Class 2

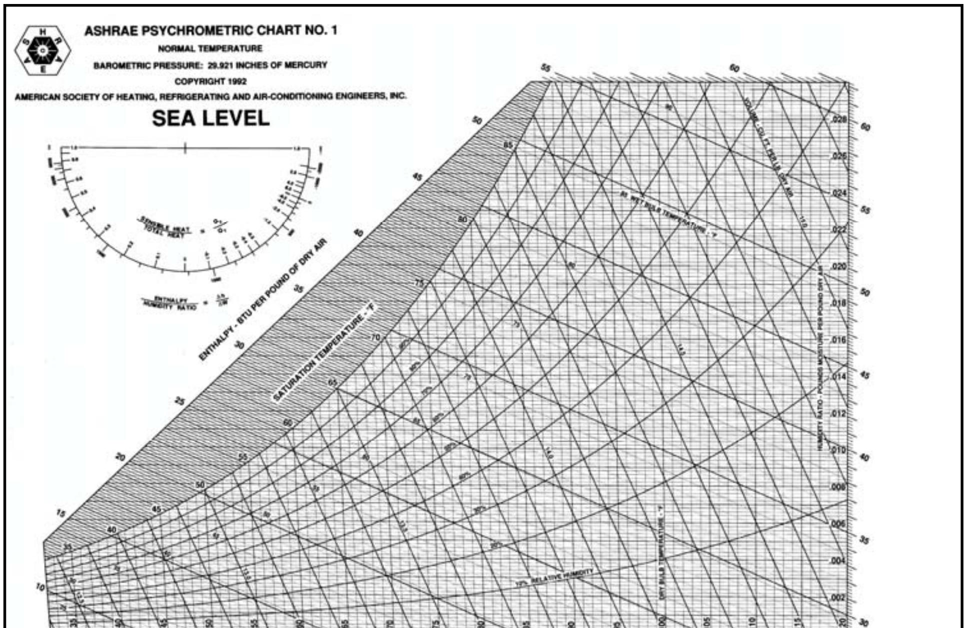
Psychrometrics

Comfort

Case Study

Assignment 2

Psychrometric Chart – properties of moist air



Construction of the psychrometric chart

Drybulb (T_{db}) [ºF]

Saturation temperature (T_{sat}) [ºF]

Humidity ratio (W) [$lbs_{water}/lbs_{dry\ air}$]

“iso”: follows constant line

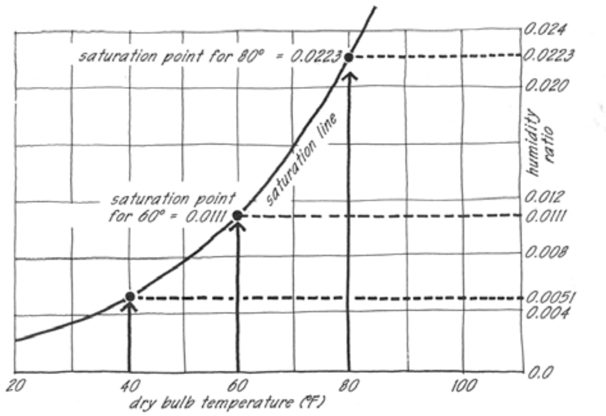
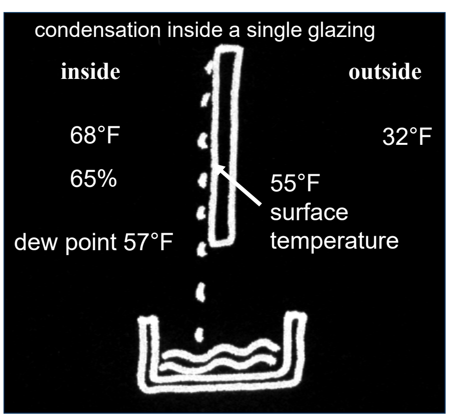


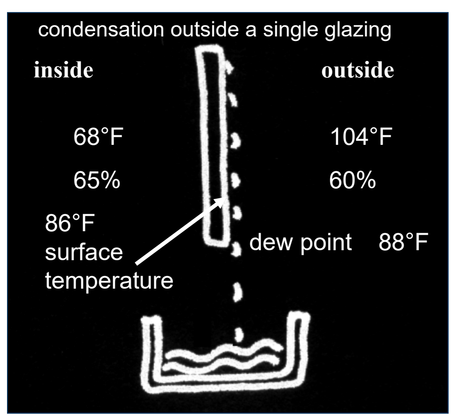
Figure 1.25: Construction of the psychrometric chart: air saturation as a function of temperature.

Construction of the psychrometric chart

Condensation

Water phase change from vapor to liquid on surface below the dew point.





Construction of the psychrometric chart

Relative
Humidity (RH) [%]

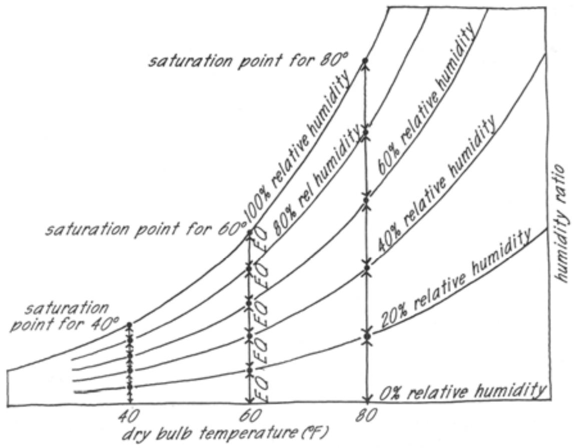


Figure 1.26: Construction of the psychrometric chart: relative humidity curves created by dividing vertical dry-bulb temperature lines into equal parts below the saturation line.

Construction of the psychrometric chart

Wetbulb
temperature (T_{wb}) [ºF]

The wet-bulb temperature is the temperature a parcel of air would have if it were cooled to saturation (100% relative humidity) by the evaporation of water into it, with the latent heat being supplied by the parcel.

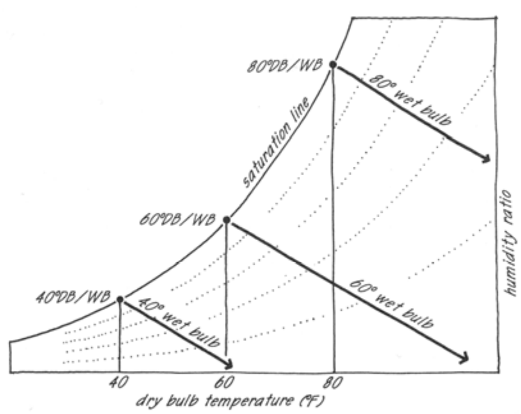


Figure 1.27: Construction of the psychrometric chart: At saturation, wet-bulb temperature equals dry-bulb temperature; to maintain constant wet-bulb temperature, the humidity ratio must be lowered while dry-bulb temperature increases. Constant wet-bulb lines extend down and to the right from the saturation line origin.

Construction of the psychrometric chart

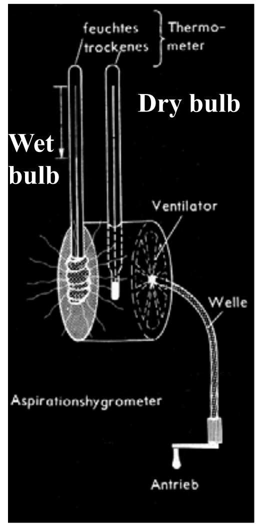
Dry bulb + wet bulb temperature

Dry bulb temperature

Temperature of a gas or mixture of gases indicated by an accurate thermometer after correction for radiation.

Wet bulb temperature

Temperature, at which water, by evaporating into air, can bring the air to saturation adiabatically at the same temperature.



Construction of the psychrometric chart

Enthalpy (i) [btu/lbs$_{air}$]

Enthalpy is basically **defined** as the Net energy possessed by the system at any given point of time.

Enthalpy - h - (kJ/kg) of moist air is defined as the total enthalpy of the dry air and the water vapor mixture per kg of moist air, includes the

• enthalpy of the dry air - the sensible heat - and

• the enthalpy of the evaporated water - the latent heat

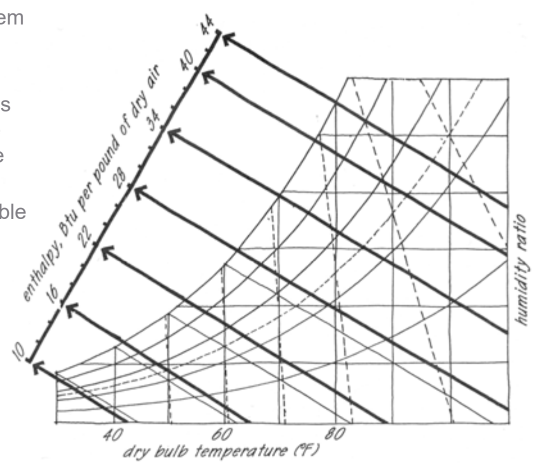
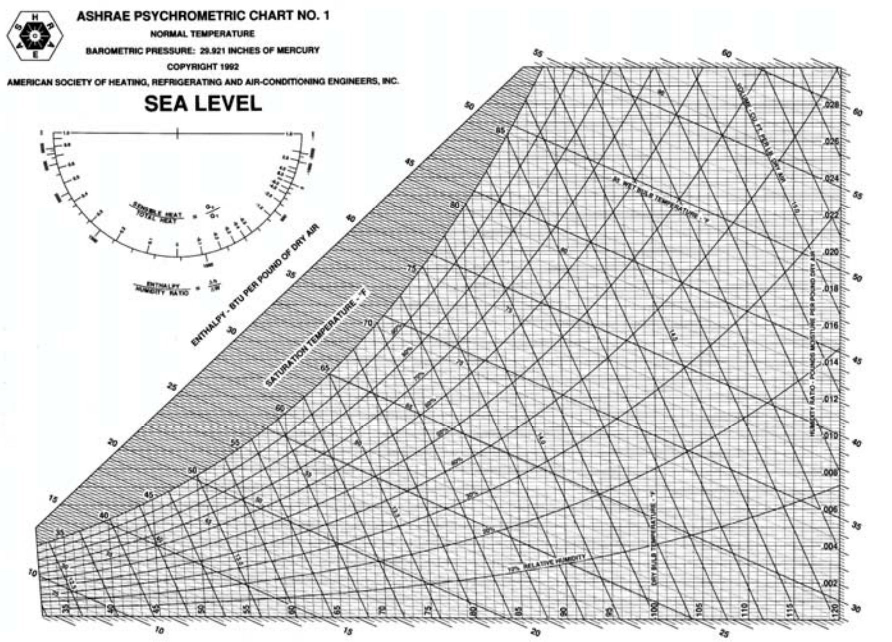


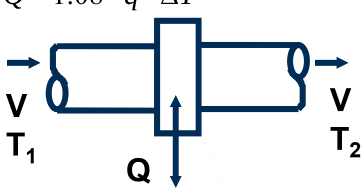
Figure 1.28: Enthalpy lines (which coincide with wet-bulb temperature lines) projected to the enthalpy scale outside of the chart.



Typical air conditioning processes

Sensible heating
& cooling

$$Q = \rho * c_p * V * \Delta T = \rho * V * \Delta i$$
$$Q = 1.08 * q * \Delta T$$



Q=sensible heat (Btu/hr)
q=air flow rate (ft3/min)
ΔT=temperature change (deg F)
Cp=specifc heat Btu/lbm deg F
V= flow rate ft3/hr
Δi=enthalpy of condensation btu/lbm

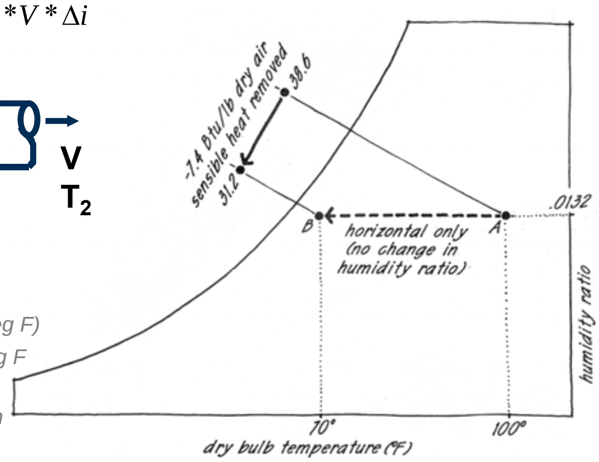


Figure 1.31: Determining change in enthalpy and sensible heat due solely to changing dry-bulb temperature (horizontal-only movement on the chart).

Example 1

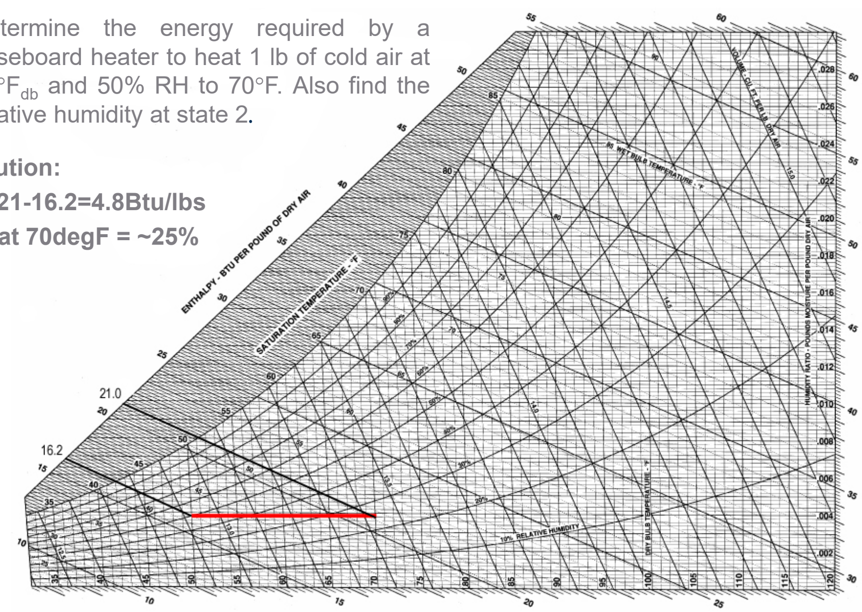
Determine the energy required by a baseboard heater to heat 1 lb of cold air at 50°F_{db} and 50% RH to 70°F. Also find the relative humidity at state 2.

Determine the energy required by a baseboard heater to heat 1 lb of cold air at 50°F_{db} and 50% RH to 70°F. Also find the relative humidity at state 2.

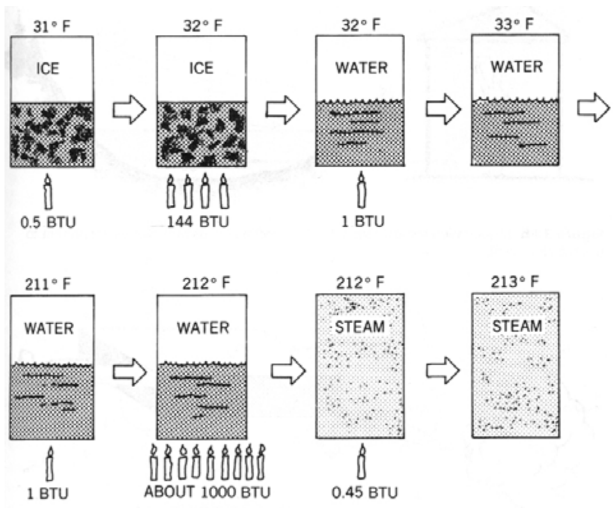
Solution:

Q= 21-16.2=4.8Btu/lbs

RH at 70degF = ~25%



Latent and Sensible Heat

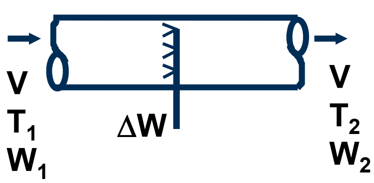


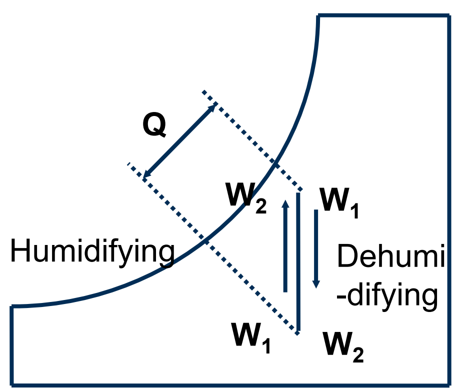
Typical air conditioning processes

Humidification
& dehumidification

$$Q = \rho * V * i_s * \Delta W = \rho * V * \Delta i$$

i_s=1050btu/lbs (enthalpy of condensation of water @ 75ºF)





Example 2

Determine the energy required by a spray humidifier to humidify 1 lb of dry air at 10% RH and 70°F$_{db}$ to 50% RH using 70°F water (assume there is no change in drybulb temperature).

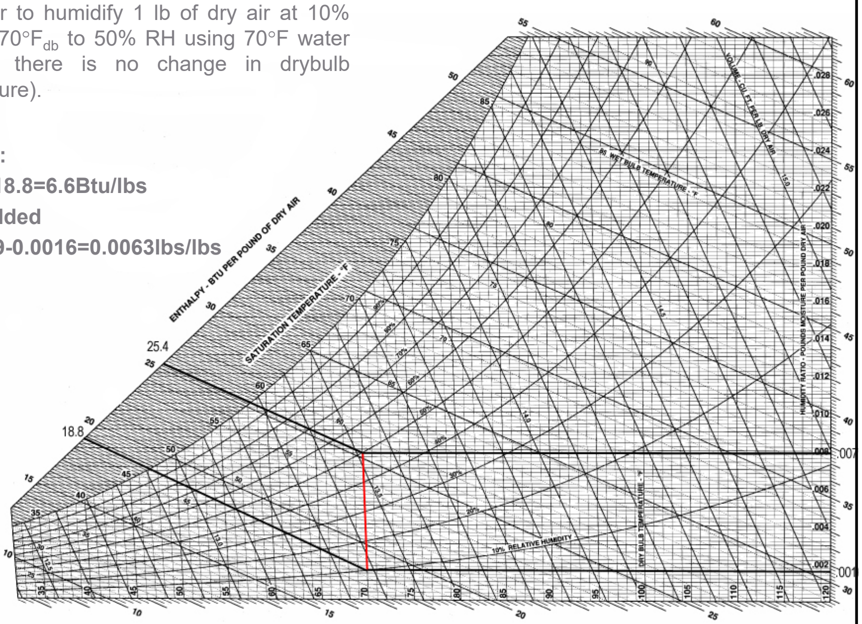
Determine the energy required by a spray humidifier to humidify 1 lb of dry air at 10% RH and 70°F$_{db}$ to 50% RH using 70°F water (assume there is no change in drybulb temperature).

Solution:

Q=25.4-18.8=6.6Btu/lbs

Water added

h=0.0079-0.0016=0.0063lbs/lbs



Typical air conditioning processes

Cooling & dehumidification

$$Q = \rho * V * c_p * \Delta T + \rho * V * i_s * \Delta W = \rho * V * \Delta i$$

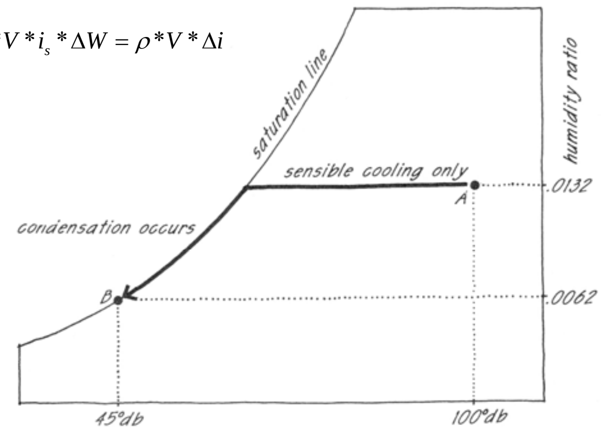


Figure 1.35: Dehumidification is achieved by cooling the air below the dew point temperature causing condensation.

Example 3

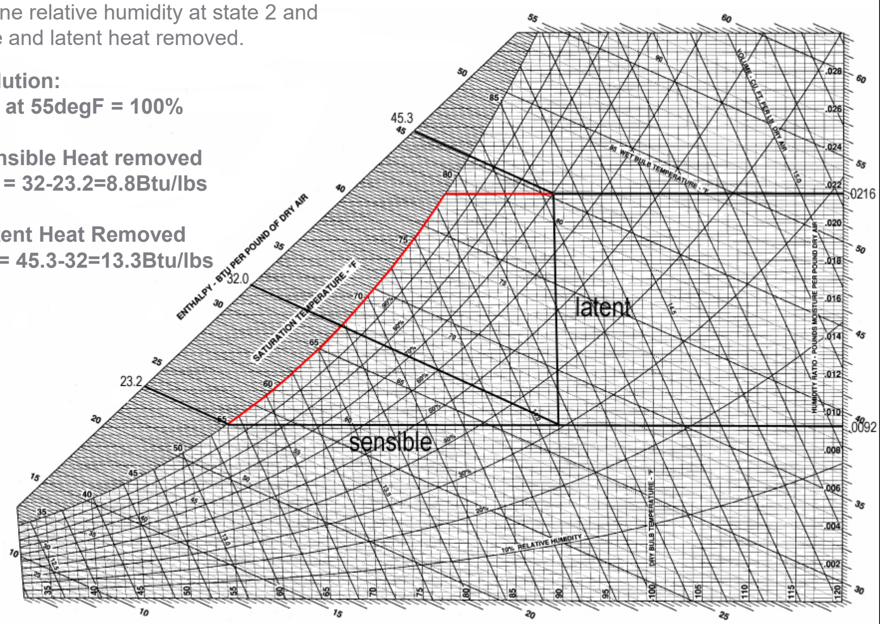
An air conditioner is used to cool 1 lb of warm air at 70% RH and 90°F_{db} to 55°F_{db}. Determine relative humidity at state 2 and sensible and latent heat removed.

An air conditioner is used to cool 1 lb of warm air at 70% RH and 90°F_{db} to 55°F_{db}.
Determine relative humidity at state 2 and sensible and latent heat removed.

Solution:
RH at 55degF = 100%

Sensible Heat removed
Qs = 32-23.2=8.8Btu/lbs

Latent Heat Removed
Ql = 45.3-32=13.3Btu/lbs



Typical air conditioning processes

Adiabatic humidification

$$Q = \rho * V * c_p * \Delta T + \rho * V * i_s * \Delta W = \rho * V * \Delta i$$

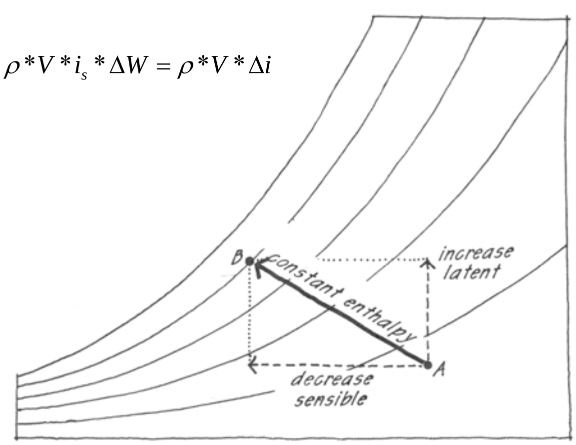


Figure 1.33: Evaporative cooling, a constant-enthalpy process achieved by exchanging latent and sensible heat, occurs along the wet-bulb/enthalpy lines.

Typical air conditioning processes

Desiccant dehumidification

$$Q = \rho * V * c_p * \Delta T + \rho * V * i_s * \Delta W = \rho * V * \Delta i$$

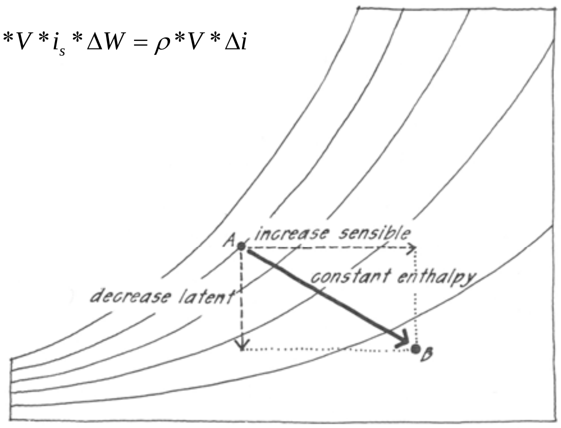


Figure 1.34: Desiccant dehumidification — a constant-enthalpy process achieved by exchanging latent and sensible heat.

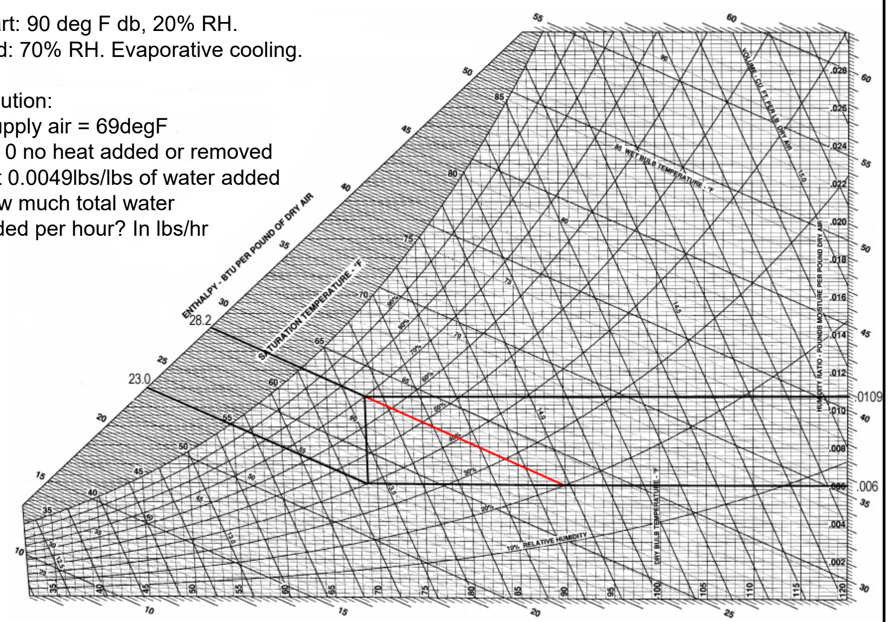
Example 4

In Phoenix, it is possible to use evaporative cooling in summer. A building receives a supply of 1000cfm of outside air to meet its cooling demand. The outdoor air is at 90^{o}F$_{db}$ and 20% relative humidity. If the relative humidity is allowed to be increased to 70% by evaporative cooling, determine the dry bulb temperature of the supply air and how much sensible or latent heat is added to the air.

Note: $$\frac{Q}{time} = \rho \frac{V}{time} C_p \Delta T$$ e.g. btu/h, ft^3/h

Start: 90 deg F db, 20% RH.
End: 70% RH. Evaporative cooling.

Solution:
Tsupply air = 69degF
Q= 0 no heat added or removed
But 0.0049lbs/lbs of water added
How much total water
added per hour? In lbs/hr



Example 4 continued

Total Water added per hour?

How many pounds of dry air in 1000cfm?
Density of air $\rho_{(air)}$=0.075lb/ft^3

0.075lb/ft^3 * 1000ft^3/minute=75lb/minute =
75*60lb/hour=4,500lb/hour

W1=0.006 lb_w/lb_a
W2=0.0109 lb_w/lb_a
Change in water = W2-W1=0.0049 lb_w/lb_a

Amount of water added = 0.0049lb_w/lb_a *
4,500lb_a/hour = **22.05lb_w/hour**

Comfort

Site

Source

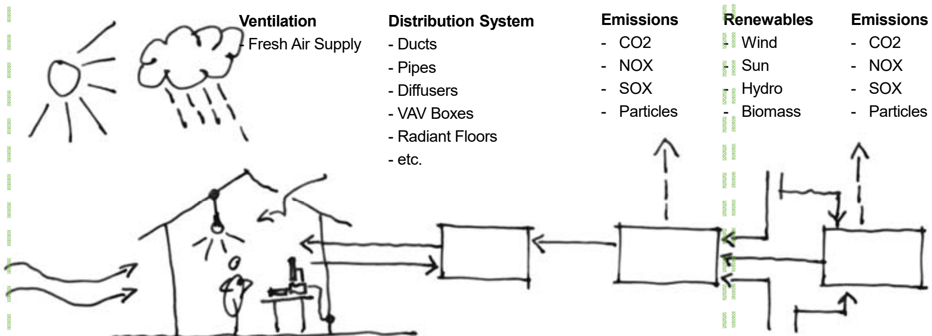
Ventilation
- Fresh Air Supply

Distribution System
- Ducts
- Pipes
- Diffusers
- VAV Boxes
- Radiant Floors
- etc.

Emissions
- CO_2
- NOX
- SOX
- Particles

Renewables
- Wind
- Sun
- Hydro
- Biomass

Emissions
- CO_2
- NOX
- SOX
- Particles



Climate
- Air Temperature
- Solar Radiation
- Rain
- Wind
- etc.

Individual
- Clothing
- Activity level
- Comfort Criteria

Enclosure
- Walls
- Windows
- Roof

Conditioning Systems
- Chillers
- Heat Pumps
- Air Handling Units
- Fans
- Pumps
- etc.

Energy Systems
- Boilers
- Transformers
- Cooling Towers
- Generators
- PV's
- Windturbines

Fossil Fuels
- Gas
- Oil
- Diesel
- Coal

Utility Infrastructure
- Powerplants
- Steam Plants
- District Energy Plants

Comfort Criteria

Individual
- Clothing
- Activity level
- Comfort Criteria

Thermal comfort

The Building as Shelter

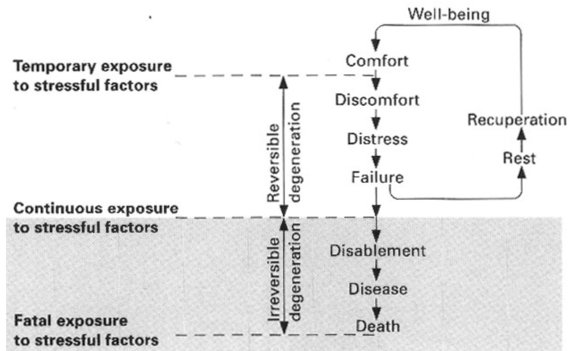
Environmental stress can range from discomfort to failure of the human body.

Comfort is subjective while failure is objective.

Buildings are designed to meet comfort criteria.

The primary comfort factors are:

- Thermal comfort
- Air Quality
- Visual Comfort
- Acoustic Comfort



Fitch, American Building, Page 25

Thermal comfort

Human heat balance

The heat produced by the metabolism should be equal to the amount of heat lost from the body

The comfort equation (energy balance):

$$M - W = H + E + C_{res} + E_{res}$$

- M = Metabolic rate
- W = Work delivered
- H = Dry heat loss by convection and radiation
- E = Evaporative heat exchange at skin
- C_{res} = Respiratory convective heat exchange
- E_{res} = Respiratory evaporative heat exchange

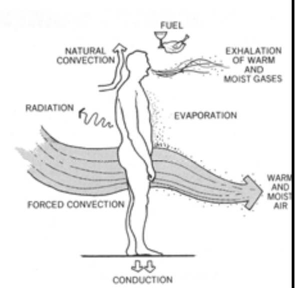


Figure 4.1b Methods of dissipating waste heat from a biological machine.

Thermal comfort

Human body temperature: 98.6ºF

A person will wake up from sleep if s/he loses 80 Btu/h or more. This will cause a decrease in skin temperature of 5ºF.

A person will feel uncomfortable or sick, if his/her body temperature is increased by 2ºF.

When T_{body} > 110ºF …DEATH!

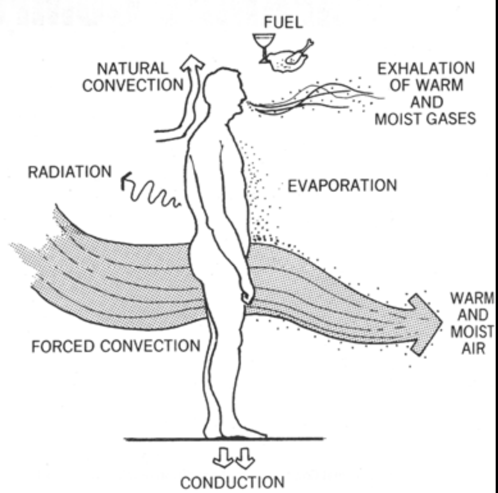


Figure 4.1b Methods of dissipating waste heat from a biological machine.

Thermal comfort parameters

Dry bulb temperature
Relative humidity or other measure of moisture
Local air velocity
Mean radiant temperature

Clothing level
Activity level

ASHRAE Standard 55: Thermal environmental conditions for human occupancy
ISO 7730: Moderate thermal environments - determination of the PMV and PPD indices and specification of the conditions for thermal comfort

Body heat loss mechanisms

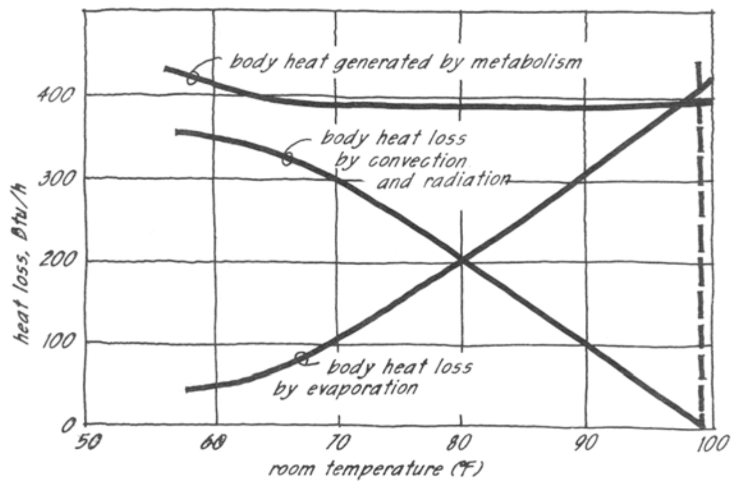


Table 7 Typical Insulation and Permeation Efficiency Values for Clothing Ensembles

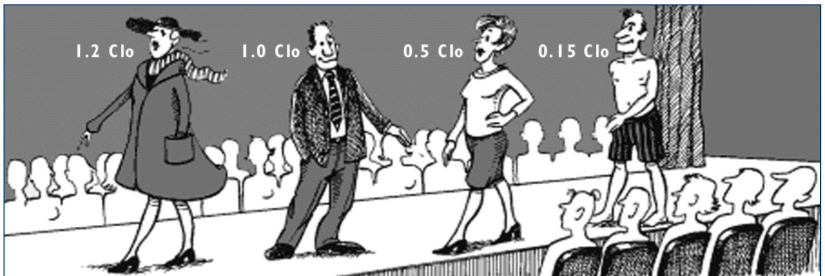
Ensemble Description [a]	I_{cl} (clo)	I_t[b] (clo)	f_{cl}	i_{cl}	i_m[b]
Walking shorts, short-sleeved shirt	0.36	1.02	1.10	0.34	0.42
Trousers, short-sleeved shirt	0.57	1.20	1.15	0.36	0.43
Trousers, long-sleeved shirt	0.61	1.21	1.20	0.41	0.45
Same as above, plus suit jacket	0.96	1.54	1.23		
Same as above, plus vest and T-shirt	1.14	1.69	1.32	0.32	0.37
Trousers, long-sleeved shirt, long-sleeved sweater, T-shirt	1.01	1.56	1.28		
Same as above, plus suit jacket and long underwear bottoms	1.30	1.83	1.33		
Sweat pants, sweat shirt	0.74	1.35	1.19	0.41	0.45
Long-sleeved pajama top, long pajama trousers, short 3/4 sleeved robe, slippers (no socks)	0.96	1.50	1.32	0.37	0.41
Knee-length skirt, short-sleeved shirt, panty hose, sandals	0.54	1.10	1.26		
Knee-length skirt, long-sleeved shirt, full slip, panty hose	0.67	1.22	1.29		
Knee-length skirt, long-sleeved shirt, half slip, panty hose, long-sleeved sweater	1.10	1.59	1.46		
Same as above, replace sweater with suit jacket	1.04	1.60	1.30	0.35	0.40
Ankle-length skirt, long-sleeved shirt, suit jacket, panty hose	1.10	1.59	1.46		
Long-sleeved coveralls, T-shirt	0.72	1.30	1.23		
Overalls, long-sleeved shirt, T-shirt	0.89	1.46	1.27	0.35	0.40
Insulated coveralls, long-sleeved thermal underwear, long underwear bottoms	1.37	1.94	1.26	0.35	0.39

Sources: McCullough and Jones (1984) and McCullough et al. (1989).
[a]All ensembles include shoes and briefs or panties. All ensembles except those with panty hose include socks unless otherwise noted.
[b]For $\bar{t}_r = t_a$ and air velocity less than 40 fpm (I_a = 0.72 clo and i_m = 0.48 when nude).

Table 4 Typical Metabolic Heat Generation for Various Activities

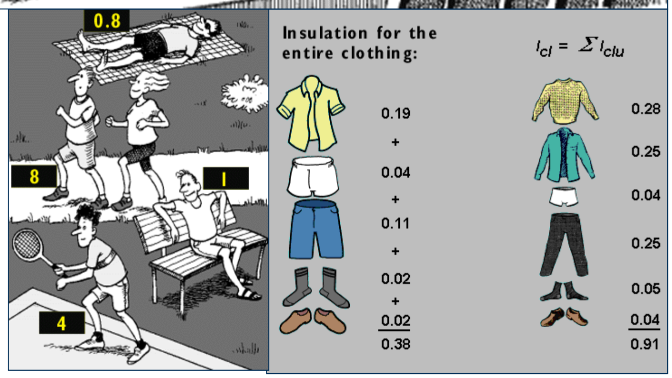
	Btu/h·ft²	met[a]
Resting		
Sleeping	13	0.7
Reclining	15	0.8
Seated, quiet	18	1.0
Standing, relaxed	22	1.2
Walking (on level surface)		
2.9 fps (2 mph)	37	2.0
4.4 fps (3 mph)	48	2.6
5.9 fps (4 mph)	70	3.8
Office Activities		
Reading, seated	18	1.0
Writing	18	1.0
Typing	20	1.1
Filing, seated	22	1.2
Filing, standing	26	1.4
Walking about	31	1.7
Lifting/packing	39	2.1
Driving/Flying		
Car	18 to 37	1.0 to 2.0
Aircraft, routine	22	1.2
Aircraft, instrument landing	33	1.8
Aircraft, combat	44	2.4
Heavy vehicle	59	3.2
Miscellaneous Occupational Activities		
Cooking	29 to 37	1.6 to 2.0
Housecleaning	37 to 63	2.0 to 3.4
Seated, heavy limb movement	41	2.2
Machine work		
sawing (table saw)	33	1.8
light (electrical industry)	37 to 44	2.0 to 2.4
heavy	74	4.0
Handling 110 lb bags	74	4.0
Pick and shovel work	74 to 88	4.0 to 4.8
Miscellaneous Leisure Activities		
Dancing, social	44 to 81	2.4 to 4.4
Calisthenics/exercise	55 to 74	3.0 to 4.0
Tennis, singles	66 to 74	3.6 to 4.0
Basketball	90 to 140	5.0 to 7.6
Wrestling, competitive	130 to 160	7.0 to 8.7

Sources: Compiled from various sources. For additional information, see Buskirk (1960), Passmore and Durnin (1967), and Webb (1964).
[a]1 met = 18.4 Btu/h·ft²



1clo= 0.880 [ºF*ft^2*h/btu] (the average adult has 19.6ft^2 of surface area)

1 met=18.4 [btu/h*ft^2]



Definition

MRT [T_{mrt}]
Area weighted average of all surface temperatures

$$T_{mrt} = \frac{\sum_i A_i \times T_i}{\sum_i A_i}$$

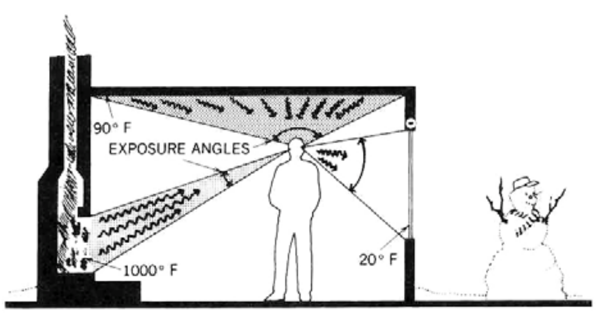


Figure 3.12 The mean radiant temperature (MRT) at any point is a result of the combined effect of a surface's temperature and angle of exposure.

Definition

Operative Temperature [T_{op}]

The combined air and surface temperature are often used as one parameter for the thermal condition of a space.

$$T_{op} = aT_{air} + (1-a)T_{mrt}$$

	0-40fpm	40-120fpm	120-200fpm
a	0.5	0.6	0.7

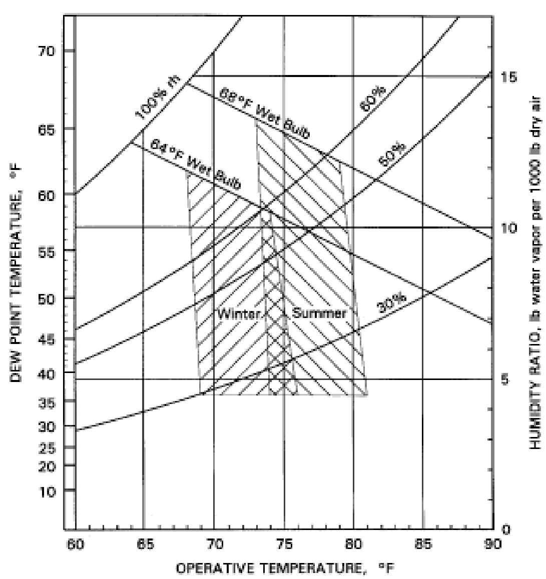
For low velocities: $$T_{op} = \frac{T_{air} + T_{mrt}}{2}$$

ASHRAE comfort zone

Effective temperature: the temperature at 50% RH that yields the same total heat loss from the skin as for the actual environment

Operative temperature: the weighted average of MRT and air temperature based on heat transfer coefficients

1.2 met; 0.5 clo (summer), 1.0 clo (winter)



Definition

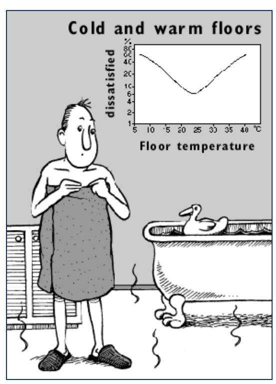
Wind speed in buildings
(source squ1.com)

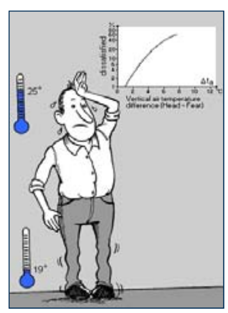
DESCRIPTION	m/s	km/h	fpm	mph
Still	0	0	0	0.0
Not Noticeable	0.1	0.4	20	0.2
Barely Noticeable	0.3	1	60	0.7
Pleasant Breeze	0.5	1.8	100	1.1
Light Breeze	0.7	2.5	140	1.6
Hair and Papers Move	1	4	200	2.2
Noticeably Draughty	1.4	5	280	3.1
Unpleasant Breeze	1.7	6	330	3.8
Gusting Breeze	2.0+	6.5+	330+	3.8+

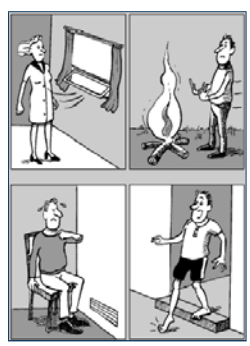
Wind speed outside
(source windows.ucar.edu)

Beaufort number	Wind speed (km/hr)	International description	US Weather Bureau description	Effect of wind on the Sea
0	<1	Calm	Light Wind	Small wavelets
1	1-5	Light Air	Light Wind	Small wavelets
2	5-11	Light Breeze	Light Wind	Small wavelets
3	11-12	Gentle Breeze	Gentle-moderate	Large wavelets to small waves
4	20-28	Moderate Breeze	Gentle-moderate	Large wavelets to small waves
5	29-38	Fresh Breeze	Fresh wind	Moderate waves, many whitecaps
6	39-49	Strong gale	Strong wind	Large waves, many whitecaps
7	50-61	Fresh Breeze	Strong wind	Large waves, many whitecaps
8	62-74	Fresh gale	Gale	High waves, foam streaks
9	75-88	Stong gale	Gale	High waves, foam streaks
10	89-102	Whole gale	Whole gale	Very high waves, rolling sea
11	103-117	Storm	Whole gale	Very high waves, rolling sea
12-17	>117	Hurricane	Hurricane	Sea white with spray and foam

Other conditions for thermal comfort







Vertical temperature difference
- From 4”-67” (occupied zone), air temperature difference shall not exceed 5ºF

Radiant temperature asymmetry
- Vertical asymmetry shall not exceed 5ºF
- Horizontal asymmetry shall not exceed 10ºF

Floor temperature shall be between 65ºF-84ºF

Drafts shall be kept low

Increased air speed

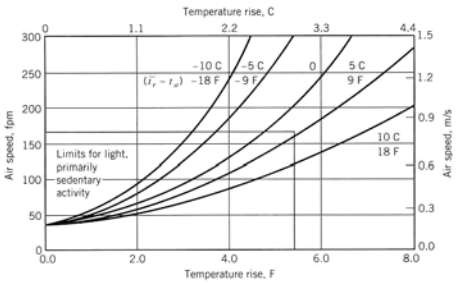


Figure 4-9 Air speed required to offset increased temperature. (Reprinted by permission from ASHRAE Standard 55-92, 1992)

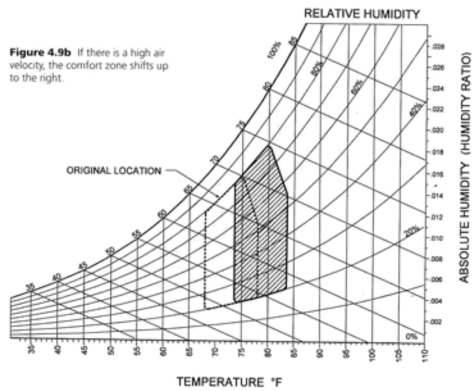


Figure 4.9b If there is a high air velocity, the comfort zone shifts up to the right.

Discomfort

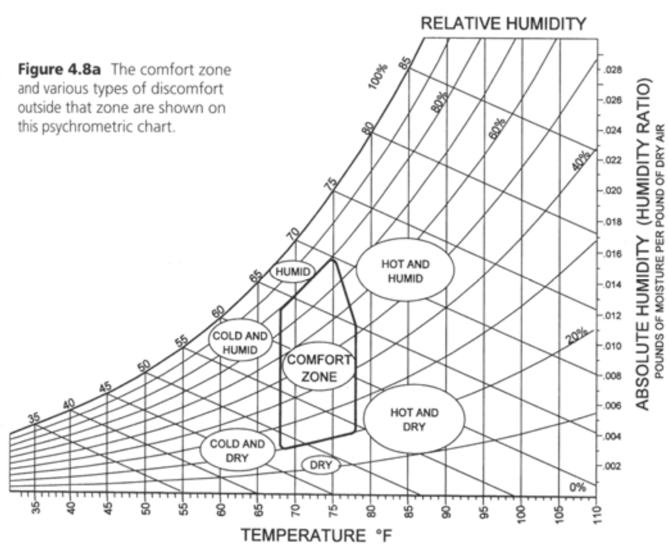
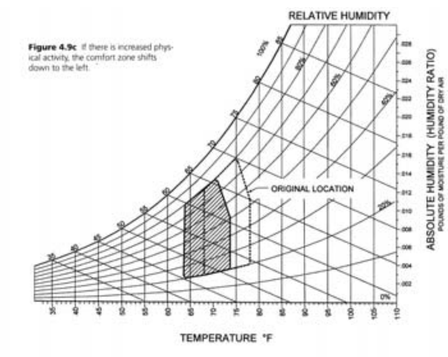
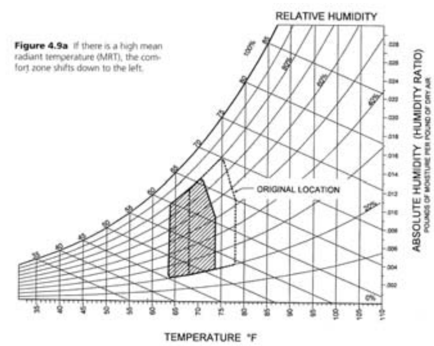


Figure 4.8a The comfort zone and various types of discomfort outside that zone are shown on this psychrometric chart.

Shifted comfort range by activity or space conditions

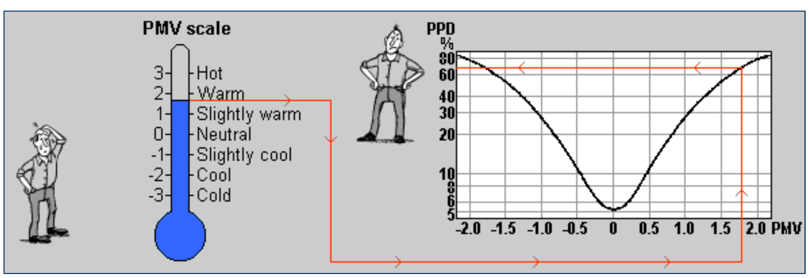


Increased
activities need
cooler spaces



High MRT,
cooler air temp
is needed.

Evaluating Comfort: Predicted Mean Vote & Predicted Percentage of Dissatisfied (Fanger)



PMV ... Predicted mean vote -3 to 3

PPD ... Percentage of dissatisfied People 0 to 100%

No direct measurement value, but personal rating of ~ 1000 test persons in a certain test room environment.

Evaluating Comfort: Predicted Mean Vote & Predicted Percentage of Dissatisfied (Fanger)

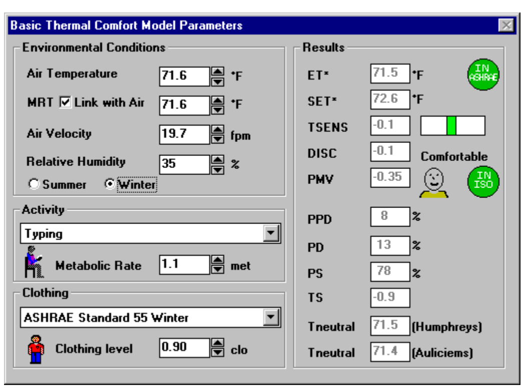
```
pmv1=(.352*EXP(-.036*met)+.032)*(.86*met-.35*(43-.052*met-.0075*pa)-.42*(.86*met-50)-
    .002*met*(44-.0075*pa)-.0012*met*(34-tluft+273.15)-3.4e-8*fkl*(tkleid**4-trad**4-
    irad/5.67e-8)-.86*fkl*alfa*(tkleid-tluft) )
```

met(sadu)	activity
pa	humidity level
Tluft	tair
fkl	clothing factor
tkleid	surface temperature of cloths
trad	mean radiative temperature
irad	total shortwave radiation
alfa	heat transfer coefficient depening on air velocity

```
pmv=MIN(ABS(pmv1),3)*pmv1/ABS(pmv1)
ppd=100-95*EXP(-pmv**2 / (4.1-0.6*pmv))
```

Evaluating Comfort: Predicted Mean Vote & Predicted Percentage of Dissatisfied (Fanger)

Modelling Thermal Comfort



Asymmetries as a function of PPD

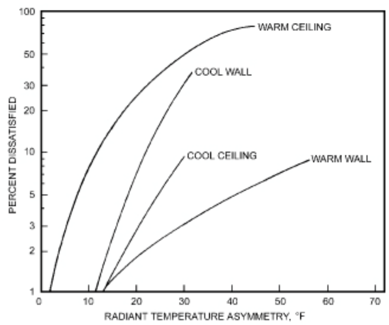


Fig. 6 Percentage of People Expressing Discomfort due to Asymmetric Radiation

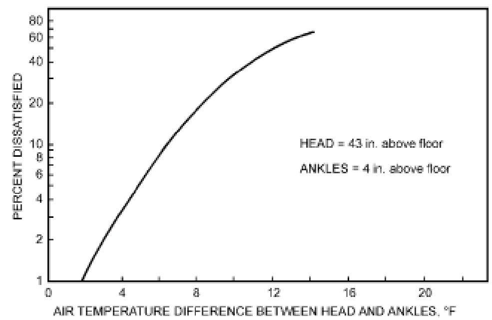
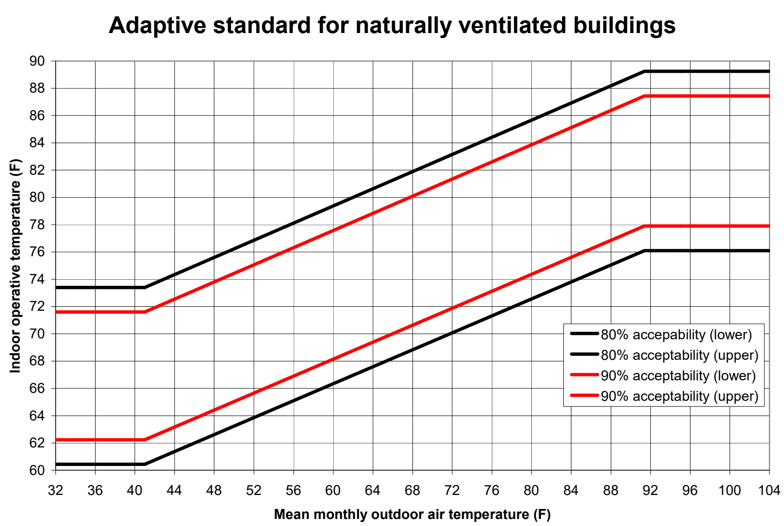


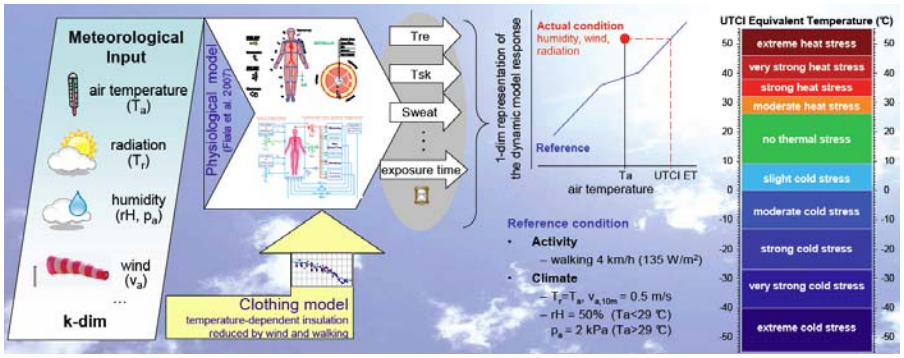
Fig. 9 Percentage of Seated People Dissatisfied as Function of Air Temperature Difference Between Head and Ankles

Adaptive thermal comfort (ASHRAE 55)

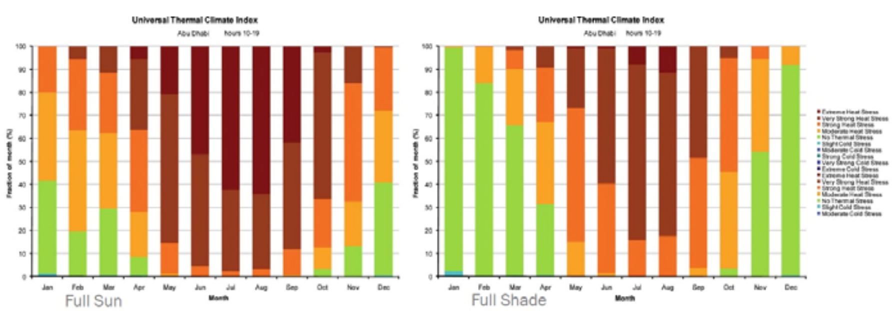


Universal Thermal Comfort Index

http://www.utci.org/utci_poster.pdf



Universal Thermal Comfort Index



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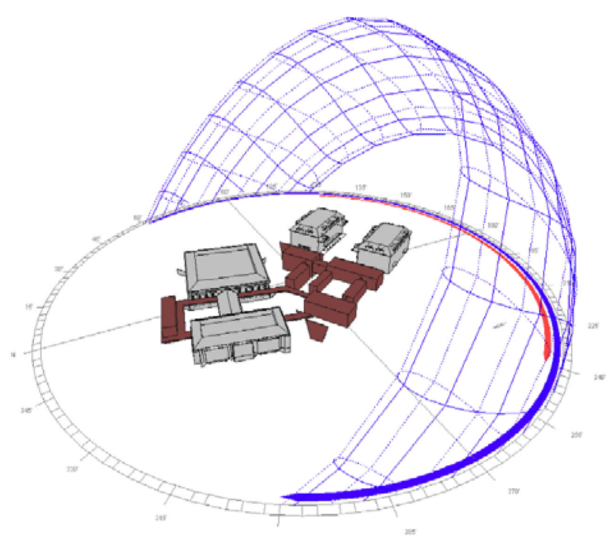
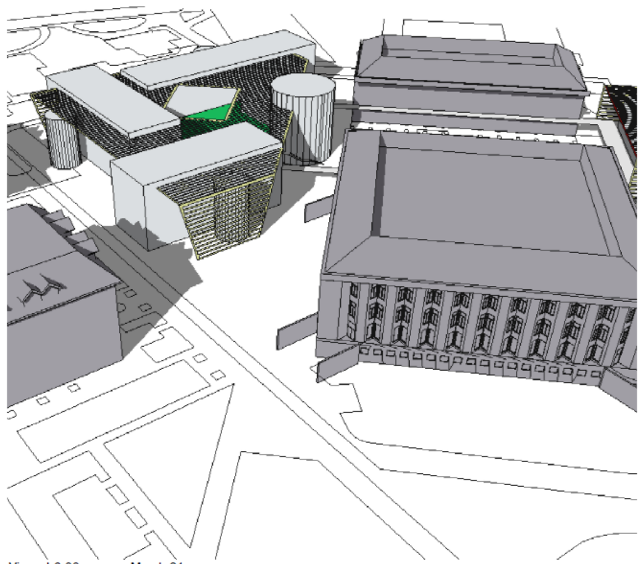
Ennead Architects

Natural Ventilation Study – Stanford University

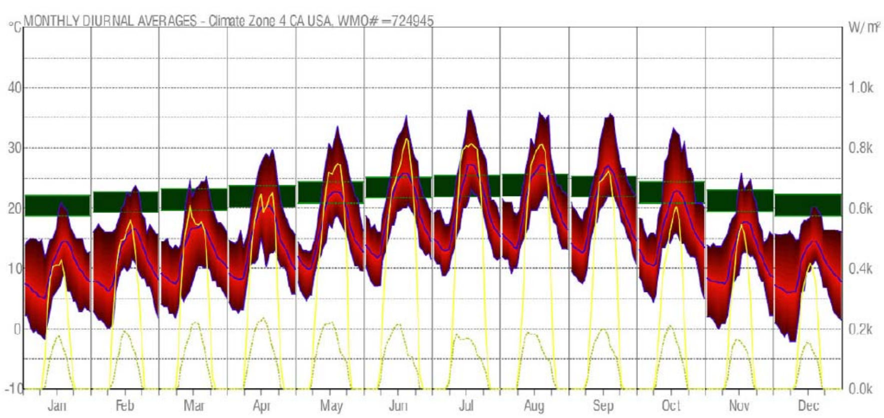


Natural Ventilation Study – Stanford University



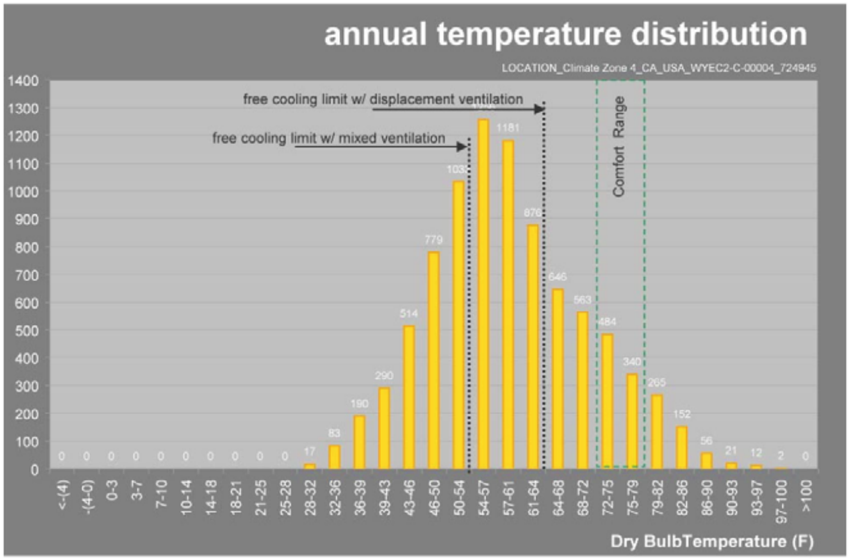
View at 9:00 a.m. on March 21

Natural Ventilation Study – Stanford University



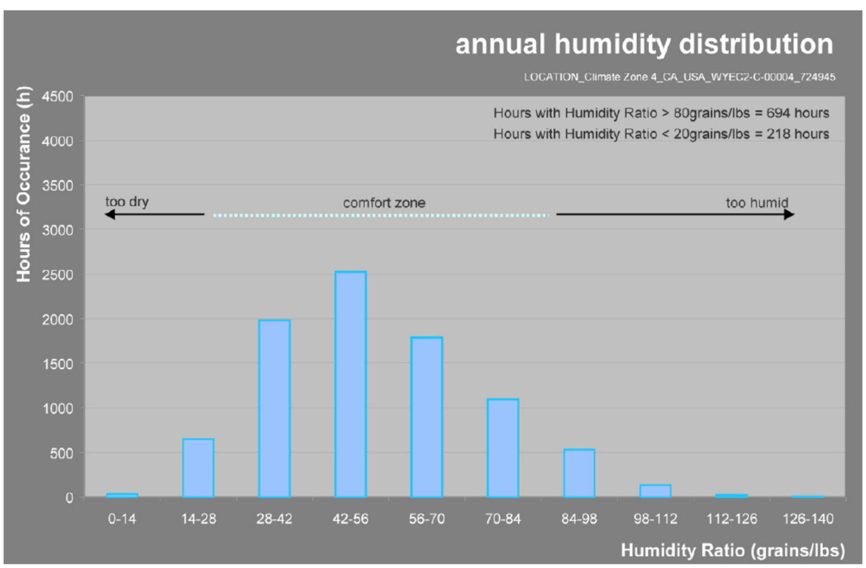
Monthy temperature fluctuations

Natural Ventilation Study – Stanford University



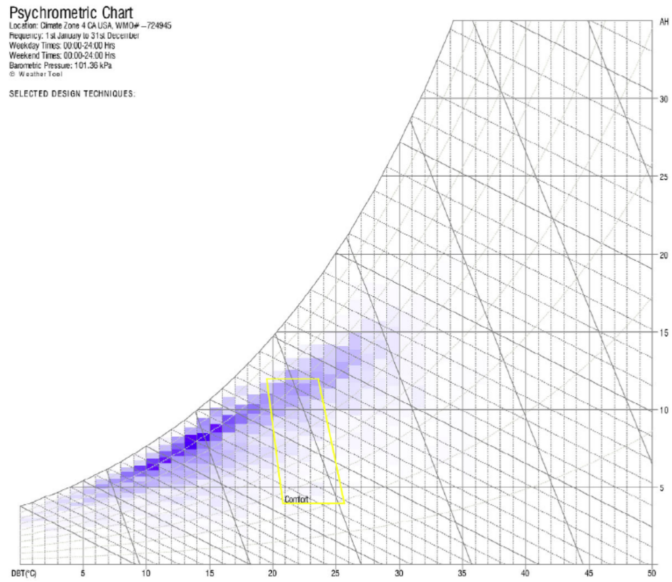
Annual temperature distribution

Natural Ventilation Study – Stanford University

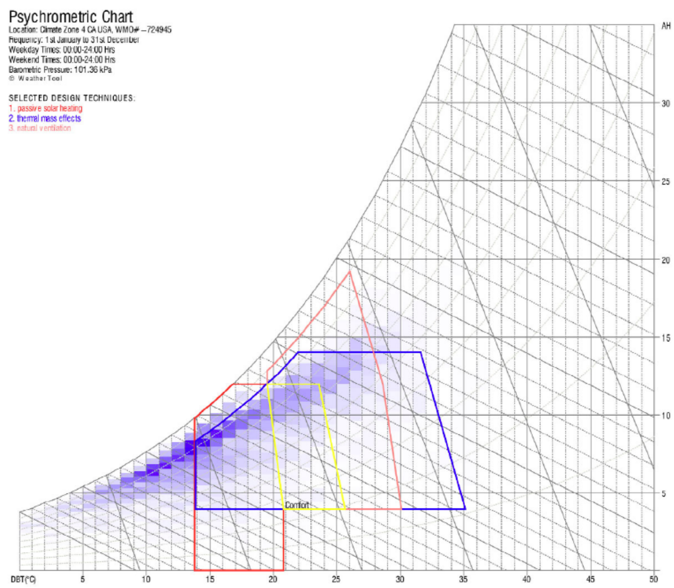


Annual humidity distribution

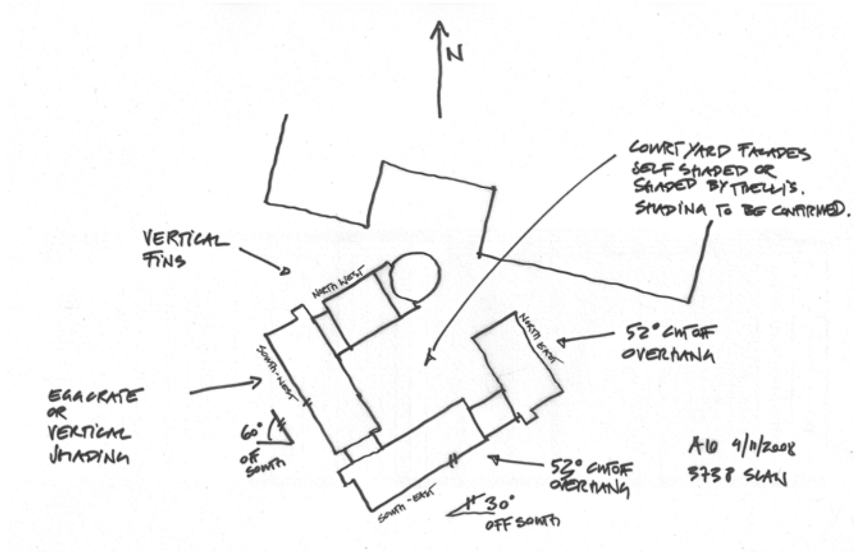
Natural Ventilation Study – Stanford University



Natural Ventilation Study – Stanford University



Natural Ventilation Study – Stanford University



Natural Ventilation Study – Stanford University

These are the assumptions we modeled:

NE Facing Façade:
Type: Horizontal Overhang
Depth: 2'
Spacing: 2' 7"

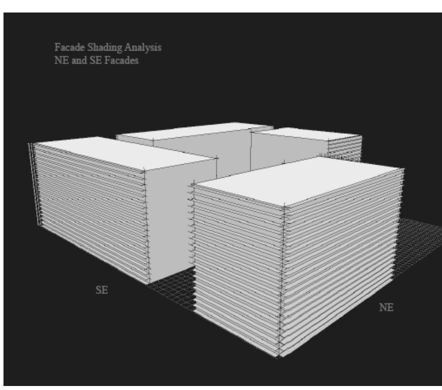
SE Facing Façade:
Type: Horizontal Overhang
Depth: 2'
Spacing: 2' 8"

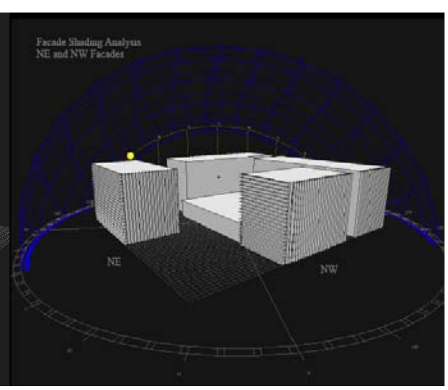
SW Facing Façade:
Type: Eggcrate
Depth: 2' Vertical Fins / 2' Horizontal Overhangs
Spacing: 1' 6" V / 1' 6" H

NW Facing Façade:
Type: Vertical Fins
Depth: 2'
Spacing: 2' 7"

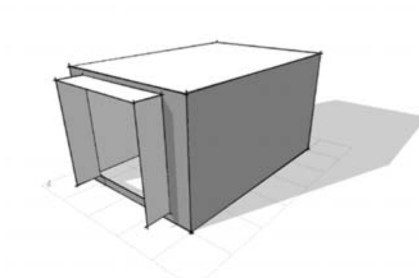
3738 - Stanford Law School
Incident Solar Radiation on Facades by Orientation

	w/o Shading *Btu/hr.ft2*	w/ Shading *Btu/hr.ft2*	Percent Peak Load Reduction	Time & Day Peak Occurs
North East	212	127	40%	7:00 am June
Sourth East	130	27	80%	10:00 am June
South West	182	12	90%	4:00 pm June
North West	120	36	70%	5:00 pm June

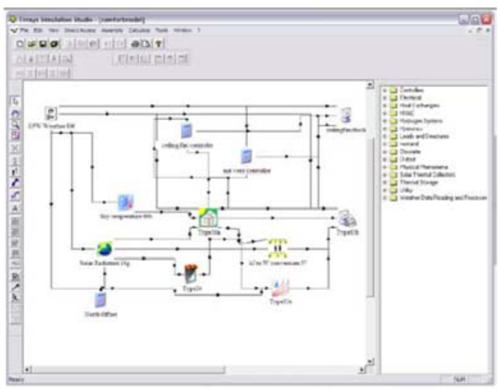




Natural Ventilation Study – Stanford University

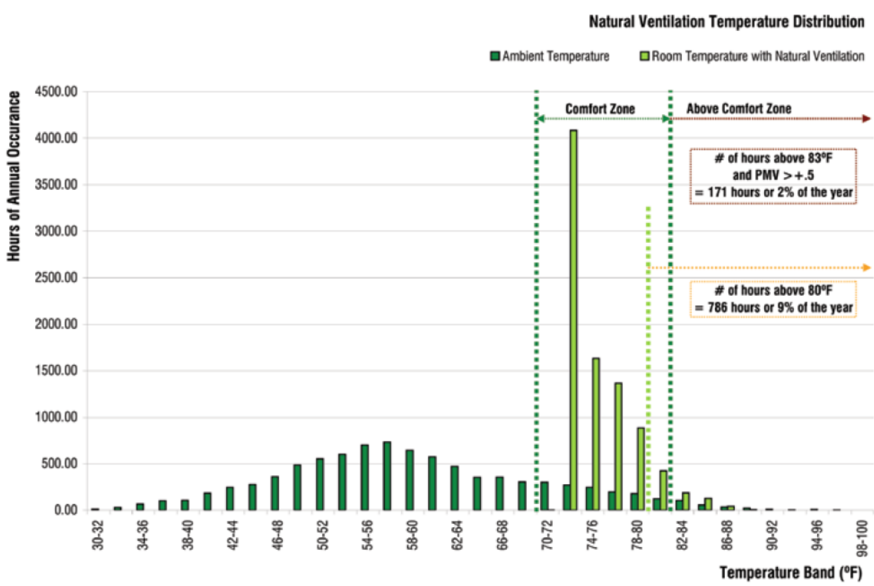


Typical faculty office with external shading



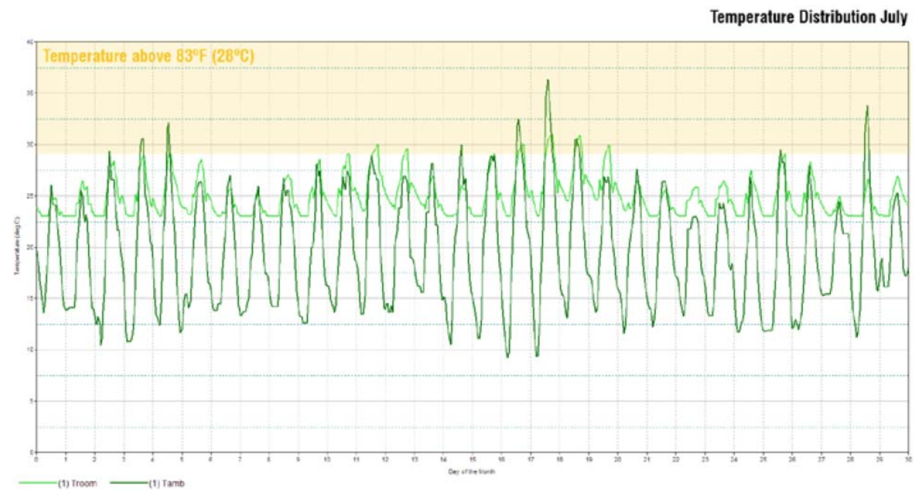
TRNSYS Model

Natural Ventilation Study – Stanford University



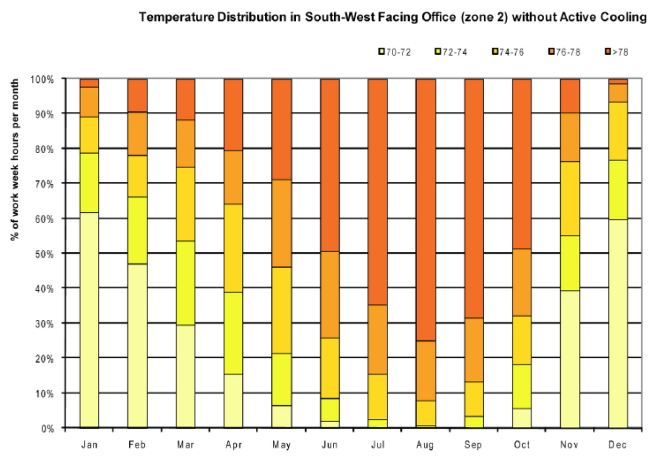
Temperature distribution for naturally ventilated office

Natural Ventilation Study – Stanford University



Simulated Temperature Distribution in July

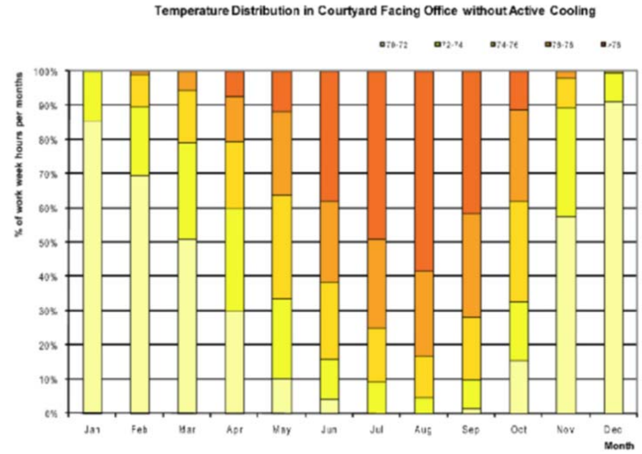
Natural Ventilation Study – Stanford University



Number of hours within temperature bands by month during work week

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total	%
>78	5	19	26	44	64	106	143	166	147	108	21	3	852	33%
76-78	19	25	30	33	55	53	44	38	39	42	30	10	418	16%
74-76	23	24	47	54	55	37	29	16	21	31	45	32	414	16%
72-74	38	38	53	50	33	14	5	1	7	28	34	33	334	13%
70-72	136	94	65	33	14	4	0	0	0	12	84	115	557	22%

Natural Ventilation Study – Stanford University



Number of hours within temperature bands by month during work week

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Total	%
>78	0	0	0	16	26	81	109	129	89	25	0	0	475	18%
76-78	0	2	12	28	54	51	57	55	65	59	4	0	387	15%
74-76	0	19	34	42	67	48	35	27	39	65	19	1	396	15%
72-74	32	40	63	64	52	25	20	10	18	38	68	16	446	17%
70-72	189	139	112	64	22	9	0	0	3	34	123	176	871	34%

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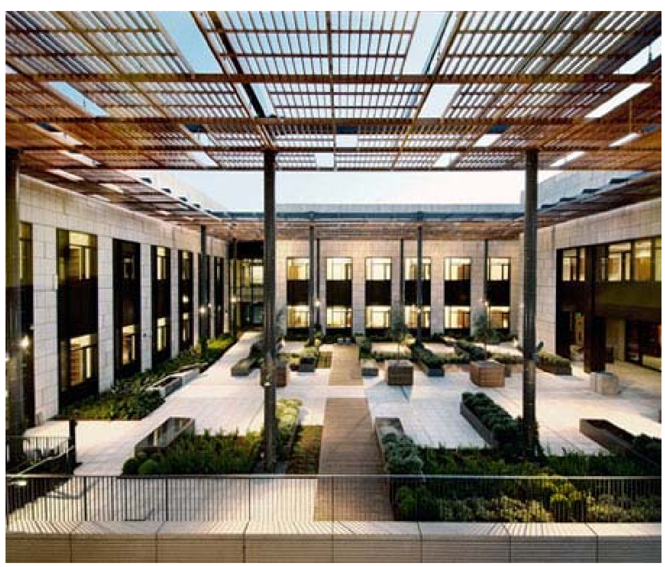

The End

Assignment 2: Comfort Criteria

Next Week: Building Fabric Losses

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