situations. Lee (1980) considers that the modified ET index is the most appropriate for this purpose, provided that allowance is made for acclimatisation.
3. Evaluating past exposures, e.g. for the purposes of compensation (even court cases), which would require a method more sensitive than the WBGT, probably one of the stress/strain indices.
4. Determining the optimum control measures (e.g. choice between air movement or air conditioning; screening against radiant heat (in industry) or reducing the exposure period, etc.), for this the individual contributing variables must be examined.
5. Climate classification; zones determined by individual variables (e.g. DBT or RH) are not very useful, use of one of the stress/strain indices is more meaningful (e.g. Venville, 1959 or Hounam, 1969)

The concern of NatHERS is probably item (2) above: the definition of comfort limits (comfort zone). For this reason the use of the ET* index is recommended.

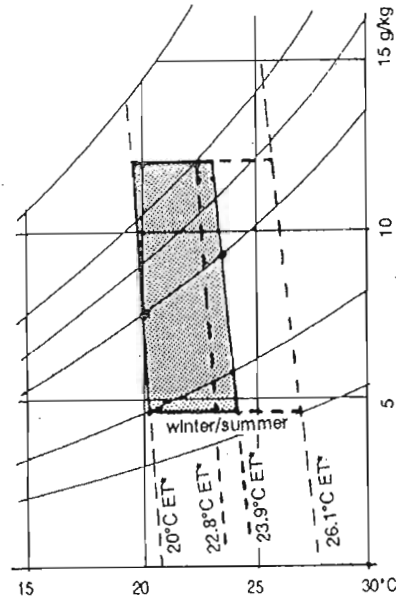
The diagram overpage shows the development of the ASHRAE comfort zone definition over the last 30 years – or so. ET* was introduced for temperature limits in 1974 and the previous definition of humidity limits in terms of RH was replaced by vapour pressure (or AH). In 1981, for the first time, a distinction was made between summer and winter comfort. (Subsequent modifications relate to the upper humidity limit only, are still controversial and the subject of several research projects, especially relating to warm humid climates). Because of this, it is the 1981 version we propose to build on.



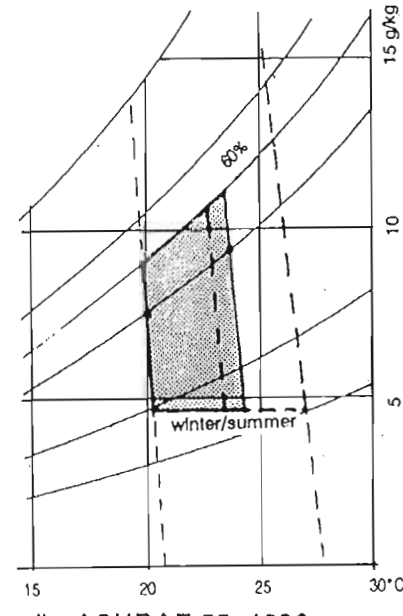
a) ASHRAE 55-1966



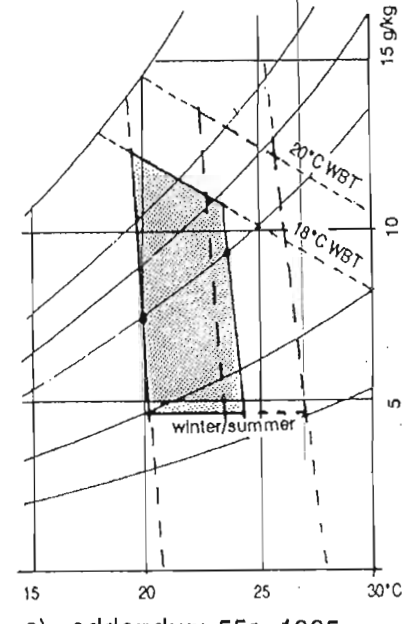
b) ASHRAE 55-1974



c) ASHRAE 55-1981



d) ASHRAE 55-1992



e) addendum 55a-1995

- a) up to 1966 the temperature boundaries are defined by DBT lines and the humidity limits by RH curves
- b) ET* lines define the temperature boundaries and vapour pressure (or AH) set the humidity limits
- c) a distinction is made between summer and winter comfort
- d) temperature and lower humidity limits remain, but it reverts to the 60% RH line for upper humidity limit
- e) the upper humidity limits are set as 18°C WBT for winter and 20°C WBT for summer

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Cooling effect of air movement

The cooling effect of air movement, thus the upward extension of the comfort zone depends on clothing and activity levels, but also on other environmental conditions. The following approximations are valid for the conditions stated, which are likely to occur in warm climates, under overheated conditions.

Olgay's (1963) original bioclimatic chart (his Fig.41, p.20) includes the extensions of the comfort zone by air movement (at 50% RH) as shown by curve **O** of the graph overpage.

The ASHRAE thermal comfort standard 55-1992 (its Fig.3, p.9) allows the extension of the comfort zone resulting from up to 1.5 m/s air velocity, as shown by curve **A** of the graph overpage, for a situation where MRT = DBT, with sedentary activity (1.2 met) and light clothing (0.5 clo). This is increased to curve **B** if MRT = DBT + 10 K.

The ASHRAE handbook of Fundamentals (1985, p.8.19) permits the extension of the comfort zone, allowing for the cooling effect of air movement by 1 K for each 0.275 m/s (above 0.25 m/s), up to 0.8 m/s. This is shown by line **C** of the graph.

Givoni (1994, p.40) contends that this is far too restrictive and suggests that the limit should be extended to 2 m/s. McIntyre (1976) also suggests that the maximum acceptable velocity generated by ceiling fans is 2 m/s. Accordingly, line **C** is extrapolated to 2 m/s. Hunt et al. (1976) found that outdoors 6 m/s is acceptable, but beyond 4 m/s the incremental benefits are barely noticeable.

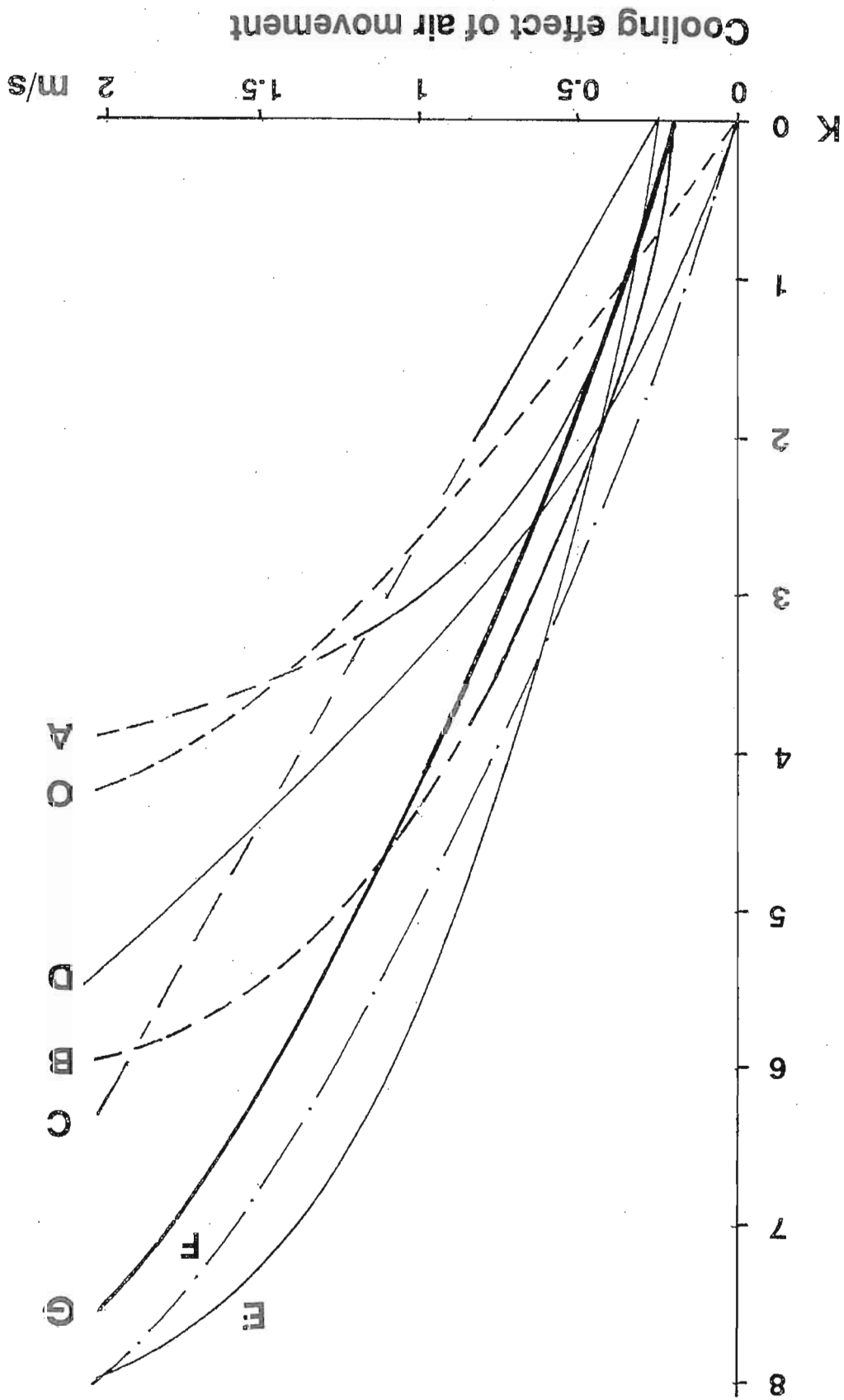
The cooling effect of air movement according to Bedford's effective temperature nomogram (basic scale, at 25°C and 50% RH) is shown by plot **D** of the same graph.

Arens et al. (1980), on the basis of the J.B. Pierce laboratory's two-node mathematical model (originally by Gagge et al, 1970) in their new bioclimatic chart suggest the extensions of the comfort zone shown by curve **E** of the graph, (at the 12 g/kg humidity level, which is the humidity limit, with 1.3 met activity rate and 0.8 clo clothing). The cooling effect would be higher with lower humidities and with 0.5 clo, which is normal in warm climates.

Curve **F** has been suggested as a numerical approximation of the above by the simple function : $dT = 6 \cdot v - v^2$
but to allow for no cooling effect below 0.2 m/s and to move towards the ASHRAE handbook recommendations this has been modified to curve **G** :

$$dT = 6 (v-0.2) - (v-0.2)^2$$

This can readily be built into any computer algorithm.



Numerical data for the curves plotted:

A standard (MRT=DBT)	0.2 0	0.35 1.1		0.65 2.2	1.2 3.3			m/s K
B standard (MRT=DBT+10)	0.2 0	0.29 1.1	0.45 2.2	0.7 3.3	1 4.4			m/s K
C Fundamentals	0.25 0	0.525 1	0.825 2	1.1 3	1.375 4	1.65 5	1.925 6	m/s K
	0.25	0.5	1		1.5		2	m/s
O Olgyay		1.4	2.65		3.65		4.2	K
D Bedford	1.2	2.15	3.4		4.4		5.4	
E Arens	0	2.55	5.6		7.3		8	
F $6v-v^2$		2.75	5		6.75		8	
G $6(v-0.2)-(v-0.2)^2$		1.71	4.16		6.11		7.56	

References for the cooling effect

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Recommendations

1. The ET^* be accepted as a measure of comfort
2. The 1981 ASHRAE comfort zone definition be taken as the starting point, but – accepting Lee's advice to allow for acclimatisation – we recommend the adoption of Auliciems' neutrality expression

$$T_n = 17.6 + 0.31 T_m \quad (\text{but } 18^\circ\text{C} < T_n < 30^\circ\text{C})$$

locate this on the 50% RH curve and take the comfort zone as 4 K wide, i.e. from $T_n - 2$ to $T_n + 2^\circ\text{C}$. The side boundaries will be the corresponding ET^* lines, as defined on p.15. This procedure is to be carried out for the coolest and the warmest month.

3. The degree of overheating be measured above the upper limit of the warmest month and underheating below the lower limit of the coolest month.
4. Where air movement can be guaranteed to be present during overheated periods, the upper comfort limit can be extended according to the expression proposed above (p.24), by

$$dT = 6 \cdot (v - 0.2) - (v - 0.2)^2 \quad (\text{up to } v = 2 \text{ m/s})$$

measured at the 12 g/kg level.

1.6 x

Auliciems &
See Szokolay 1997

PLEA - Thermal Comfort

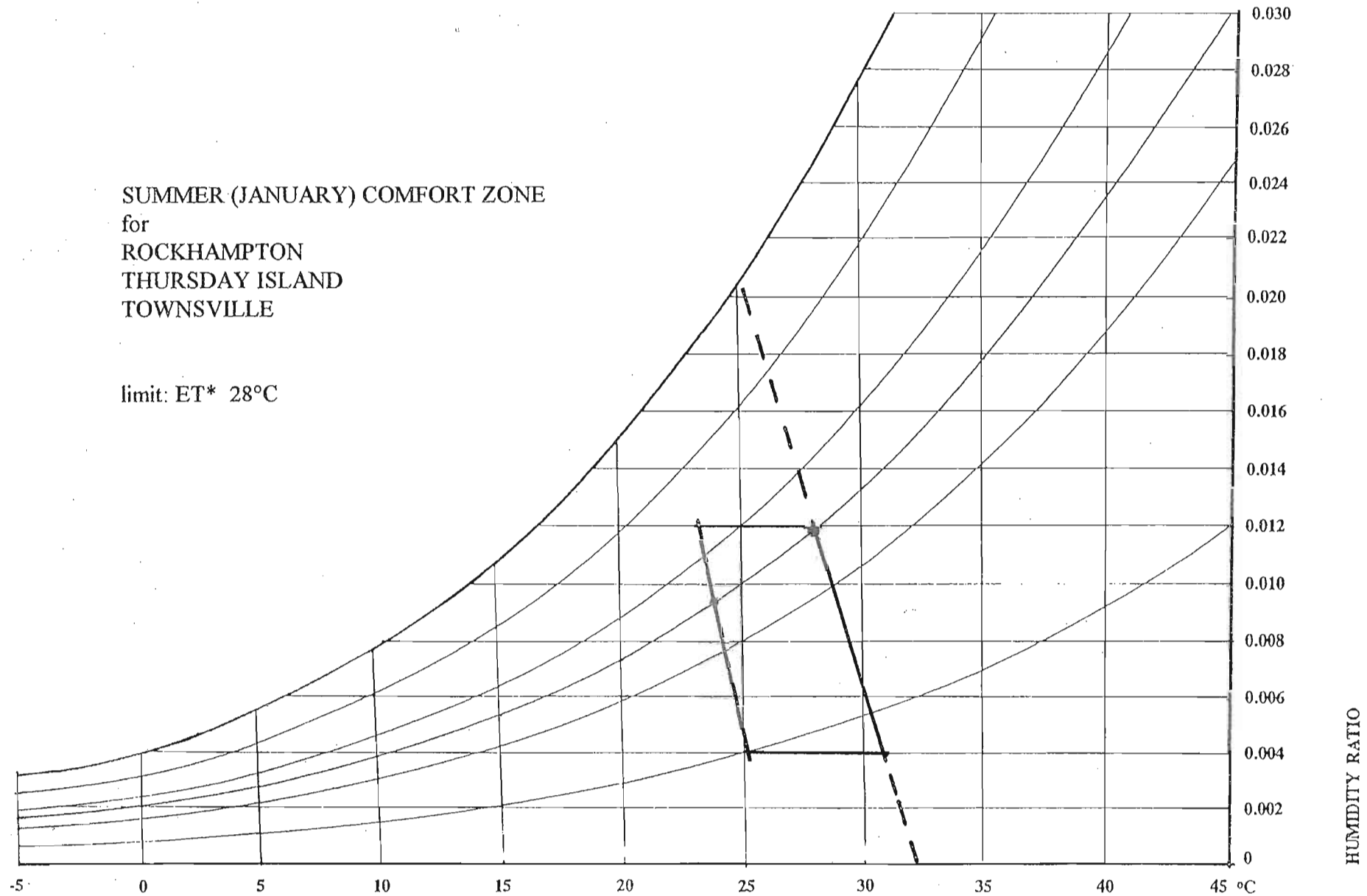
PROPOSED THERMAL COMFORT LIMITS

DBT°C @ 50%RH (ET*) Rounded to nearest 0.5°C

Location (alphabetical)	January			July		
	T _O av.	T _n	Comf. Limits	T _O av.	T _n	Comf. Limits
Adelaide	22.8	24.5	22.5 - 26.5	11.3	21.0	19.0 - 23.0
Alice Springs	29.4	26.5	24.5 - 28.5	11.9	21.5	19.5 - 23.5
Amberley	25.2	25.5	23.5 - 27.5	13.1	21.5	19.5 - 23.5
Brisbane	24.9	25.5	23.5 - 27.5	15.1	22.5	20.5 - 24.5
Canberra	20.1	24.0	22.0 - 26.0	5.2	19.0	17.0 - 21.0
Carnarvon	26.7	26.0	24.0 - 28.0	16.5	22.5	20.5 - 24.5
Cloncurry	31.4	27.5	25.5 - 29.5	18.0	23.0	21.0 - 25.0
Coffs Harbour	22.8	24.5	22.5 - 26.5	12.5	21.5	19.5 - 23.5
Darwin	28.3	26.5	24.5 - 28.5	24.6	25.0	23.0 - 27.0
Geraldton	25.1	25.5	23.5 - 27.5	14.3	22.0	20.0 - 24.0
Hobart	16.9	23.0	21.0 - 25.0	8.0	20.0	18.0 - 22.0
Launceston	16.8	23.0	21.0 - 25.0	6.5	19.5	17.5 - 21.5
Longreach	31.4	27.5	25.5 - 29.5	15.6	22.5	20.5 - 24.5
Melbourne	20.7	24.0	22.0 - 26.0	9.8	20.5	18.5 - 22.5
Mildura	24.5	25.0	23.0 - 27.0	9.8	20.5	18.5 - 22.5
Moree	26.2	25.5	23.5 - 27.5	10.9	21.0	19.0 - 23.0
Nowra	20.6	24.0	22.0 - 26.0	10.9	21.0	19.0 - 23.0
Perth	24.5	25.0	23.0 - 27.0	13.4	22.0	20.0 - 24.0
Port Hedland	30.8	27.0	25.0 - 29.0	19.4	23.5	21.5 - 25.5
Richmond	23.2	25.0	23.0 - 27.0	10.4	21.0	19.0 - 23.0
Rockhampton	26.6	26.0	24.0 - 28.0	15.8	22.5	20.5 - 24.5
Sale	19.1	23.5	21.5 - 25.5	8.4	20.0	18.0 - 22.0
Sydney	22.2	24.5	22.5 - 26.5	12.7	21.5	19.5 - 23.5
Tamworth	24.0	25.0	23.0 - 27.0	9.1	20.5	18.5 - 22.5
Townsville	27.6	26.0	24.0 - 28.0	19.0	23.5	21.5 - 25.5
Wagga	23.6	25.0	23.0 - 27.0	7.5	20.0	18.0 - 22.0
Williamstown	22.5	24.5	22.5 - 26.5	11.4	21.0	19.0 - 23.0

SUMMER (JANUARY) COMFORT ZONE
for
ROCKHAMPTON
THURSDAY ISLAND
TOWNSVILLE

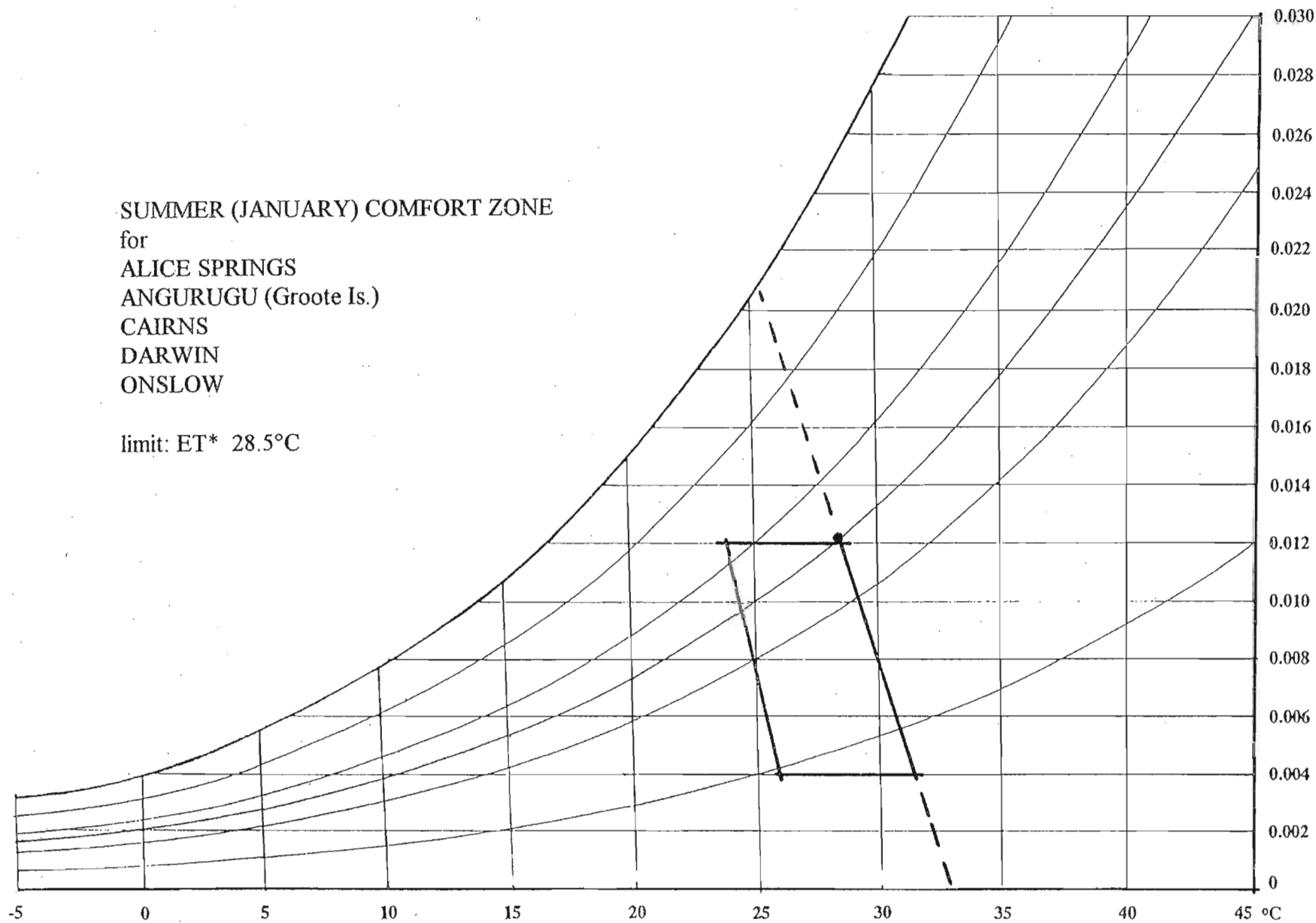
limit: ET^* 28°C



SUMMER (JANUARY) COMFORT ZONE
for
ALICE SPRINGS
ANGURUGU (Groote Is.)
CAIRNS
DARWIN
ONSLOW

limit: ET^* 28.5°C

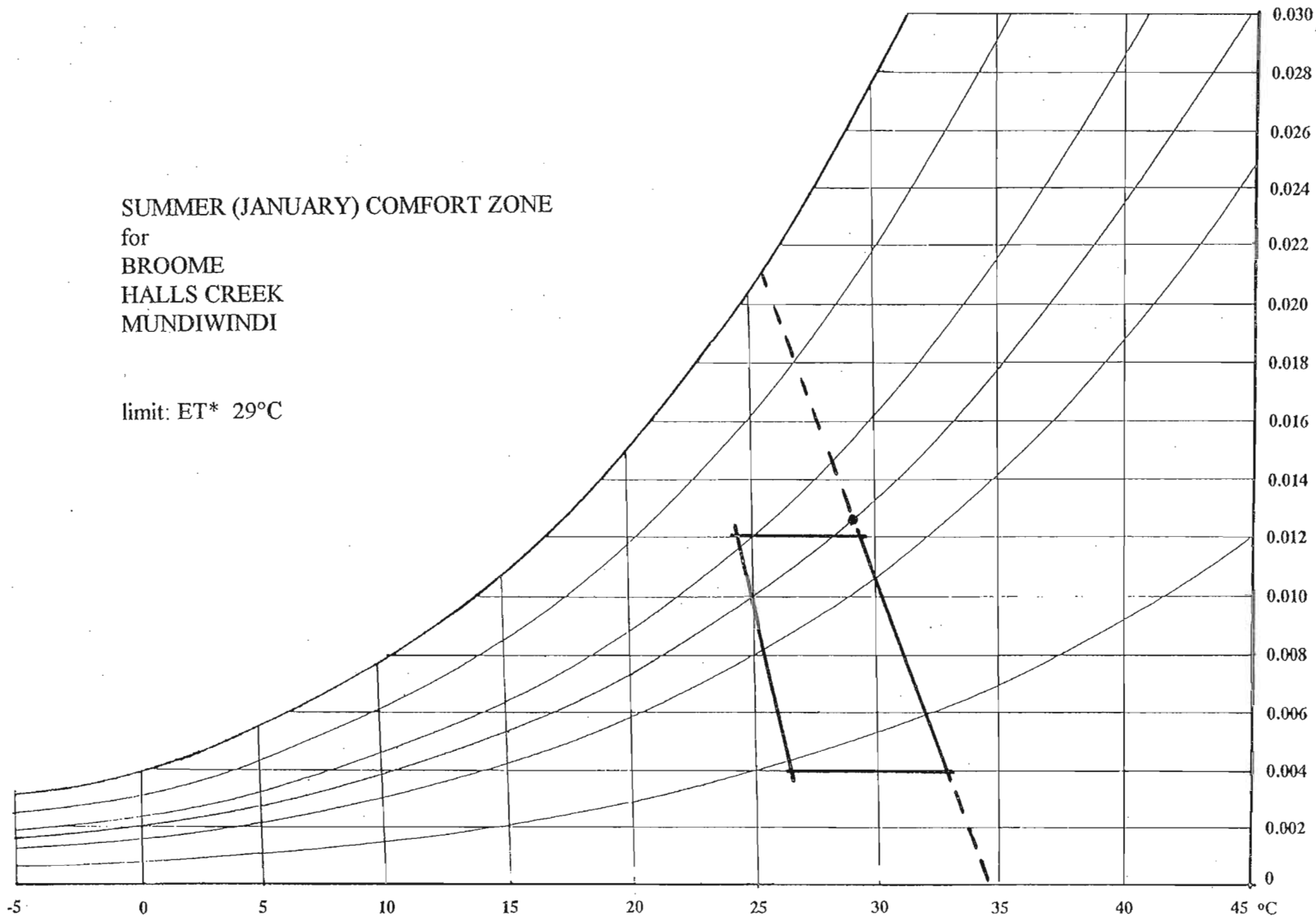
30



HUMIDITY RATIO

SUMMER (JANUARY) COMFORT ZONE
for
BROOME
HALLS CREEK
MUNDIWINDI

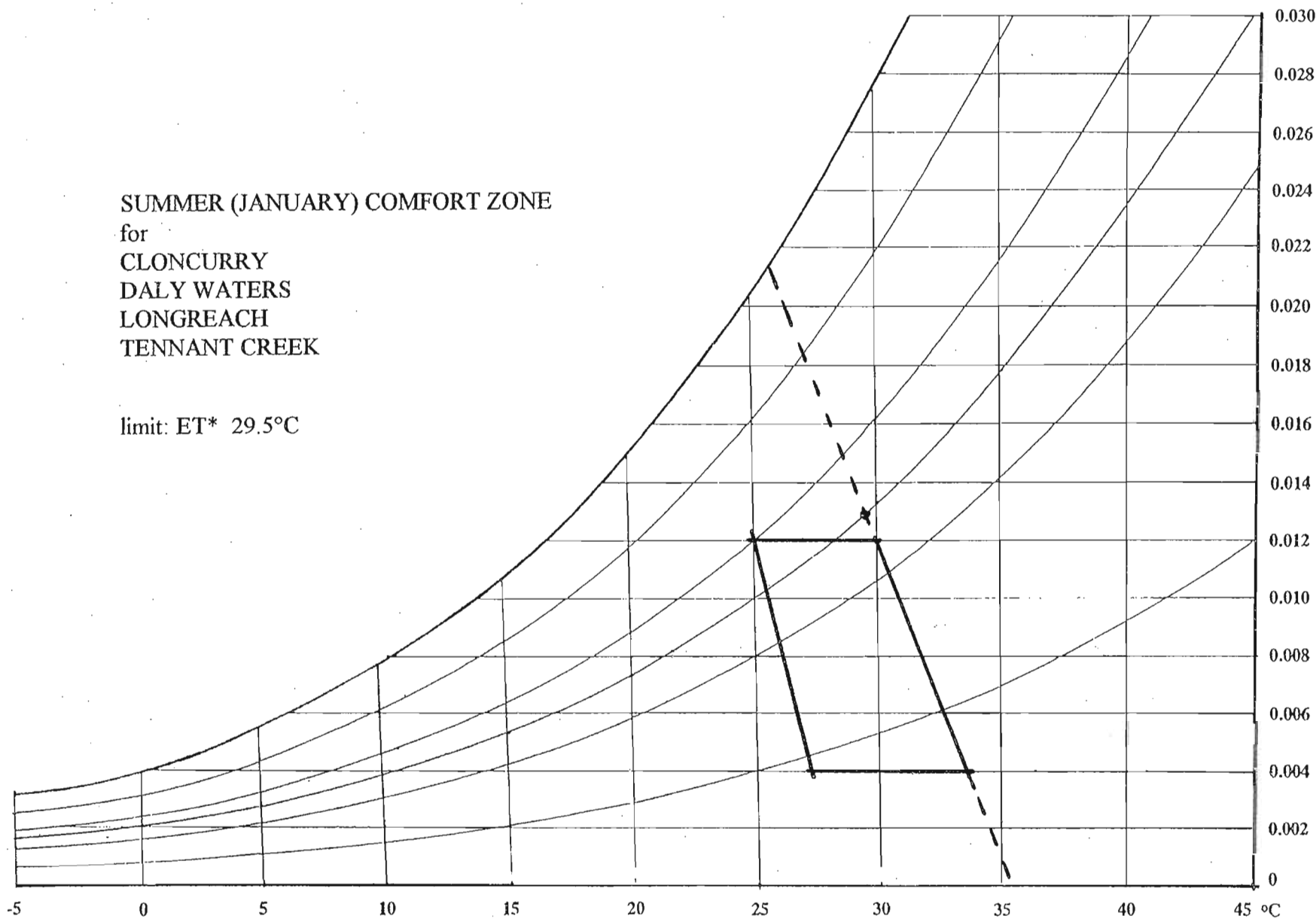
limit: ET^* 29°C



HUMIDITY RATIO

SUMMER (JANUARY) COMFORT ZONE
for
CLONCURRY
DALY WATERS
LONGREACH
TENNANT CREEK

limit: ET^* 29.5°C



HUMIDITY RATIO

PRINCIPAL INFLUENCES OF NATURAL VENTILATION ON NatHERS

The principal influences of summer natural ventilation in free running buildings are:

1. Removal of heated indoor air
2. Air cooling of the building structure
3. Cooling of occupants by air movement

The first two of these influences can already be accommodated in a fashion via air change rates within the NatHERS software. It should be noted however that some of the most debatable issues in heat transfer in buildings are involved, namely surface heat transfer coefficients and local indoor and outdoor air flow patterns and local velocities adjacent to building surfaces.

External Convective Heat Transfer Coefficients

Recommended design values in ISO/DIS 13791 for external convective heat transfer coefficients purposes of estimating heat flow in free running buildings are determined by the equation:

$$h_{ce} = 4 + 4v \quad \text{W/m}^2\text{K}$$

where:

h_{ce} is the convective heat transfer coefficient of an external surface
 v is the wind velocity in m/s near the surface

It is important to note that no consideration is given to roughness of the surface. Surface roughness is a key factor in determining turbulence in a boundary layer. This turbulence is responsible for mass transfer across the airflow boundary layer. Mass transfer due to turbulence is the mechanism for convective heat transfer across the boundary layer. Indoor summer air movement is often less than 0.3 m/s yet is still quite beneficial. With smooth wall, floor and ceiling surfaces and these low air speeds, Reynolds Numbers near boundaries will be low, <2000, and boundary layers will tend to be laminar. This explains the offsets of around 0.3 m/s in many of the functions relating air flow cooling effects on occupants.

No indication is given of the range of temperature differences between air and the surface to which the equation applies although reference surface temperature tends to be around 300K. This is reasonable for high thermal capacity materials as is common in North America and Europe but may not be so for low heat capacity materials such as sheet metal cladding which is common in Australia, particularly on roofs.

Internal Convective Heat Transfer Coefficients

Internal convective heat transfer coefficients are a function of surface orientation and convective air movement and temperature difference between the air and the surface. European data usually assumes $\Delta T < 10\text{K}$. Recommended design values in ISO/DIS 13791 for purposes of estimating heat flow due to air mass transfer due to infiltration or natural airflow through large openings are:

Vertical surface (Resistance 0.4 m ² K/W)	2.5 W/m ² K
Horizontal surface with buoyant flow (Resistance 0.2 m ² K/W)	5.0 W/m ² K
Horizontal surface stagnant air (Resistance 1.4 m ² K/W)	0.7 W/m ² K

It is important to note that the research supporting these design values was performed in Europe and North America. This may be a reason why there is no consideration of the influence of surface roughness or consideration given to significant indoor air movement due to the dominant concern for winter discomfort from draughts.

Equivalent indoor surface air film resistance suggested in the AIRAH Handbook (page 6.08) is 0.08 m²K/W. This allows for 0.5 m/s air movement and for any surface orientation and heat flow in any direction. Still air surface resistances for high emittance surfaces range from 0.11 m²K/W for upward heat flows to 0.16 m²K/W for downward heat flows for horizontal surfaces and 0.12 m²K/W for vertical surfaces with heat flow horizontal. These are substantial differences.

External Long-Wave Heat Transfer Coefficients

Recommended design values in ISO/DIS 13791 for long-wave heat transfer at external surfaces of buildings are determined using the equation:

$$q_e = \varepsilon \sigma (F_{sk} T_{sk}^4 + F_b T_b^4 + F_g T_g^4 - T_{es}^4)$$

where:

- q_e is the external long-wave heat transfer coefficient
- F_{sk} is the view factor with the sky
- F_b is the view factor with other buildings
- F_g is the view factor with the ground
- T_{sk} is the absolute temperature of the sky
(daytime & cloudy nights >16°C, clear nights <16°C
ASHRAE Fundamentals 1993)
- T_b is the absolute temperature of other buildings
- T_g is the absolute temperature of the ground
- T_{es} is the absolute temperature of the outside surface of the wall
- ε is the emissivity of the surface
- σ is the Stefan Boltzman constant, 5.6697×10^{-8} W/m²K⁴
- T_{ea} is the temperature of external air

Suggested Design Values for View Factors

	Vertical Surface			Horizontal Surface		
	F_{sk}	F_b	F_g	F_{sk}	F_b	F_g
City Centre	0.33	0.34	0.33	1.00	0.00	0.00
Suburban Area	0.41	0.18	0.41	1.00	0.00	0.00
Rural Area	0.45	0.10	0.45	1.00	0.00	0.00

Where emissivities of building surfaces are greater than 0.8 (most non-metallic surfaces) an external long-wave heat transfer coefficient design value of 5.5 W/m²K is suggested in ISO/DIS 13791.

Difficulties in obtaining reliable detailed data on the local airflow near building surfaces and solid angles and surface temperatures (including that for the sky) associated with radiation exchanges in particular

situations, demonstrate the complexities of estimating surface heat transfer coefficients and subsequently the thermal performance of buildings and thermal comfort, particularly in free running buildings.

ESTIMATING NATURAL INDOOR AIRFLOW

Natural ventilation is often the design strategy used by house designers chosen to deal with indoor summer comfort (de Dear et al, 1991), (Hoyano et al, 1996). While research into methods for estimating has been carried out around the world for decades, the most impressive effort has been the multi-million dollar programs funded by the International Energy Agency (IEA) coordinated through the Air Infiltration and Ventilation Centre in Coventry, UK. Australia is not a contributor to this program which makes access to data from these programs more expensive for individual Australian researchers.

Energy Sources For Natural Ventilation

An important element of the IEA research program was development of the *Cp Generator* software (Knoll et al., 1996) by researchers at TNO Building and Construction Research facility in Delft, the Netherlands. A previous study (Swami & Chandra, 1987) funded by ASHRAE fitted a curve to the average wind pressure coefficient data for walls of rectangular buildings and so could not provide correction for the influence of nearby buildings. *Cp Generator* is simple to use requiring no special expertise with input data being coordinates and orientations of both the building and surrounding obstacles. Wind pressure over building surfaces is the principal energy source for natural ventilation. Wind pressure distribution is a function of building shape, orientation and nearby obstacles and is fundamental to estimating natural ventilation. The other source of energy for natural ventilation is the stack effect which is a function of the difference in density of indoor and outdoor air due to temperature differences. Heating and ventilating handbooks (ASHRAE, 1993), (CIBSE, 1986) include procedures for estimating airflows due to stack effect.

Methods for Estimating Natural Ventilation

Again the most impressive efforts in developing methods for estimating natural ventilation of buildings in the last decade has been those by the IEA. These efforts culminated in the development of the computer software PASSPORT-Air based on evaluation of seven existing global airflow and velocity coefficient methods for estimating single side and cross natural ventilation. The methods were evaluated in single and two zone conditions in full scale buildings in Greece and Spain using tracer gas decay techniques. The methods evaluated were:

Global Airflow Prediction

Aynsley (1977)
Vickery (1987)
Aynsley (1988)
Murakami (1991)

Velocity Coefficient Methods

Gouin (1984)
Givonni (1988)
Ernest (1991)

All methods had limitations which were not easy to describe or were aimed at particular cases and IEA efforts were directed to developing a model capable of general application. The model developed was a network flow model based on electrical circuit resistance analogy (Aynsley, 1988), (Walton, 1989), (Roulet, 1996).

Corrections Resulting from Full Scale Measurements

As a direct result of data from more than 100 full scale tests of *single-sided* and cross ventilation in Athens, Madrid and Lyon a correction factor, CF, was developed to take better account of Reynolds Number, Re , for the depth, D , of the cavity and Grashof Number, Gr , for the height of an opening influences on flow. Reynolds Number is the ratio of inertia force to the viscous force at a point in a fluid flow. Grashof Number is the ratio of buoyancy force to viscous force in free convection flows. Another change was made to the ventilation model used in PASSPORT-Air software to include a term for wind turbulence and for cases where there are significant temperature differences involved.

Correlation coefficient of estimated airflow with experimental observations, without correction factor was 0.4 and 0.75 with the correction factor incorporated.

Computer Program Comparisons and Limitations

With a generalised model developed and PASSPORT-Air computer software operational, comparisons were made with other airflow network computer software packages: ESP, AIRNET, COMIS, NORMA, and BREEZE. Correlation coefficients with output from these programs ranged from 0.93 to 1.00, which is quite good considering the complexities involved.

Before thinking that estimation of natural airflow is now a non-problem, it is important to recognise the limitations of such programs. Convergence often has to be forced in the solution of the numerous simultaneous equations describing an airflow network (Walton, 1989). The current explanation for this difficulty often suggested is that a steady flow is not a normal state in natural ventilation which is usually characterised as a low frequency pulsing flow through large openings. While resistance analogy network computer programs can reliably predict airflows between the interior and exterior given reliable pressure data, and airflow between internal zones, they are not capable of determining indoor airflow patterns or local velocities away from openings.

The IEA is currently pursuing the use of computational fluid dynamics (CFD) programs to provide the detailed internal flow information not available from PASSPORT-Air. Better known CFD software packages include such as PHOENICS, VORTEX-2, CFD-2000, CFD-ACE, FLUENT, FIDAP, CFX, and STAR-CD. Both resistance network and CFD software require substantial computing power and time for accurate results and convective effects near boundary surfaces still require development. It should be noted that CFD software using the common K- ϵ turbulence model are not capable of accurately modelling external flows around buildings (Cochran, 1997). Acceptable modelling of external flows around buildings can be achieved using large eddy simulation (LES) software but this requires approximately three times the computation involved in CFD K- ϵ programs and is only feasible on super computers (Murakami, 1990).

Natural Ventilation Estimation Methods

Suited for use in NatHERS Software

While they are currently state-of-the-art, given the heavy computation requirements and significant data input, programs such as PASSPORT-Air or COMIS etc and CFD programs are probably not suitable for use in NatHERS. However, adoption of anything less than state-of-the-art is likely to attract criticism. What is needed for NatHERS is a *reputable* method with minimal input data requirements, preferably making maximum use of building data already used in NatHERS. There are a number of methods described in heating and ventilating handbooks for estimating indoor airflow and at least two methods for estimating average wind pressure coefficients on walls. Fully integrated methods are less common.

In July 1995 SAA Committee BD/58/2 reviewed an ISO standard which included a recommended method for estimating ventilation in free running buildings. This ISO standard was reviewed as SAA technical committee TC 156 could not provide a better proposal. The standard reviewed was:

ISO/DIS 13791 Thermal performance of Buildings - Internal temperatures in summer of a room without mechanical cooling - general criteria and calculation procedures.

This document contains a comprehensive method for estimating natural ventilation due to wind and stack effects with reasonable corrections for local terrain (ANNEX H). Data input for this method is modest and software could easily incorporate the few look-up tables involved. Choosing an acceptable simplified method for estimating natural ventilation in free running buildings is a difficult task. Given Australia's widespread adoption of ISO standards, NatHERS use of the method in ISO 13791 which the SAA is considering for the same purpose NatHERS proposes, would probably be acceptable to people working in the field of building energy.

This method is only capable of estimating average flow through (near) openings and not local velocities within rooms away from openings. The relatively small dimensions of rooms in houses together with the tendency for occupants seeking thermal comfort to position themselves to benefit from any cooling breeze during summer, suggests that air flow limited to that near openings should be acceptable within NatHERS.

CONCLUSIONS AND RECOMMENDATIONS RELATED TO AIR FLOW

Given the restriction of climatic data sets within NatHERS to 28 selected locations, the writers consider that the following recommendations are within the levels of precision currently achieved within NatHERS computations.

Given attempts to limit heavy computation demands and data input in NatHERS, the writers recommend the adoption of the method for estimating natural ventilation in free running buildings as described in ISO 13791. Average air flow rates near each opening for assessing occupant cooling can be calculated by dividing the discharge rate through each opening by the area of the opening concerned. This air flow rate in m/s would be used in the equation $dt = 6(v-0.2) - (v-0.2)^2$ to determine the cooling effect on occupants.

With respect to air flow for summer cooling during calm conditions, advice should recommend the use of ceiling or whole house exhaust fans for energy efficient provision of air movement up to 2 m/s.

As passive houses require a significant level of understanding and participation by their occupants it is important that any output from the NatHERS program for free running buildings be qualified or explained by suitable notes to the user explaining assumed contextual limitations.

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FURTHER COMMENTS RELATED TO RECOMMENDATIONS

[In Response to Committee Feedback]

Thermal Comfort

There are basically two approaches to the determination of thermal comfort:

- 1) the analytical tracing of heat transfer from metabolism to the environment, with all its contributing factors (eg. Fanger's PMV or the J B Pierce Lab.'s 2-node model), usually coupled with laboratory measurements. This approach is today referred to as the "constancy theory", as it denies the role of acclimatisation;
- 2) observation, measurement and votes of people in their normal environment, engaged in their normal activity and wearing the clothes of their choice, with subsequent statistical evaluation and determination of "neutrality". This shows a close correlation with prevailing climatic conditions and is referred to as the "adaptive model".

The former may have a relevance in controlled, air conditioned environment work situations, where there is a uniformity of activity and clothing, as well as of expectations. In a domestic situation, where activities, clothing and expectations vary, where tolerance is greater, the latter approach is more realistic. Any pseudo-accuracy would be misleading and irrelevant.

Of all the comfort indices surveyed the ET* is the most reliable, proven and practical. Hence our recommendation to combine the ET* index with the adaptive model for thermal comfort.

At or near thermal comfort the DBT is the best measure, hence the best to use for the setting of neutrality. For the reasons stated, we recommend the adoption of the Auliciems definition:

$$(T_n = 17.6 + 0.31 \cdot T_{av})$$

This is to be taken as valid for 50% RH (with suggested limits of ± 2 K) and is subsequently compensated for humidity by the slope of the corresponding ET* line, labelled by that temperature.

Most definitions of the ET* are based on using an OT scale on the abscissa. OT = DBT, if and when MRT=DBT and the velocity v is negligible (< 0.2 m/s). For any other condition the OT value should be recalculated, preferably on an hourly basis. Clearly, this is impossible, not because of the magnitude of the computing task, but due to the lack and unpredictability of the necessary input data.

Air movement can be allowed for in three ways:

- 1) by using h_c (convective heat transfer coefficient) as a function of such air velocity in the OT expression for still air,
eg. $h_c = 8.3 \cdot v^{0.6}$ (for sedentary activity)
- 2) by taking the h_c as constant and using the subsequent correction of OT for v , ie:

$$OT = \frac{h_r MRT + h_c \left[DBT^* \sqrt{\frac{v}{v_o}} - t_s \left(\sqrt{\frac{v}{v_o}} - 1 \right) \right]}{h_r + h_c}$$

3) by using DBT (or OT for still air) on the abscissa of the psychrometric chart, fix the comfort limits in DBT terms and modify the upper comfort limit in a subsequent step as a function of air velocity according to the expression suggested on p.24. The air velocity value employed should be that which can be guaranteed to occur at the time of overheating either by natural means or by the use of a low velocity ceiling fan. We suggested the use of 1.5 m/s as the comfortable maximum velocity, which would give an adjustment to the upper comfort limit of some 6 K (although 2 m/s may be considered under overheated conditions in a domestic situation).

If Env.T is used in lieu of DBT in our suggested procedure, this will obviate the need for the adjustment for MRT and simplify calculations.

Further Comments on Thermal Comfort Indexes

There is a growing consensus among thermal comfort researchers that thermal preference of people strongly depends on temperatures they are normally exposed to. The analysis of a large amount of data by Humphreys (1975) showed a correlation of:

$$T_n = 2.6 + 0.831 * T_m$$

where T_n is the neutral temperature and T_m is the mean monthly air temperature experienced. Correlations of indoor neutrality with T_{om} , outdoor mean temperatures vary between:

$$T_n = 11.9 + 0.534 * T_{om} \quad (\text{Humphreys, 1978})$$

$$T_n = 17.6 + 0.31 * T_{om} \quad (\text{Auliciems, 1981})$$

$$T_n = 17 + 0.38 * T_{om} \quad (\text{Nicol \& Roaf, 1996})$$

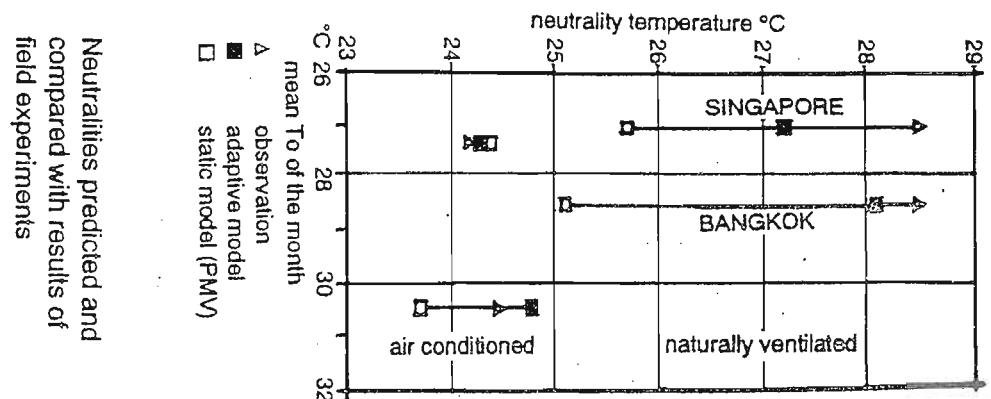
or, in terms of the definitive work of de Dear et al (1997), relating neutrality to outdoor effective temperature (E_{to}^*):

$$T_n = 18.9 + 0.255 * E_{to}^*$$

These are referred to as 'adaptive models' in contrast to Fanger's 'constancy model'.

The following graph compares observed preferences with predictions based on this expression and with predictions by the PMV method. Whilst there is a reasonable agreement for people in air conditioned buildings, even the adaptive model seems to underestimate the preferences of people in free-running buildings. (based on de Dear et al. 1997).

GRAPH OF OBSERVED PREFERENCES (adaptive/constancy comfort models)



"Comfort index" - the jury is no longer "out": at the 1996 CIBSE / ASHRAE joint conference the "adaptive model" won by miles. The Humphreys, Nicol, Auliciems, De Dear line, also supported by the work of Williamson (1995) carried the day. The ISO 7730 adopted Fanger's method. The ASHRAE standard is largely based on the J B Pierce 2-node model, which is quite similar to ISO 7730. Both are the products of cool northern climates. After the 1995 conference ASHRAE commissioned De Dear et al. to review the relevant information and produce an "adaptive model". The final report was submitted about six months ago and the end product is very similar to Auliciems' results (at least for non-conditioned buildings). Consequently Fanger's PMV or its computerised version (probably based on Spencer's 1975 work) are irrelevant. The reasoning behind the rejection of some of the other existing thermal comfort indexes is given below.

ECI: the formula $ECI = WBT + X(DBT - WBT) - Y\sqrt{v}$ could be included, but it is meaningless without the empirical graphs giving the values of X and Y. The index is based on a very small sample and it requires a trial-and-error method for its calculation. As it is clearly inappropriate, there is no need for explanation beyond what we have included.

Tropical Summer Index: adequately defined by the Tsi expression given. The table missed out in the production is very simple:

v(m/s) decrease Tsi	
0.5	1.4 K
1.0	2.0
1.5	2.5
2.0	2.8
2.5	3.2

and relates only to the "simplified" quick assessment. It was included for the sake of completeness (Indian Standard SP:41), no-one has evaluated it outside India, to the best of my knowledge (as stated), for this reason any further discussion is unnecessary.

RSI: this is based on the stated air velocity (100 ft/min [~ 0.5 m/s]+ 110 ft/min induced) and other velocities are not catered for. I am uncertain what further discussion of the interpretation could be expected. On p.21 we give a reason for its rejection. I could add that the main significance of RSI is the theoretical development, which led to later refinements both at the J B Pierce Lab. and at Fanger's laboratory.

ITS: there is a whole page presentation of ITS and p.21 suggests that it is far too complicated for practical use.

SET: the main benefit of this index is its comprehensiveness and universal applicability (including hyper- and hypobaric environments). We state (p.21) that - as we are facing fairly limited set of environmental conditions in tropical housing, the ET* (which is a subset of SET) is perfectly adequate.

ET*: we are quite familiar with the development of this index (the original work of Gagge, Stolwijk, Nishi, Gonzales, Hayter and Millikan, which found its way to various reference books, such as the ASHRAE "Fundamentals"), but the references suggested can be consulted by anyone interested.

ASHRAE Fundamentals volume states that the cooling effect of air movement can be extended by 1 K for each 0.275 m/s increase in air velocity, up to 28°C and 0.8 m/s air velocity this is repeated in ASHRAE Standard 55. The ASHRAE handbook is very conservative and it is by no means a "bible". Recent *adaptive*

developments will probably find their way into this handbook in a few years' time. A reviewer of this report asserted that Macpherson's work has been superseded by more recent work. This is not the case as the most recent work, is returning to the quoted 1962 views. Those were views or gut feelings then, but proven facts now. Even Fanger started retreating from his rigid assertion that thermal comfort is the same for the Eskimo as for the Hottentot. It is interesting to note that the Fundamentals still quotes from Fanger's 1972 book. We recommend the use of DBT to establish the neutrality, then extend it by the sloping ET* lines and air movement cooling effect. Beyond these limits we have discomfort. This may be elegant simplicity but certainly not "simplistic".

Use of ET* Lines

The ASHRAE definition (page 14 of this report) uses OT on the abscissa of the psychrometric chart. With negligible air movement OT = DBT.

As CHENATH calculates Env.T, $\{(2 \cdot \text{MRT} + \text{DBT})/3\}$ this is a good approximation of OT,

$\{(h_r \cdot \text{MRT} + h_c \cdot \text{DBT}) / (h_r + h_c)\}$ and if $h_r = 2 \cdot h_c$ then they are identical.

(eg with negligible air movement $h_c = 3 \text{ W/m}^2\text{K}$, and with light activity $h_r = 6 \text{ W/m}^2\text{K}$)

Therefore it is recommended that Env.T is used on the abscissa, when depicting indoor conditions.

Comfort limits $T_n + 2\text{K}$ and $T_n - 2\text{K}$ are located on the 50% RH curve, as by definition the ET* values coincide with DBT at the 50% RH curve.

The Auliciems expression gives T_n in terms of DBT (in the absence of radiant heat gain). This will coincide with Env.T when $\text{MRT} = \text{DBT}$, so the same chart can be used for comfort zone definition.

A more detailed but complex procedure for setting out the ET* lines is described on page 37 of the PLEA Note 3 (Appendix A). For the purposes of this report it is suggested that the simplified procedure using Equation 1 on page 15 which was produced based on the following reasoning:

- up to 14°C the ET* lines are vertical, thereafter they show an increasing slope, hence the term (T-14).
- the base-line (X-axis) intercept is found from the ABC triangle (Appendix B) where the side AB is the AH corresponding to 50%RH, which is 0.5•saturation humidity at that temperature.
- in Equation 1 the 0.025 coefficient is chosen empirically to give the nearest match to the irregularly sloping ET* lines of ASHRAE, in the temperature range normally experienced in our climates. Strictly speaking the ET* lines which this expression is attempting to match are valid for 0.6 clo and sedentary activity ($\approx 1.2 \text{ MET}$), but deviations from this in normal domestic conditions are negligible.

Appendix C shows the extensions of comfort zone by 1 m/s and 1.5m/s air movement, taken at the 12g/kg AH level. The hand-drawn lines would be produced if the extension were plotted at the 50% RH curve. These would produce differences d1 and d2. Admittedly, this may seem to be the logical thing to do, but the increasing slope of the 50%RH curve would exaggerate the cooling effect, especially at higher dry bulb air temperatures. The cooling effect is in DBT and not in ET*, as the plots of different estimates of air movement cooling effect (page 25 of this report) on which the approximation is based, are all in DBT. There is nothing wrong with adding a DBT value to an ET* value (per analogiam $15^\circ\text{C} + 5\text{K} = 20^\circ\text{C}$).

The authors do not offer an expression for calculating ET^* from a given DBT and AH, but one could probably be developed.

With respect to the thermal comfort zone, 80% acceptability, means that this the limit for 80% of respondents in the thermal response surveys. These comfort zone limits are not neutral temperatures; but if for example $T_n=25^\circ\text{C}$, then 80% of people would find temperatures between 23°C and 27°C acceptable. In the PMV system the span from -1 to +1 would include only some 52% of respondents, so it would be more restrictive than the proposal in this report. These limits are somewhat fuzzy as some authors suggest $T_n\pm 2.5\text{K}$ for 90% acceptability and $T_n\pm 3.5\text{K}$ for 80% acceptability. Fanger's argument for PMV relates to his analytically derived optimum temperature, which is taken to be the same for all people anywhere on Earth. He denies acclimatisation. Indeed $T_n-7\text{K}$ would be extremely low if T_n is defined by the adaptability model used in this report. An adjustment for increased clo is relevant for Fanger's constancy hypothesis, but the adaptability model defines T_n for people in their normal environment wearing clothes of their choice. Taking the above into account, an argument could be put for expanding the thermal comfort zone based on the adaptive hypothesis to $T_n\pm 2.5\text{K}$.

Early Australian Thermal Comfort Zones

It is interesting that the techniques proposed by Macfarlane in 1958, give results very close to those discussed above. Macfarlane provided summer and winter thermal comfort zones associated with ranges of latitude and very simple procedures for adjusting these thermal comfort zones to account for the influences of elevated humidity, radiant heat gain, and air movement. His adaptive thermal comfort zones for normal clothing were:

Latitude Range, Degrees	Summer DBT $^\circ\text{C}$ ($^\circ\text{F}$)	Winter DBT $^\circ\text{C}$ ($^\circ\text{F}$)
Greater than 30°	21.2 to 27.8 (70-82)	20.0 to 25.0 (68-77)
Less than 30°	23.9 to 30.6 (75-87)	21.2 to 27.8 (70-82)

His adjustment to these thermal comfort zones for RH greater than 60% was to lower the comfort zone by 0.83K (1.5°F) for each 10% increase in RH above 60%.

His adjustment to these thermal comfort zones for indoor radiant heat gain was to lower the comfort zone by 0.56K (1°F) for each 2.78K (5°F) increase of indoor surface temperatures above 37.8°C (100°F).

His adjustment to these thermal comfort zones for air movement was to raise the comfort zone by 0.56K (1°F) for each 0.15m/s (30ft/min) up to 1.02m/s (200ft/min) for DBT's less than 36.7°C (98°F). This cooling effect in K is $(0.56/0.15)$ or 3.73K for each m/s . This upper limit of 1.02m/s was commonly used in the 1950-60's based on the air movement which caused disturbance of loose papers. These days the limit on indoor air movement in naturally ventilated houses by Givoni and others is 2.0m/s .

It is worth noting that most thermal comfort is based on subjects in work environments in sitting or standing position. Macfarlane refers to experiments at the Experimental Building Station in Sydney on subjects in both seated and prone positions on a bed. Evans (1979) provided the following table of thermal comfort limits suggesting comfort zones for day as well as those for night which tend to reflect statements regarding *sleeping cool* by Macfarlane (1981) and Macpherson (1956).

Average Monthly RH%	Average Annual DBT >20°C	Average Annual DBT >20°C	Average Annual DBT 15-20°C	Average Annual DBT 15-20°C	Average Annual DBT <15°C	Average Annual DBT <15°C
	Day	Night	Day	Night	Day	Night
0-30%	26-34°C	17-25°C	23-32°C	15-23°C	21-30°C	14-21°C
30-50%	25-31°C	17-24°C	22-30°C	15-22°C	21-27°C	14-20°C
50-70%	23-29°C	17-23°C	21-28°C	15-21°C	19-26°C	14-19°C
70-100%	22-27°C	17-21°C	20-25°C	15-20°C	18-24°C	14-18°C

On Star Ratings

Our recommendation is not to the use of duration of discomfort "solely". It is quite clearly stated (p.41) that the calculations should find the duration of comfort conditions and the degree-hours (ie. the product of duration and magnitude) of under- or overheating and (p.42) the basis of star rating in warm climates should be the **degree-hours of overheating** as ventilation for cooling is not relevant during winter. The correspondence between star rating and number of degree-hours of overheating must vary with the locations and its definition is beyond the scope of the present study. Aggregation of results of several zones: This was not mentioned in our brief but individual zones can be rated so all that has to be agreed is an aggregation method to give a similar ranking to the NatHERS energy version.

The rating - in our view - should be based on the passive performance of the house (with the exception of ceiling fans). A house good for cross-ventilation would be hopeless for air conditioning if such an appliance is subsequently installed. How to cope with this problem is a policy decision (carbon tax ?): beyond the scope of the present study.

Kowalczewski (1968, Inst.Eng.Aus. Mech.& Chem.Eng.Transactions vol. MC4, 1.May, p.55-61), discusses the problem of **magnitude versus duration** and concludes that the yearly stress index will be the same whichever way the product is generated (ie. twice the duration but half the magnitude gives the same result). This would adequately justify the use of overheated K.h (kelvin-hours) as the basis of star rating.

Our recommendation with respect to "accepting Lee's advice to allow for acclimatisation" refers to his 1980 paper, (p.353) (see References to our section on "The choice of an index" on p.23): "For domestic and office situations ... the comfort zone on the effective temperature (ET) scheme, particularly as modified by Koch and others appears adequate, particularly *if allowance is made for variations in acclimatisation.*" (our italics).

Regarding reference by a reviewer on Dean Beltrame's thesis on Thermal capacitance, etc, whilst the document is a well constructed student exercise, attributing any significance to it beyond this would be a grave mistake. It is very naive, displays an ignorance of the huge amount of work done in this subject area in the last 50 years, it is very narrow in the alternatives examined and simplistic in its conclusions. The writer seems to arrive at these conclusions based on a very selective set of evidence. To follow his recommendations would be very erroneous.

He concludes that fibro is better than brick. He recommends not using any wall or roof insulation, to use galvanised iron, rather than tiled roofs and timber floor, rather than concrete. This may all be valid, if and only if the inside is warmer than the outside (or indeed the sol-air temperature of the outside surface). If there is an outward heat flow, one would not want to prevent it.

Fanger's PPD, which Beltrame uses as an evaluation criterion may be applicable to heated buildings in cold climates, but not to 'free-running' buildings and certainly not in tropical climates.

Beltrame uses 2-5 air changes per hour, which may reduce the indoor temperature nearer to the outdoor one, but one would need up to 30 air changes per hour (which can be achieved by full cross-ventilation) to almost equalise the two temperatures. Estimates of air changes (from 307 to 636 ac/hr) through the Living Room of this building due to typical onshore breezes in Townsville around 3pm in January using the procedures described in Appendix H of our report are provided in Appendix D.

Beltrame totally ignores the physiological cooling effect of air movement (where the critical parameter is local air velocity and not volumetric air flow).

Rating and its Purpose

Ostensibly, the purpose of HERS is to promote energy conservation (and thereby reduce greenhouse gas emissions). Energy rating in cool climates is a fairly clear-cut issue. In warm climates a house would be judged as 'good' if it has an adequate passive performance, thus obviate the need for air conditioning. I do think that the term 'Comfort Rating' is a misnomer. We need a system which rates the free-running performance of houses. Perhaps the term 'Passive Rating' would be more appropriate.

The whole process of developing such a scheme could quickly become over-complicated. What is needed is a comparative rating system. Thermal comfort is a huge subject and not the prime concern of HERS and to avoid hair-splitting we suggest the following:

- 1) set a notional upper comfort limit for each climate zone: any system or index would do, but why not use the Auliciems' expression, which is the only one 'made in Australia' (perhaps modified to use the ET^* , in lieu of DBT)
- 2) predict the indoor temperatures by simulation (based on a use pattern, including ventilation, which is reasonable for the area) and express the results as the product of excess temperature magnitude and duration, ie. in terms of Kelvin-hours
- 3) consider also underheating and produce the number of underheated indoor Kelvin-hours (nights can be uncomfortably cool even in Darwin, down to about 15°C in July) - how does the house cope with this?
- 4) do this simulation for the best possible house for the given climate, as well as for the worst, least suitable. The span of values thus produced can then be divided into (equal?) bands and any house would receive the star rating according to which band it falls into.

The assumption of a reasonable use-pattern for the purposes of such simulation needs to be discussed. For ventilation we suggest that after sunrise the house should be taken as closed (with minimum ventilation, perhaps not exceeding infiltration) as long as the inside is cooler than the outside. Then increase the ventilation as the indoor-outdoor differential increases, to quite high levels, perhaps up to 30 ac/hr. The house must be judged whether it has adequate openings to make such air exchange possible: if not, then the maximum rate would be lower. One could argue on this basis that no allowance needs to be made for air movement (physiological cooling) effects, as this would generate difficulties and it would be unpredictable for the future. It can be taken for granted that any house could have ceiling fans, thus this need not be included in the comparison.

Note that even the rating of domestic refrigerators is based on a standardised use-pattern. If the user keeps the frig door open for extended periods, or puts hot

dishes into the refrigerator to cool these down quickly, the energy use will increase. This of course does not reduce the star rating of the product itself.

Strategies for Design of Houses in the Tropics

Essentially there are two strategies for the design of houses in warm humid tropical areas of Australia:

a) Slab-on-ground floor, well shaded or externally insulated massive construction, well insulated roof; operated with closed openings during much of the day and maximised ventilation as soon as indoor air temperature exceeds outdoor shade air temperature, mainly overnight; this is the only way to keep the indoors cooler than the outdoors, for at least part of the day-time.

b) A lightweight, elevated building, with a single row of rooms, for full cross-ventilation. This would rely on keeping the indoor temperature not much higher than the outdoors and on the physiological cooling effect of air movement; even this would need at least the roof well insulated, to avoid the MRT increase caused by a hot ceiling and the inward heat flow caused by the substantially elevated solar temperature of the outer surface.

Both can be successful, if well done. Beltrame seems to have used the worst features of both. In both alternatives all openings must be fully shaded and indoor heat sources reduced to a minimum.

The houses by Troppo Architects in Darwin come close to the second alternative. This is the expression of 'traditional wisdom' (of the last 50 years, or so). It is agreed that such houses would be hopeless for the subsequent installation of air conditioners. This is why preference should be given to the first alternative, which can be equally successful as a purely passive, free-running building or as an air conditioned one.

Comments on Thermal Simulation of Buildings

A few problems associated with current thermal simulation of buildings on computers are:

1) all simulations (we know of) ignore the physiological cooling effect of air movement: they only allow for volumetric displacement ventilation and do not consider air velocity, so a study based exclusively on such simulation may lead to errors

2) whilst there are some difficulties in relating ventilation to meteorological wind data, we suggest that this need not be considered necessary for the purposes of a rating scheme: ceiling fans can reliably provide the desired velocities

3) observation of the life-style of people in the Australian tropics shows that they prefer open air, indoor-outdoor living and would hate to be 'cooped up' in a sealed box, even if it is air conditioned

4) a study (with Bal Saini) quite some time ago (for the CCPSO), which included the use of Terry Brealy's 'home improvement game' showed that only 24% of respondents would like an air conditioner for one room (usually the master bedroom), 5% would like half-house air conditioning and none would want the whole house to be air conditioned.

Alternative Options for Estimating Indoor Air Flow due to Wind

If indoor air flow is used to evaluate summer cooling of occupants, there are a number of alternative procedures which can be used.

Computation of Indoor Airflow

Professor Shuzo Murakami, University of Tokyo, chaired a task group on correlating CFD output from various researchers for flow around a cube immersed in a turbulent boundary layer with wind tunnel flows. The task group reported to the Second International Symposium on Computational Wind Engineering at Colorado State University, Fort Collins CO in 1996. Best correlations were achieved using large eddy simulation methods but the most useful studies by Murakami have taken up to 24 hours for computation and a further 24 hours to generate the flow visualisation for a few minutes of real time flow simulation on a Fujitsu computer equivalent to a Cray 1 super computer. The conclusion of this international group of experts was that CFD, although it shows great potential, is not yet at a stage where it can be recommended for serious engineering design purposes (proceedings to be published soon in the *Journal of Wind Engineering and Industrial Aerodynamics*). Many recent publications on the application of CFD on small computers to the evaluation of indoor airflow include modelling of the external airflow. Such applications of CFD are severely compromised by the limitation to very coarse grids.

Naturally cross ventilated houses with large openings are unstable *high flow - low resistance* flow network systems. When air flow in these networks is computed using simpler resistance network flow programs such as COMIS there is a tendency to pulsing flow over a range of low frequency when using mean wind pressure differences at external openings. This pulsing flow results in slow convergence in computation as reported by G. Walton, *ASHRAE Transactions*, 95,2,611-620, 1989 (and this is without the additional complexity of external pressure fluctuations arising from turbulence in the approaching wind or bluff body vortex shedding).

With wind-generated natural cross ventilation one could attempt either 2D or preferably 3D CFD simulation of indoor airflow provided suitable wind pressure distributions were available. By replacing CFD modelling of external air flow by wind pressures at wall openings computation is significantly reduced. A 2D approach would be more compatible with NatHERS data formats but in either case local mean air velocity will be the quantity needed for evaluating indoor thermal comfort/discomfort. This raises the issue of policy on integrating comfort/discomfort levels throughout each space or zone. How would one establish a standard grid, multi layered vertically if 3D simulation is attempted. Such a system would dramatically increase the computation within NatHERS and create dilemmas of how to handle the large amounts of velocity data generated in comfort/discomfort evaluation.

Nathers has adopted simplified data structures for good reasons. The current NatHERS has data outputs related to zones not arrays of points within a zone. On balance use of a zonal method such as that set out in Appendix H is in keeping with the practical approach already adopted in NatHERS. As house plan geometry is restricted to simple rectangles Swami and Chandra (1988) or Knoll et al (1996) models could be used to estimate wind pressure distributions on external walls. Another modification worth considering is Cosine correction for the area of windward openings when wind is at inclined incidence. Although the examples output air changes per hour, it can easily be converted to output mean velocity through (near) external and internal openings. Such an approach can be argued if an adaptive model for thermal comfort is adopted. Householders observed in free-running buildings who adjust their clothing and activity levels with changes in their thermal environment are very likely to position themselves near a window or doorway opening to benefit from the cooling effects of a breeze. There is not much point in developing sophisticated indoor air flow simulation from natural ventilation in NatHERS if in a large number of cases in summer evenings as there are frequent calms.

A more important consideration in tropical regions is the occurrence of calms, particularly during the evening (see attached 20 year wind frequency data for Townsville at 1500 hrs and 2100 hrs). Darwin being of lesser latitude is likely to have even higher frequencies of calms at night. One simple way out of this energy versus comfort issue is to promote the installation of ceiling fans (as McPherson did in 1962) which are virtually universal in the tropics in any case. After all NatHERS assumes that there will be adequate heating and cooling equipment installed in its current closed envelope model. While natural ventilation is desirable in warm humid tropical environments it is not as welcome in dusty hot arid environments. Simple advice could be incorporated in NatHERS to inform householders in the choice of ceiling fans in warm humid environments or evaporative coolers based on Givoni's suggested limit of 22°C WBT (page 134, Passive and low energy cooling of buildings, Van Nostrand Reinhold, 1994). This would certainly simplify the programming aspects of NatHERS.

Wind Incidence to Windward Openings

The procedures used in ISO/DIS 13791 are based on what is referred to as the *zonal method*. There is a typical deficiency of the zonal method in accounting for the momentum of the jet of air entering the windward openings. Air flow through leeward openings tends to be less effected by air flow directionality due to the influence of non-directional suction pressures in the building wake. An important influence of such jets at windward openings is the directionality of the air flow through the windward openings. These jets tend to maintain the direction of the approaching airflow as they pass through a windward opening. For wind incidences other than normal this reduces the effective width of windward openings in proportion to the Cosine of the angle of incidence of the wind (Aynsley et al, 1977). Allowance for this effect in a new CHEENATH would be relatively simple.

Wind Data

Consideration of long term wind frequency should be given to wind data in the NatHERS climate data files. A typical year of hour by hour wind speeds and directions is unable to reflect significant changes in wind patterns over 20 years due to the Southern Oscillation Index and sunspot cycles of 7 to 10 years. This wind frequency data should use wind speed intervals of 0.2m/s between calms and 3m/s to reduce catenation errors in calculating cooling effects of indoor air movement. Larger wind speed intervals of say 0.5m/s could be used for wind speeds greater than 3m/s to reduce computation time. Indoor wind speed coefficients referenced to local outdoor 10m wind speeds typically range from 0 to 0.8 (Aynsley, 1996).

Irrelevance of Stack Effect in Air Flow Through Houses in Summer

As air movement for indoor thermal comfort is only relevant during summer in locations where DBT's are less than 36.7°C (Macfarlane, 1958). No one needs breezes through their house during winter to improve indoor thermal comfort. The thermal buoyancy contribution to indoor airflow in ISO/DIS 13791 could be eliminated as the temperature differences and the stack heights between inlet and outlets in houses are likely to be insignificant in the case of houses with respect to air movement for indoor thermal comfort. The indoor air movement needed to restore indoor thermal comfort around the coastline of Australia during summer is likely to be between 0.5 and 1.5m/s extending up to 2.0m/s in humid tropical regions.

Zonal Methods for Estimating Indoor Air Movement

Zonal method of estimating indoor airflow can only provide reliable information on average air movement through openings. This is not necessarily critical as people seeking the comfort of a summer breeze will tend to position themselves in locations where the best breezes are, near openings. In the case of simple cross ventilation where all air flow through the house is through a number of openings in series between the windward and leeward walls, air flows can be calculated

directly from wind pressure differences and discharge coefficients of openings or air flow resistances of internal air flow paths. Where there are multiple entry and exit openings and internal branching of air flow paths air flow along each path cannot be calculated directly. In these network flows an iterative method of air flow estimation is applied in a process that models laws of balancing flows against head loss and conservation of mass within the enclosure (Aynsley, 1997). With most houses in NatHERS having three zones, network flows are to be expected. Public domain software for such analysis is available such as COMIS and CSIRO currently has software to perform such network air flow analysis.

CFD Methods for Estimating Indoor Air Movement

If detailed distribution of air movement within a house away from openings is felt to be desirable, then computational fluid dynamics (CFD) is the most suitable tool. CFD can be performed in 2 dimensions or 3 dimensions. Flows around the exterior of houses are very complex and certainly 3 dimensional. Internal air flows within single storey houses are usually less complex can be approximated by 2 dimensional CFD analysis. This approach would only require a 2 dimensional graphical description of each house, similar to that developed by Willrath in BERS of Drogemuller in ARCHIPAK.

The complexity of air flow around the exterior of houses requires supercomputing capacity to achieve realistic air flow patterns, however less rigorous analysis on personal computers of indoor air flow may be of acceptable accuracy if combined with empirical models of external wind pressure distribution such as that by Swami and Chandra (1988) or Knoll et al (1996). Such a combination of an empirical external wind pressure model with 2 dimensional CFD analysis of internal air flows could be a viable alternative with a relatively low computation demand.

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AGGREGATION OF THERMAL DISCOMFORT IN NATURALLY VENTILATED BUILDINGS

Researchers have suggested a variety of means for evaluating indoor thermal comfort in naturally ventilated buildings. In attempts to limit computation, 3 PM climatic conditions in the hottest month has been targeted as an appropriate design condition to satisfy (Aynsley et al, 1977, 1979). Diurnal assessment of indoor thermal comfort in warm humid climates have been studied using both 3 hourly average data and a selected month of actual 3 hourly data (Aynsley, 1996, 1997). Others have used monthly summary weather data (Arens & Watanabe, 1986). Only a few have attempted to make use of hour-by-hour weather data (Byrne et al, 1986), (Arens et al, 1984) due to the computation involved.

The principal difference between energy rating of closed envelope homes and thermal comfort/discomfort rating in both closed and open envelope homes is the criteria used in assessing conditions inside a home. New effective temperature, ET^* , is recommended as the thermal comfort scale which can be readily adjusted to account for the cooling effect of airflow due to natural ventilation up to 2 m/s in warm environments. Many of the same data used in NatHERS evaluation will be applicable to evaluating the incidence of discomfort in closed buildings. New evaluation procedures will be needed to assess thermal discomfort in open envelope homes utilising natural ventilation for summer comfort. In open envelope homes natural ventilation rates can be very high with air speeds up to 2 m/s. In such situations indoor air temperature, and humidity of air passing through windows and door openings will be similar to ambient outdoor air conditions.

A new procedure necessary to undertake thermal comfort/discomfort evaluation in closed envelope homes is determination of indoor New Effective Temperature, ET^* , from data produced through hourly thermal simulations using the CHENATH engine of NatHERS. The thermal comfort zone, defined by a range of ET^* , takes into account the variation in thermal comfort zone at different time of the year. The thermal comfort zone is defined as ET^* 's $\pm 2^\circ\text{C}$ either side of a monthly thermal neutrality temperature. The thermal neutrality is a function of mean monthly dry bulb air temperature.

Use of the current energy-based NatHERS software to evaluate summer energy demand of, naturally ventilated, buildings in summer would lead to lower star ratings than are warranted. This is because the current NatHERS software assumes a closed building envelope and calculates energy based on a limited use of natural ventilation for cooling to achieve indoor thermal comfort.

A low star rating for naturally ventilated homes, particularly in the warm humid or humid tropical parts of Australia could encourage excessive insulation which would increase retention of indoor heat during evening hours when night cooling by radiation to the sky is most effective (Givonni, 1994). Excessive insulation and any tendency to close windows would reduce diurnal indoor thermal comfort particularly in the afternoons and evening and lead to increased energy demand by air conditioners for summer cooling.

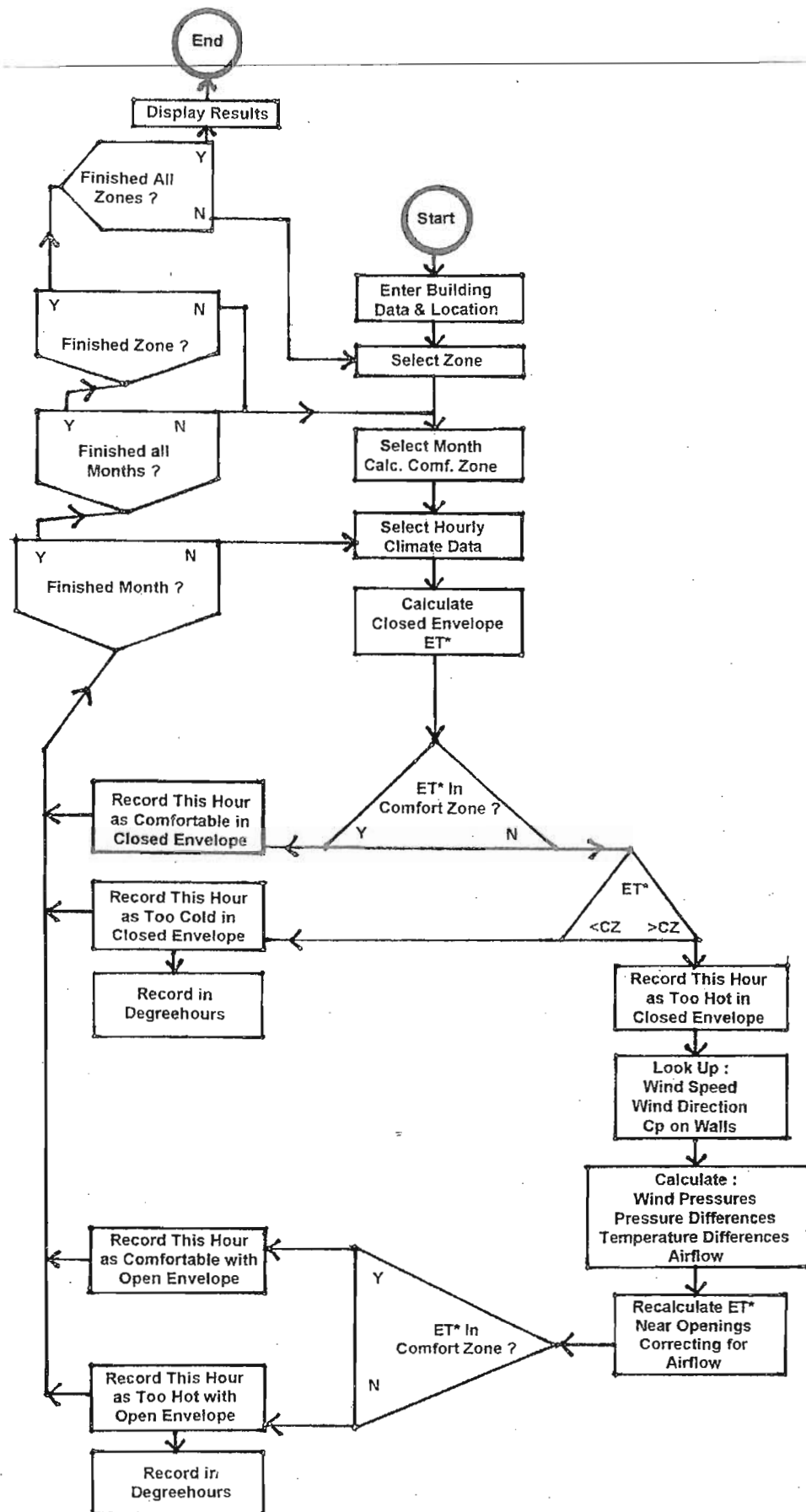
~~It should also be noted that air conditioning in warm humid tropical regions requires more energy and over many months than in more temperate climates. This is because the cooling season is longer and a larger proportion of the summer air conditioning cooling load is latent load due to dehumidification.~~

An important aspect of the monthly ET^* thermal comfort criteria is that by adopting the more tolerant indoor criteria incorporating seasonal acclimatisation (de Dear et al, 1985, 1991), (Macfarlane, 1981), the scope for natural ventilation for summer cooling of homes would be increased. There is some evidence to suggest that people are more sensitive to heat in bed on summer nights (Williamson et al, 1995). An older publication suggests daytime and nighttime thermal comfort zones for tropical climates (Evans, 1979). Further research is necessary if this aspect is considered important. Further refinements to accounting for the influence of air movement on indoor thermal comfort should be anticipated as there has been a recent increase in research activity in this field, particularly in Japan (Fountain, 1995), (Kubo et al, 1996), (Xu et al, 1996), (Fountain, 1996).

Another new procedure required for the new NatHERS proposal would be evaluation of natural ventilation through homes in open envelope mode. This would involve use of hourly wind speed and direction data, in a series of procedures which account for local terrain roughness, wind pressure distribution on walls of buildings, airflow efficiency of windows and door openings and the pressure difference created by differences in indoor and outdoor air temperature. A procedure selected for evaluating natural ventilation in homes is described in ISO Standard 13791. More sophisticated methods exist for estimating natural ventilation but they require extensive data input are sufficiently computationally demanding to require a powerful work station to run satisfactorily.

Aggregating hour-by-hour thermal comfort/discomfort assessments could follow the sequence (also see flow chart diagram) :

- Determine thermal comfort zone for the month.
- Calculate hour-by-hour indoor ET^* in a zone of the home in closed envelope mode.
- Check if ET^* is within the thermal comfort zone - note if too hot or too cold.
- If ET^* is too hot, calculate natural ventilation and air speed near openings.
- Check if ET^* for ambient air with airflow near opening openings in the zone is within thermal comfort zone.
- Note if ET^* is too hot.
- Proceed to next hour of monthly data.
- If months hour-by-hour data is completed, move on to next months data.
- If 12 months of hour-by-hour climate has been completed, proceed to next zone.
- If evaluation of hour-by-hour thermal comfort for all zones is complete, display a thermal comfort star rating (based on percentage of time per year when thermal comfort is achieved and degreehours when thermal comfort is not achieved) for each zone of the home, noting when building should be opened to encourage natural ventilation.



Recording the number of degreehours by which the comfort zone is exceeded in each zone of the home will allow differentiation in star ratings between homes with the same number of discomfort hours but with varying levels of discomfort (degrees above comfort zone).

As householders attempt to make increased use of passive thermal control in their homes, they will need to devote more of their time to operating their passive home. This is particularly true when building envelopes are opened up to take advantage of natural ventilation for cooling in summer. The new thermal comfort/discomfort based software proposed in the NatHERS Project should devote substantial effort to providing practical advice to home occupants on efficient operation of their home. When natural ventilation is insufficient to restore summer indoor thermal comfort, home owners should be encouraged to use ceiling fans or whole house attic fans delivering 2.0 m/s of air movement (Aynsley, 1977), (Chandra et al, 1985,1986), (Macfarlane, 1981).

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In accordance with the 2-node model, the method of setting out the ET* lines on the psychrometric chart consists of the following steps:

(for the illustrative example: comfort condition, the following values are used:

$$\begin{array}{lll} M = 58.2 \text{ W/m}^2 & h' = 4.56 \text{ W/m}^2\text{K} & t_{sk} = 33.5^\circ\text{C} \\ M_{sk} = 52.4 \text{ W/m}^2 & h_e = 0.04 \text{ W/m}^2\text{Pa} & w = 0.06 \end{array}$$

1. mark the skin temperature (t_{sk} , eg. 33.5°C) on the X axis and project it up to the saturation curve (PSK); on the Y axis this gives the saturation vapour pressure of the skin ($P_{ssk} = 5.33 \text{ kPa}$); extend this line (parallel with the X-axis) to the left
2. the quotient of metabolic heat reaching the skin and the heat loss coefficient gives a temperature difference:

$$\frac{M_{sk}}{h'} \rightarrow \frac{\text{W/m}^2}{\text{W/m}^2\text{K}} = \text{K} \quad \text{e.g.: } \frac{52.4}{4.56} = 11.5 \text{ K}$$

subtract this from the t_{sk} and mark it on the X-axis; project it up to the extended P_{ssk} -PSK line to define the point CP, the starting point of the corresponding ET* line

3. if Ψ is used to denote the ratio of the combined to the evaporative heat loss coefficient:

$$\Psi = \frac{h'}{h_e} \rightarrow \frac{\text{W/m}^2\text{K}}{\text{W/m}^2\text{Pa}} = \frac{\text{Pa}}{\text{K}} \quad \text{e.g.: } \frac{4.56}{0.04} = 114 \frac{\text{Pa}}{\text{K}}$$

4. then this, divided by the skin wettedness (w , non-dimensional) gives the negative slope of the ET* line:

$$\frac{114}{0.06} = 1.9 \frac{\text{kPa}}{\text{K}}$$

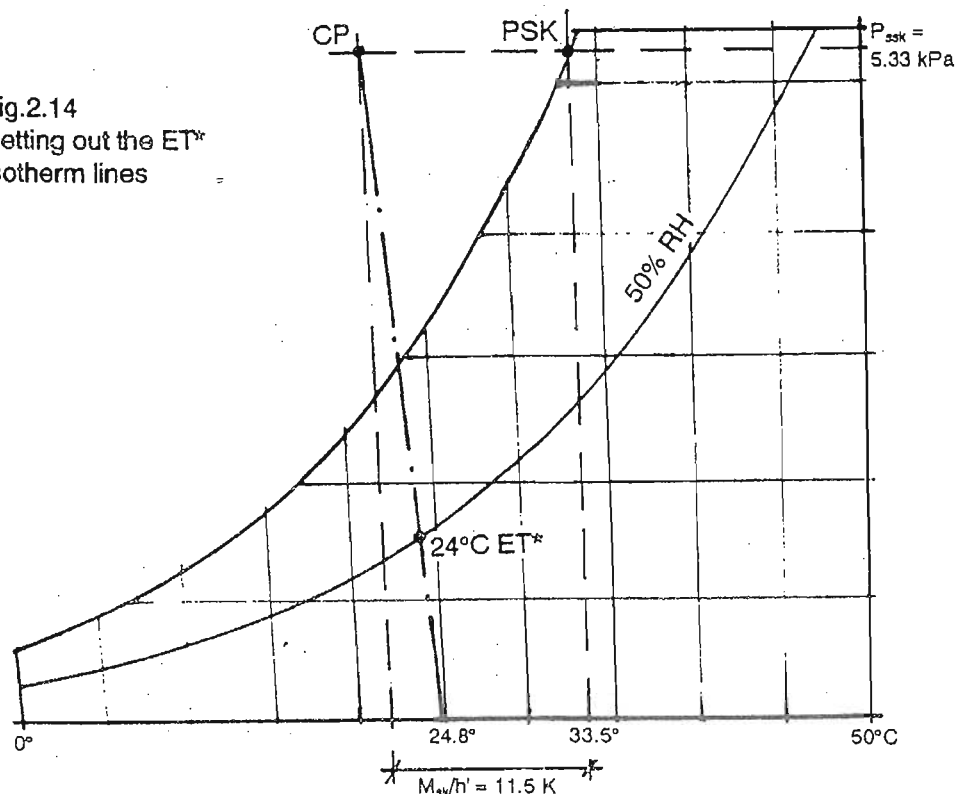
5. as $P_{ssk} = 5.33 \text{ kPa}$, the base line (X-axis) intercept of the ET* will be

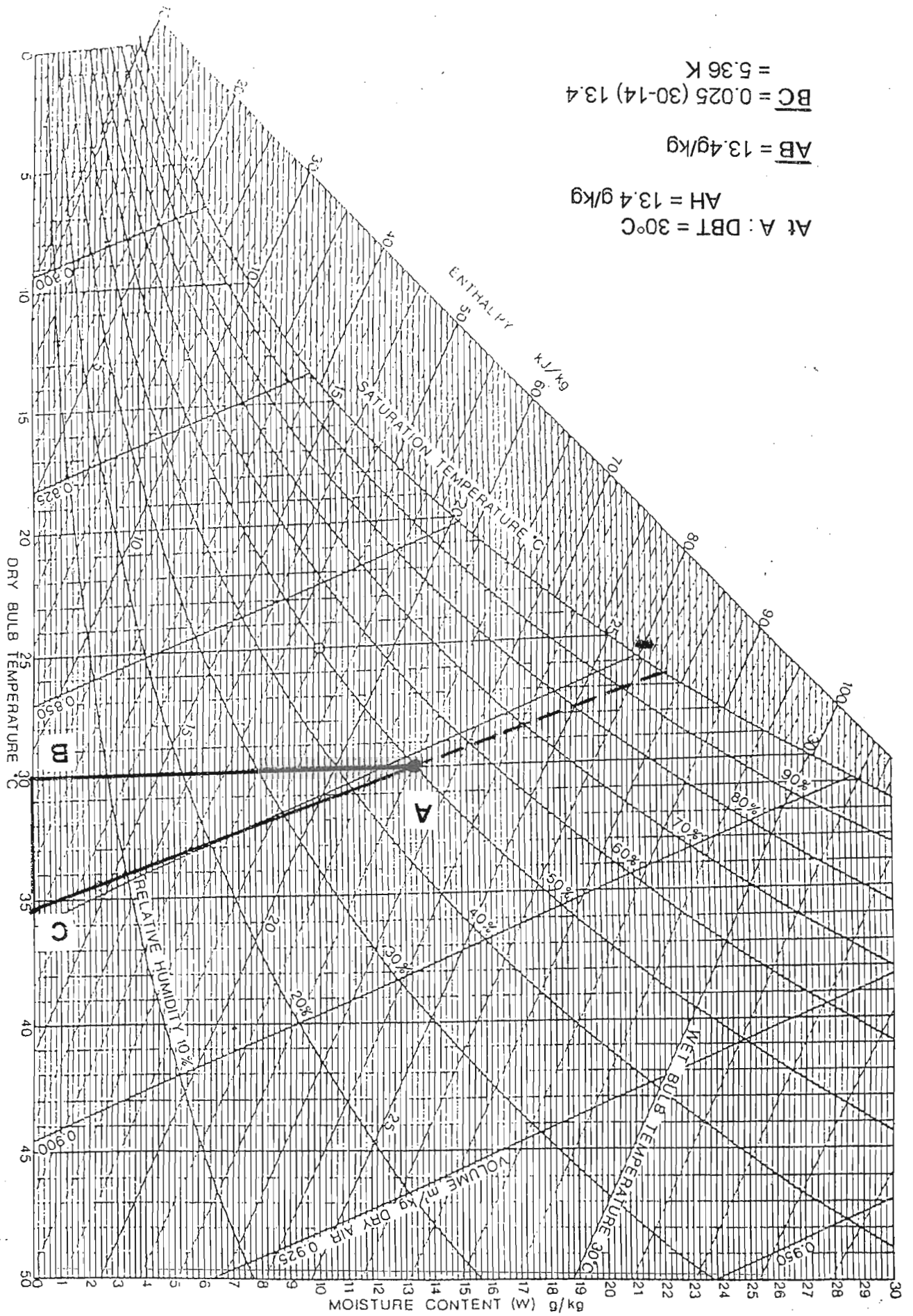
$$22 + \frac{5.33}{1.9} = 24.8^\circ\text{C}$$

6. this ET* line intersects the 50% RH curve at 24°C DBT, thus it will be calibrated as 24°C ET^*

For each subsequent ET* line the location of CP differs, the process must be repeated with the appropriate values and coefficients.

Fig.2.14
Setting out the ET*
isotherm lines



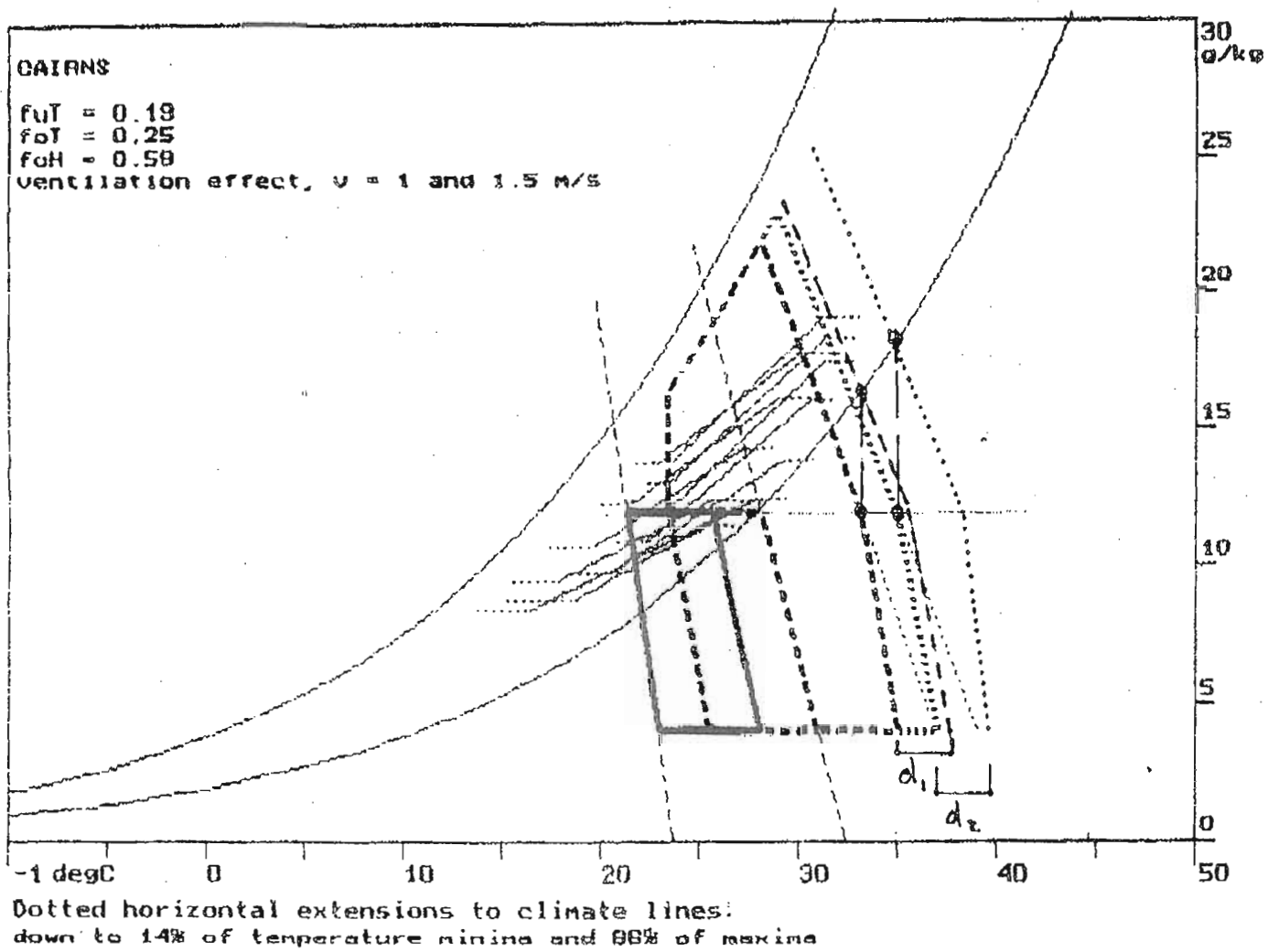


$$\overline{BC} = 0.025 (30-14) 13.4 = 5.36 \text{ K}$$

$$\overline{AB} = 13.4 \text{ g/kg}$$

$$\text{At A: DBT} = 30^\circ\text{C}$$

$$\text{AH} = 13.4 \text{ g/kg}$$



ANNEX H
AIR VENTILATION
(Informative)

NOTE : THE CONTENT OF THIS ANNEX IS WITHIN THE SCOPE OF TC 156. HOWEVER, THE FOLLOWING INFORMATION IS GIVEN, AS LONG AS NO BETTER PROPOSAL IS PROVIDED BY THIS TC.

H.1 Introduction

This annex gives a procedure for the evaluation of the air flow rate due to natural ventilation through the openings in building. It may be used when no detailed solution model based on the equilibrium of the air mass exchanges is available.

H.2 Calculation procedure

The amount of air flowing into the building depends upon the pressure difference between the inside and the outside and also on the resistance that any openings give to the flow of air. The pressure difference is produced by the action of the wind flow around the building and by the difference in the density of the internal and external air. The aerodynamics of the air flowing are complex but, by deeming openings to be classifiable within two general types, it is possible to specify simple formulae to relate the flow rate to the pressure difference. These categories are :

- a) cracks, or small openings with a typical dimension less than approximately 10 mm;
- b) openings with a typical dimension larger than approximately 10 mm

H.2.1 Cracks

For cracks the ventilation rate is given as :

$$m = \rho k L (\Delta p)^n \quad (\text{H.1})$$

where :

- ρ is the density of the air
- k is the crack coefficient
- L is the length of the crack in metres
- Δp is the applied pressure difference in Pa (N/m^2)

Table H.1 gives a range of values of k for the cracks formed around the openings lights of closed windows, to be used when National value are not available. A suitable value for n is 0.67. Pressure difference between inside and outside is obtain by adding the thermal buoyancy Δp_T and the wind contribute Δp_w .

$$\Delta p = \Delta p_T + \Delta p_w \quad (\text{H.2})$$

The thermal buoyancy Δp_T contribute is given by :

$$\Delta p_T = (\Delta T \rho g H_1 / T_m) \quad (H.3)$$

where :

- ΔT is the temperature difference between inside and outside
- T_m is the reference temperature (300 K)
- g is 9.81 m/(s²)
- ρ is the air density
- H_1 is the height between high and low level

The wind contribute Δp_w is given by :

$$\Delta p_w = \rho v_r^2 / 2 \quad (H.4)$$

where :

- ρ is the air density
- v_r is the wind reference velocity

Table H.1 Values of k for windows (in litres/s per metre of crack length for an applied pressure difference of 1 Pa)

Window type	value of k	
	average	range
Sliding	0,08	0,02 to 0,30
Pivoted	0,21	0,06 to 0,80
Pivoted(weather stripped)	0,08	0,005 to 0,20

H.2.2 Large openings

The air flow rate can be evaluated by the approximate formula :

$$m = \rho c_d A (2 \Delta p / \rho)^{0,5} \quad (H.5)$$

where :

- c_d is the coefficient of discharge
- A is the area of opening
- Δp is the pressure difference
- ρ is the air density

Values of coefficient c_d and Area A depend on the position and flow characteristics of all openings. It is conventional to assign a value to the discharge coefficient corresponding to that for a sharp-edged orifice, taken here as 0,61. The value of A for other types of opening then

becomes the equivalent area associated with that particular opening. Pressure generated by forces of wind and temperature difference produce a movement of air through these openings governed by the fact that the total flow of air into the circulation space equals the outgoing flow rate. Because of the impossibility to obtain a detailed solution for existing buildings, the solution is given for simple cases.

H.2.2.1 Simple buildings : openings on two facades

Figure H.1 shows a simple two dimensional representation of a building with no internal divisions and therefore consisting of a single cell with openings as shown, i.e. two (A_1 and A_3) at high level, and two (A_2 and A_4) at low level. According to the following situations the air flow by natural ventilation is determined as :

H.2.2.1.1 Wind only

$$m_w = c_d A_w v_r (\Delta c_p)^{0,5} \quad (H.6)$$

$$1/(A_w^2) = 1/(A_1 + A_2)^2 + 1/(A_3 + A_4)^2 \quad (H.7)$$

where :

Δc_p is the applied differential mean pressure coefficient calculated as difference between the mean surface pressure coefficients, derived from table H.2 for the opposite walls perpendicular to the wind direction.

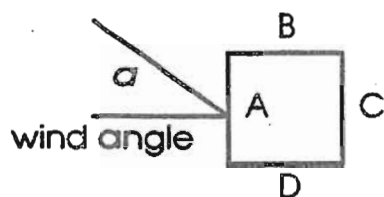
$$v_r = K z^a v_f \quad (H.8)$$

where :

K is a coefficient reported in table H.3
 a is a coefficient reported in table H.3
 z is the height of the building
 v_f is the free wind velocity

Table H.2 Coefficient c_p as a function of the wind velocity and the exposure of the opposite walls

Building height ratio (h/w)	Building plan ratio (l/w)	wind angle	c_n surfaces			
			A	B	C	D
(h/w) < 1/2	$1 < (l/w) < 3/2$	0	0,7	-0,2	-0,5	-0,5
		90	-0,5	-0,5	0,7	-0,2
	$3/2 < (l/w) < 4$	0	0,7	-0,2	-0,6	-0,6
		90	-0,5	-0,5	0,7	-0,1
$1/2 < (h/w) < 3/2$	$1 < (l/w) < 3/2$	0	0,7	-0,2	-0,6	-0,6
		90	-0,6	-0,6	0,7	-0,2
	$3/2 < (l/w) < 4$	0	0,7	-0,3	-0,7	-0,7
		90	-0,5	-0,5	0,7	0,1
$3/2 < (h/w) < 6$	$1 < (l/w) < 3/2$	0	0,8	-0,2	-0,8	-0,8
		90	-0,8	-0,8	0,8	-0,2
	$3/2 < (l/w) < 4$	0	0,7	-0,4	-0,7	-0,7
		90	-0,5	-0,5	0,8	-0,1



Number of surfaces with respect to the wind direction

l length of the building
w depth of the building
h height of the building

Table H.3 Coefficient K and coefficient a

Terrain	K	a
Open flat country	06,8	0,17
Country with scattered wind breaks	0,52	0,20
Urban	0,35	0,25
City	0,21	0,33

H.2.2.1.2 Temperature difference only

$$m_T = c_d A_T (2 \Delta\theta g H_1 / T_m)^{0.5} \quad (H.9)$$

$$1/(A_T^2) = 1/(A_3 + A_2)^2 + 1/(A_2 + A_4)^2 \quad (H.10)$$

where :

- $\Delta\theta$ is the temperature difference between inside and outside
- T_m is the reference temperature (300 K)
- g is the acceleration of gravity (9.81 m/(s²))
- H_1 is the height between high and low level

H.2.2.1.3 Wind and temperature difference together

$$m = m_T \text{ for } v_f/(\Delta T)^{0.5} < (0.26 (A_T / A_w)^{0.5} (H_1 / \Delta c_p)^{0.5})$$
$$m = m_w \text{ for } v_f/(\Delta T)^{0.5} > (0.26 (A_T / A_w)^{0.5} (H_1 / \Delta c_p)^{0.5}) \quad (H.11)$$

H.2.2.2 Openings on one wall only

H.2.2.2.1 Wind

$$m_w = 0.025 A v_f \quad (H.12)$$

H.2.2.2.2 Temperature differences with two openings

$$m_T = c_d A_T \quad (H.13)$$

where A_T is determined from equation (H.10)

H.2.2.2.3 Temperature difference with one opening

$$m_T = c_d A/3 (\Delta\theta g H_1 / T_m)^{0.5} \quad (H.14)$$

H.3 Example of calculation of natural ventilation rates for simple building

Consider a building consisting of a single undivided space 25m long, 10m wide and 8m high. Given that the building is situated in an open suburban area, calculate :

- a) the natural ventilation rate due to wind
- b) the ventilation rate due to the effect of a temperature difference of 6 °C in the absence of the wind.

There are no ventilation openings on shorter walls. On each of the longer walls there are openings of $2,5 \text{ m}^2$ at a low level, and $5,0 \text{ m}^2$ at high level, separated by a vertical distance of $6,0 \text{ m}$, and evenly distributed along the length of the wall.

a) Wind

H.3.1 Determination of the pressure coefficient difference

Building height ratio = $0,8$

Building plan ratio = $2,5$

Thus, from Table H.2, the difference in mean surface pressure coefficients at the two long sides of the building for a perpendicular wind is :

$$\Delta c_p = 0,7 - (-0,3) = 1,0$$

H.3.2 Determination of v_r

From Climatic data table $v_f = 3,75 \text{ m/s}$

terrain : country with scattered wind breaks :

From table H.3 $K = 0,52$; $a = 0,20$

Thus, from equation (H.8), using the building height of 8 m ,

$$v_r = 3,75 \cdot 0,52 \cdot (8)^{0,2} = 2,9 \text{ m/s}$$

H.3.3 Determination of A_w

From equation (H.7)

$$1/(A_w)^2 = 1/(5,0+2,5)^2 + 1/(5,0+2,5)^2$$

$$A_w = 5,3 \text{ m}^2$$

H.3.4 Determination of ventilation rate

Using equation H.6 ,

$$m_w = (0,61 \cdot 5,3 \cdot 2,9 \cdot 1)^{0,5} = 9,4 \text{ m}^3 / \text{s}$$

The volume of the building is $V = 25 \cdot 10 \cdot 8 = 2000 \text{ m}^3$

The air change rate is $3600 \cdot 9,4/2000 = 17 \text{ air changes/hour}$

D) Temperature difference

From the information given :

Temperature difference = 6°C

Height between openings $H_1 = 6,0\text{ m}$

Thus from equation (H.10)

$$1/(A_T)^2 = 1/((2,5+2,5)^2 + 1/(5,0+5,0)^2)$$

$$A_T = 4,5\text{ m}^2$$

Taking $T_m = 300\text{ K}$

$$m_T = 0,61 \cdot 4,4 \cdot (2 \cdot 9,8 \cdot 6,0 \cdot 6,0)^{0,5} = 4,2\text{ m}^3/\text{s}$$

The air change rate is :

$$3600 \cdot 4,2 / 2000 = 7,6\text{ air changes/hour.}$$

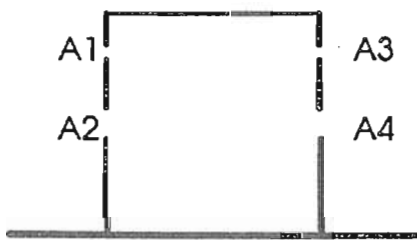


Figure H.1 Openings on two sides

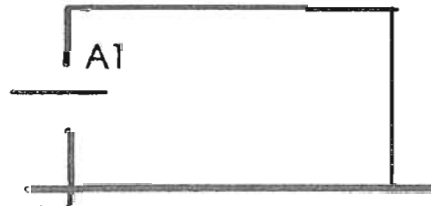


Figure H.2 Opening on one side

ANNEX L
BIBLIOGRAPHY
(Informative)

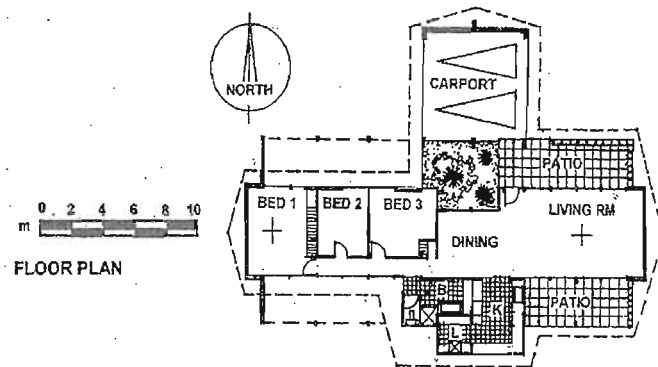
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APPENDIX D

Application of ANNEX H to the House in the Beltrame Thesis

Mr Beltrame's thesis in Mechanical Engineering at James Cook University was on the application of CHEETAH to determine indoor EnvT in the Townsville Energy Efficient Tropical House (TEETH). This building was designed by Macks & Robinson Pty Ltd, Architects of Townsville. The project was a joint effort among CSIRO, James Cook University, Townsville College of TAFE, and Macks & Robertson. The house was to have been constructed on the grounds of Townsville College of TAFE but this was not achieved. The plan of the single storey TEETH is shown below.



Beltrame used an air change rate of 2 to 5 air changes per hour in his parametric study of the influence of alternative construction materials on indoor EnvT. For a closed up building, the Beltrame study is probably reasonable. The Beltrame study is irrelevant for a free running naturally ventilated building as the air change rates used were far, far too low.

If we take the Living Room for example which is cross ventilated and use Annex H to estimate the air changes per hour for wind normal to the north facing wall with large sliding doors :

length (across wind) = 21 m; width (depth along flow) = 9.6 m, height = 4 m
 $h/w = 0.42$
 $l/w = 2.2$
 $\Delta c_p = (.7 - (-.6)) = 1.3$
 $K = 0.35$ urban terrain
 $a = 0.25$ urban terrain
 for a typical summer breeze of 7 m/s @ 10m height at nearby airport (January 3 pm)

$$v_r = 0.35 \times (3.5)^{0.25} \times 7$$

$$= 3.35 \text{ m/s}$$

Window area on each of North and South facing walls (louvres) 9.6 m^2

$$A_w = 1 / (1/9.6^2 + 1/9.6^2)^{0.5}$$

$$= 6.79 \text{ m}^2$$

$$M_w = c_d A_w v_r (\Delta c_p)^{0.5}$$

$$= 0.61 \times 6.79 \times 3.35 \times (1.3)^{0.5}$$

$$= 15.82 \text{ m}^3/\text{s}$$

$$\text{Volume of Living Room} = 6.080 \times 3.200 \times 4.600 = 89.5 \text{ m}^3$$

$$\text{Air Changes per hour} = 3600 \times 15.82 / 89.5 = \mathbf{636.3 \text{ ac/h}}$$
 (with long wall facing the breeze)

If we take the Living Room for example which is cross ventilated and use Annex H to estimate the air changes per hour for wind onto the blank end East facing wall:

length (across wind) = 9.6 m, width (depth along flow) = 21 m, height = 4 m

$$h/w = 0.2$$

$$l/w = 0.46$$

$$\Delta c_p = (-0.2 - (-0.5)) = 0.3$$

$$K = 0.35 \text{ urban terrain}$$

$$a = 0.25 \text{ urban terrain}$$

for a typical summer breeze of 7 m/s @ 10m height at nearby airport

$$v_r = 0.35 \times (3.5)^{0.25} \times 7$$

$$= 3.35 \text{ m/s}$$

Window area on each of North and South facing walls (louvres) 9.6 m²

$$A_w = 1 / (1/9.6^2 + 1/9.6^2)^{0.5}$$

$$= 6.79 \text{ m}^2$$

$$Mw = c_d A_w v_r (\Delta c_p)^{0.5}$$

$$= 0.61 \times 6.79 \times 3.35 \times (0.3)^{0.5}$$

$$= 7.63 \text{ m}^3/\text{s}$$

$$\text{Volume of Living Room} = 6.080 \times 3.200 \times 4.600 = 89.5 \text{ m}^3$$

Air Changes per hour = $3600 \times 7.63 / 89.5 = 306.9 \text{ ac/h}$ (with blank end wall facing the wind)

These air change rates compare favourably with measurements from boundary layer wind tunnel studies of the TEETH (Aynsley, 1996). With these large air change rates the temperature of the air flowing through the building will be for practical purposes the same as outdoor air temperature. In these situations the thermal comfort of occupants at a typical metabolism and with typical clothing will be determined by the ambient air temperature and humidity, the rate of air movement, and radiant heat gain or loss to indoor surfaces. The latter will depend on indoor surface temperatures. In the case of housing solar gain through a poorly insulated roof is likely to be the most common source of radiant gain to occupants. The minimum insulation required to ensure ceiling surface temperature does not exceed 38°C can be calculated in steady state mode by setting indoor air temperature to outdoor air temperature, the outer roof surface to sol-air temperature. After assigning a suitable value to the indoor air film the total resistance to satisfy steady state heat transfer can be calculated.

Reference

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