WEST POMERANIAN UNIVERSITY OF TECHNOLOGY, SZCZECIN , POLAND

WIMiM

THE FACULTY OF MECHANICAL ENGINEERING AND MECHATRONICS

Anna Majchrzycka

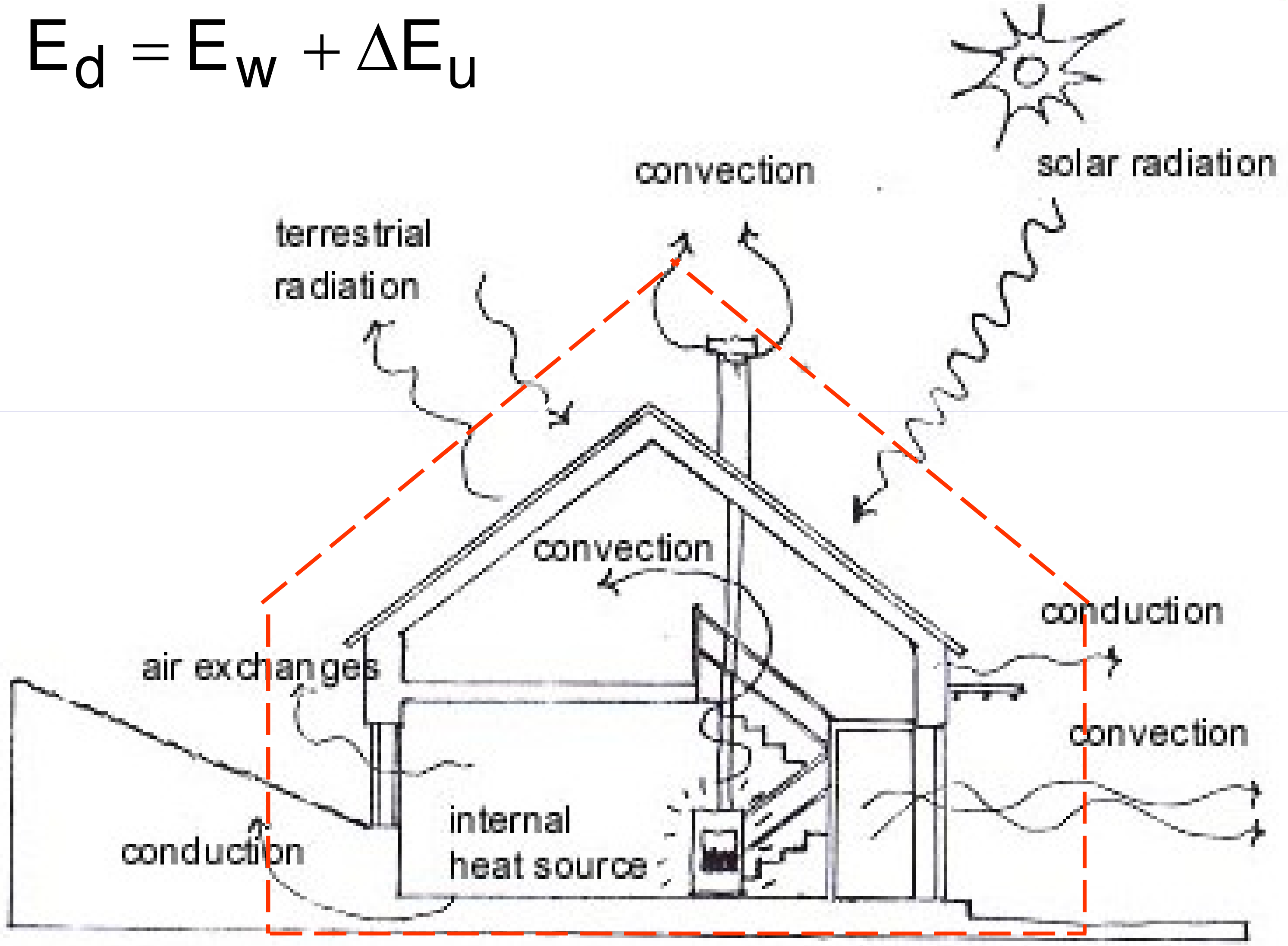
THERMAL COMFORT

ASHRAE STANDARD 55-66

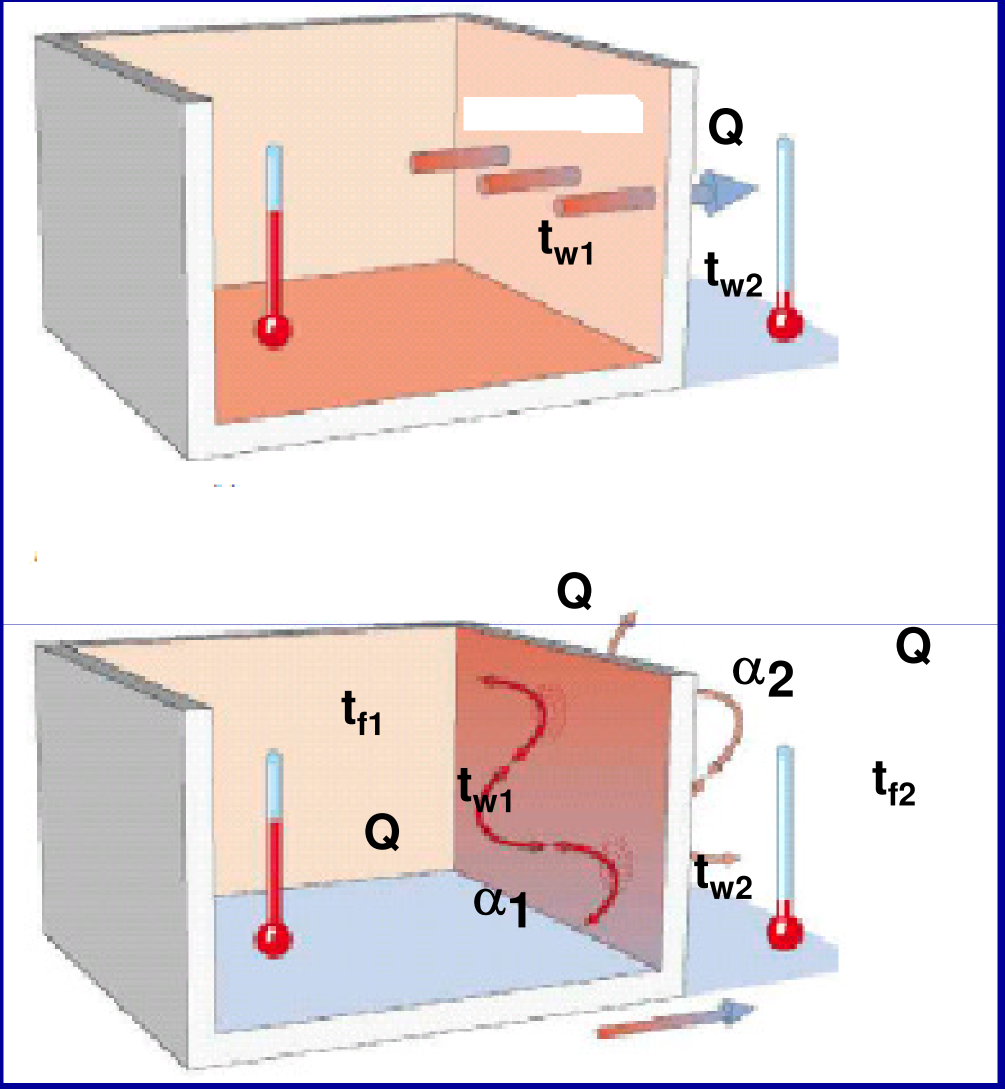
Thermal comfort is defined as that condition of mind that express satisfaction with the thermal environment.

FIRST LAW OF THERMODYNAMICS - ENERGY BALANCE

$$E_d = E_w + \Delta E_u$$



Based on :http://www.geocities.com/inescabral/housebudget.jpg



CONDUCTION

$$Q = \frac{\lambda}{\delta}(t_{w1} - t_{w2})A$$

CONVECTION

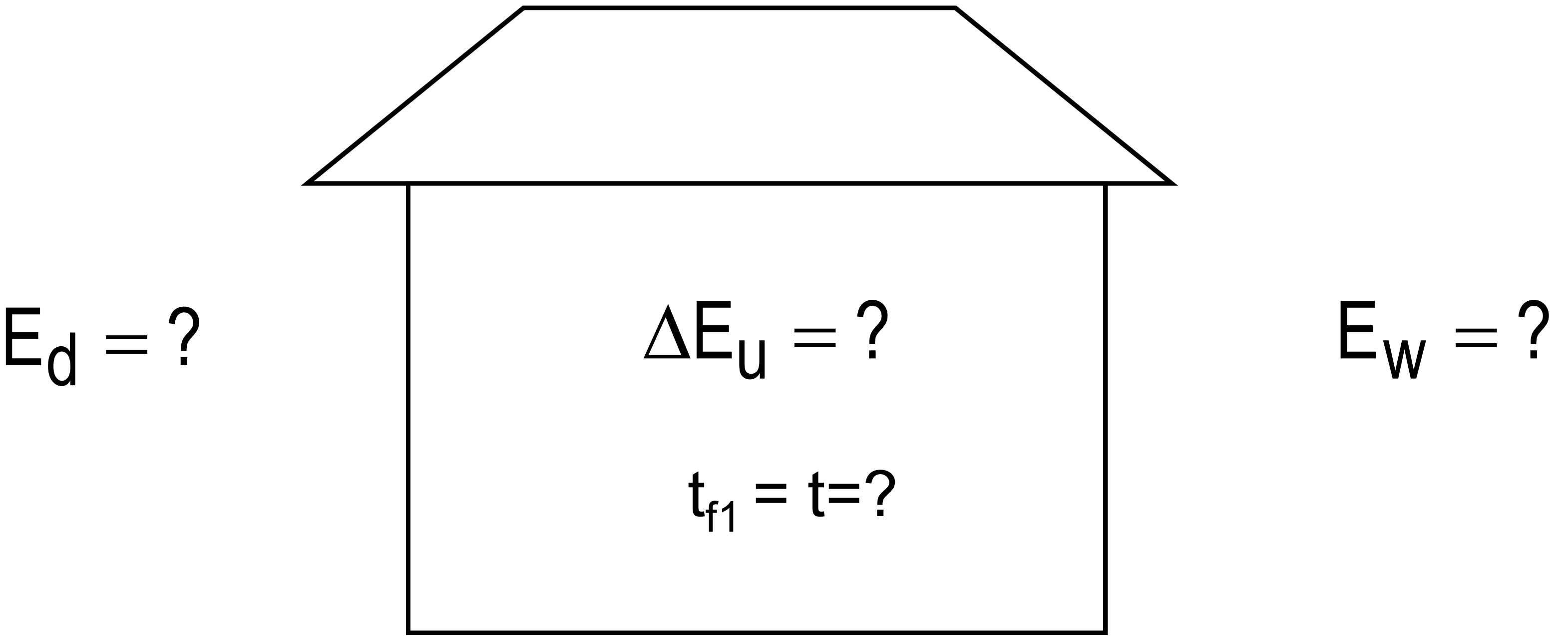
$$Q = \alpha_1(t_{f1} - t_{w1})A$$

$$Q = \alpha_2(t_{w2} - t_{f2})A$$

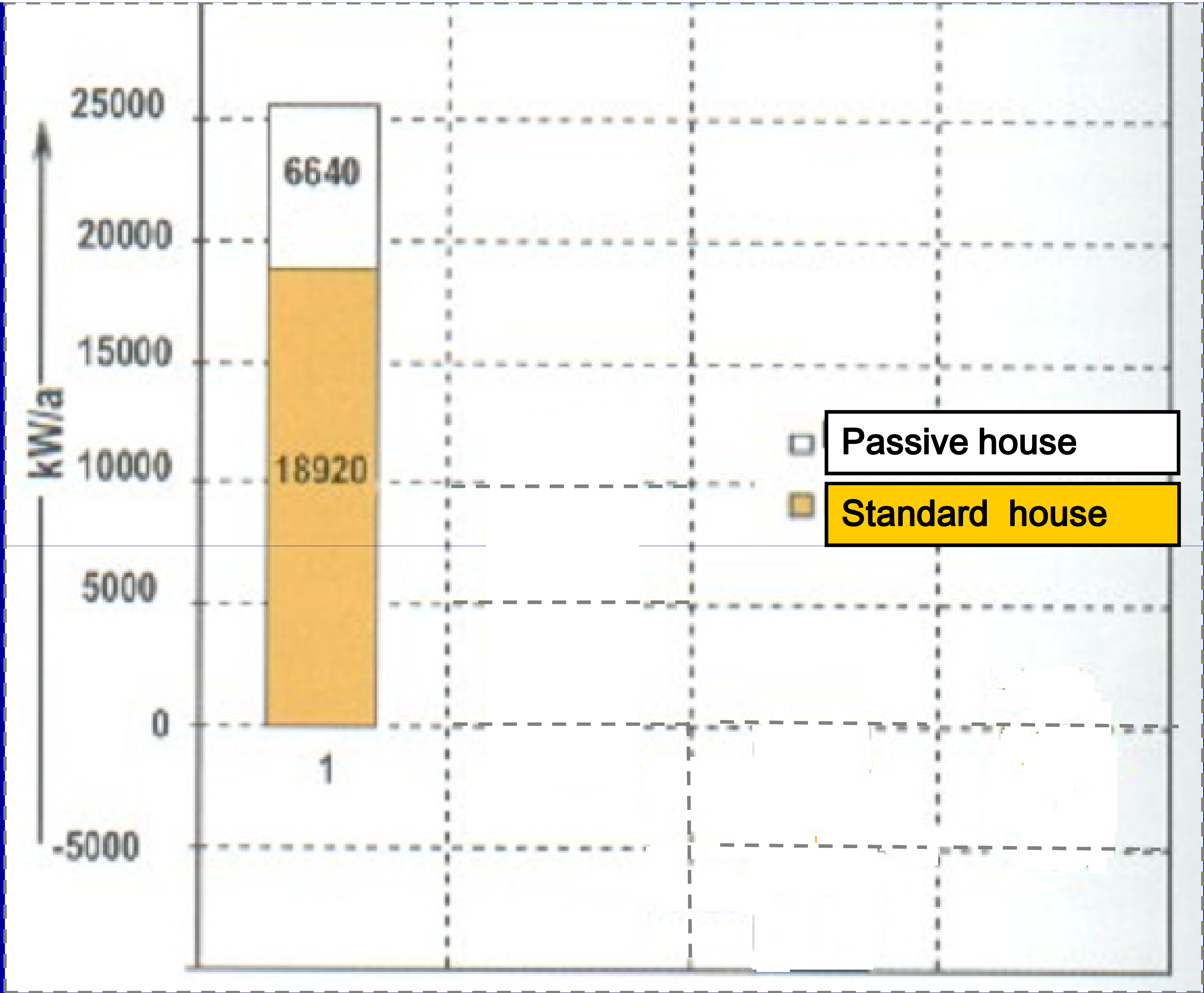
HEAT TRANSMISSION

$$Q = k(t_{f1} - t_{f2})A$$

t_{f1}=t

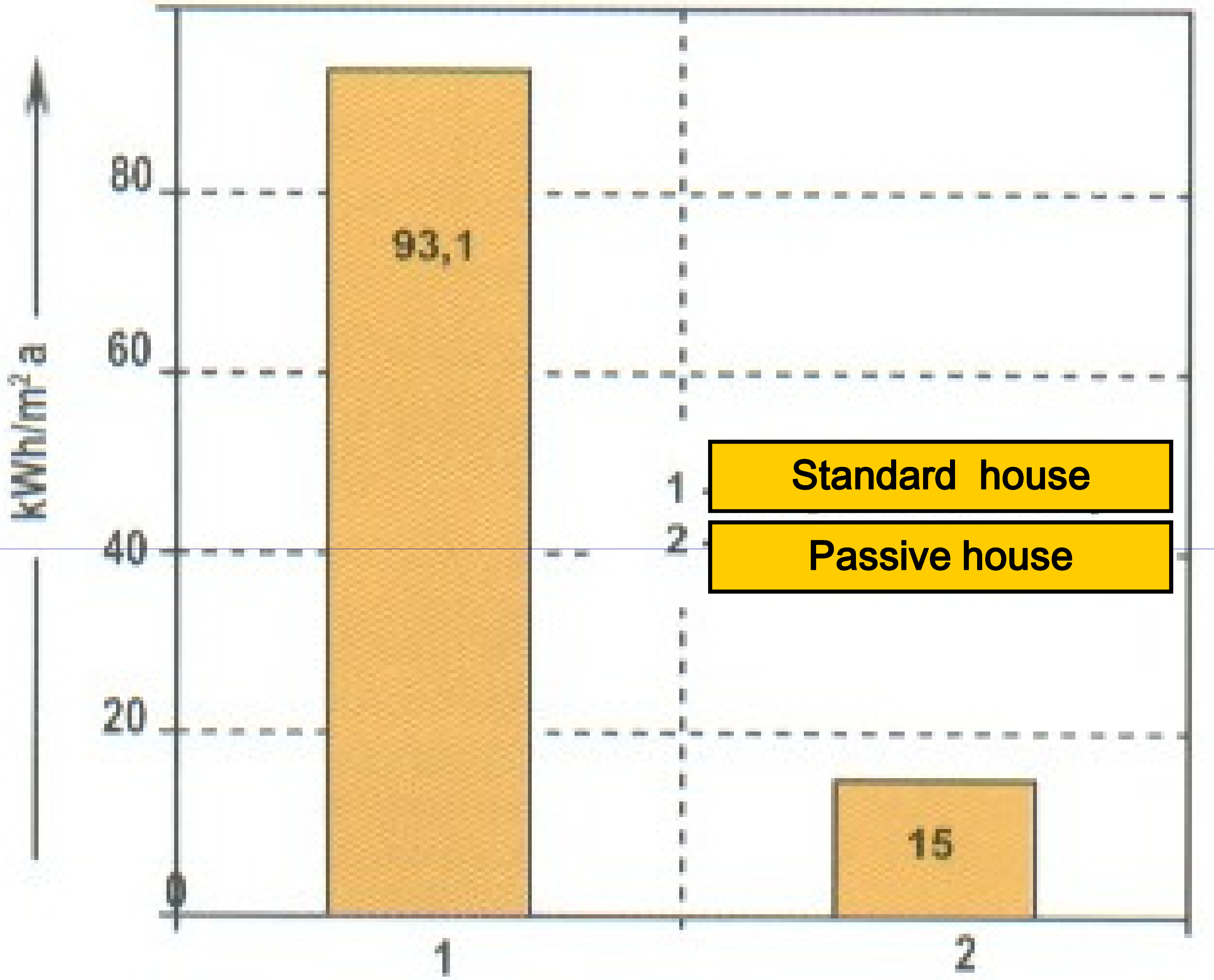


$E_d = ?$
$\Delta E_u = ?$
$t_{f1} = t = ?$
$E_w = ?$



YEARLY HEAT LOSSES IN PASSIVE AND STANDARD HOUSE

SOURCE: M.B.NANTKA:WENTYLACJA I ENERGIA W BUDOWNICTWIE TRADYCYJNYM I PASYWNYM, CZ.2 ENERGIA I BUDYNEK,02 (12),2008



HEATING REQUIREMENT FOR STANDARD AND PASSIVE HOUSE

SOURCE: M.B.NANTKA:WENTYLACJA I ENERGIA W BUDOWNICTWIE TRADYCYJNYM I PASYWNYM, CZ.2 ENERGIA I BUDYNEK,02 (12),2008

OIL CONSUMPTION FOR HOUSE HEATING IN EUROPE

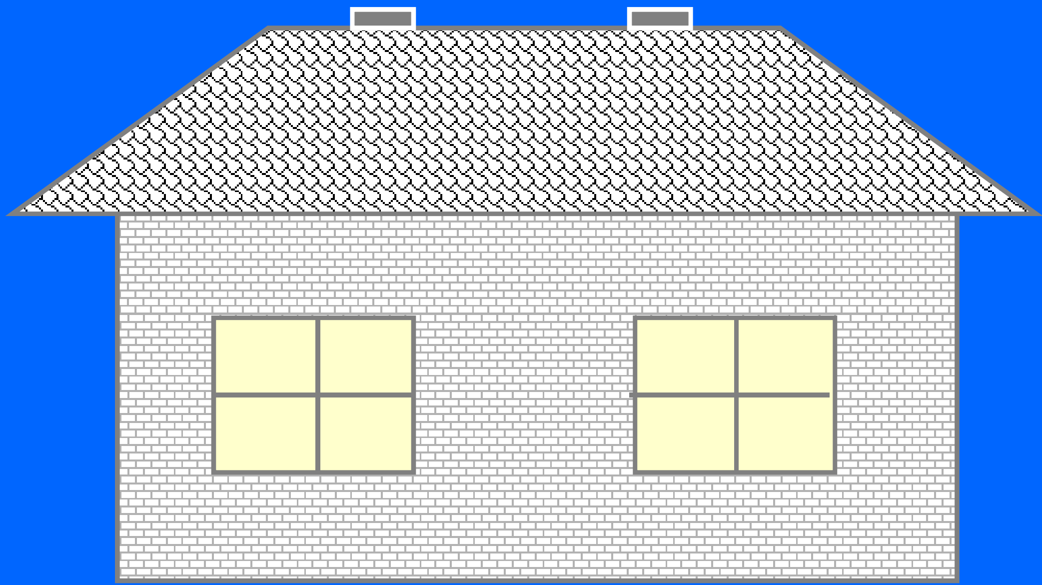
- CURRENT - 18 l / m^2 year
- NEAREST FUTURE - **5 l / m^2 year**

HOLISTIC CONCEPT OF HOMESPACE

EQUILIBRIUM BETWEEN QUALITY OF HOUSE, ESTHETICS AND ENERGY THAT COMES FROM LIGHT AND HEATING

HEATING

COOLING



BUILDING DESIGN & LAYOUT

AIR MOVEMENT

AIR CONDITIONING

ULTRA-LOW ENERGY BUILDINGS

- Space heating requirement

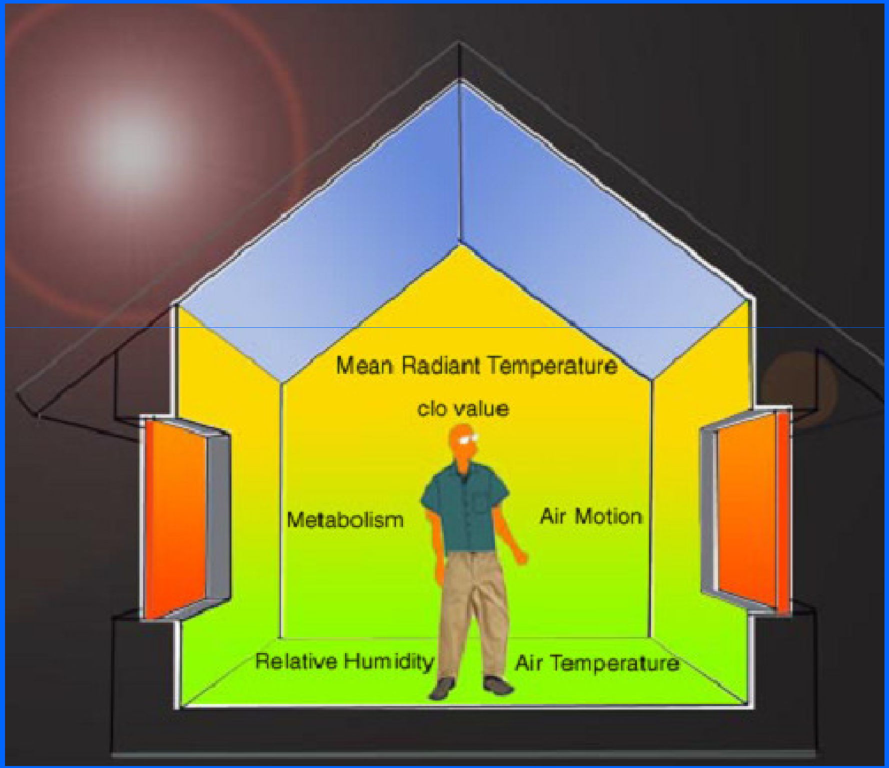
- Construction costs

- Design and construction

Design and construction

- Passive solar design
- Energy storage,
- Superinsulation,
- Advanced window technology,
- Airtightness,
- Ventilation,
- Space heating,
- Lighting and electrical appliances.

THE MOST IMPORTANT FACTORS INFLUENCING THERMAL COMFORT



ACTIVITY LEVEL

THERMAL RESISTANCE OF CLOTHING

AIR VELOCITY

AIR RELATIVE HUMIDITY

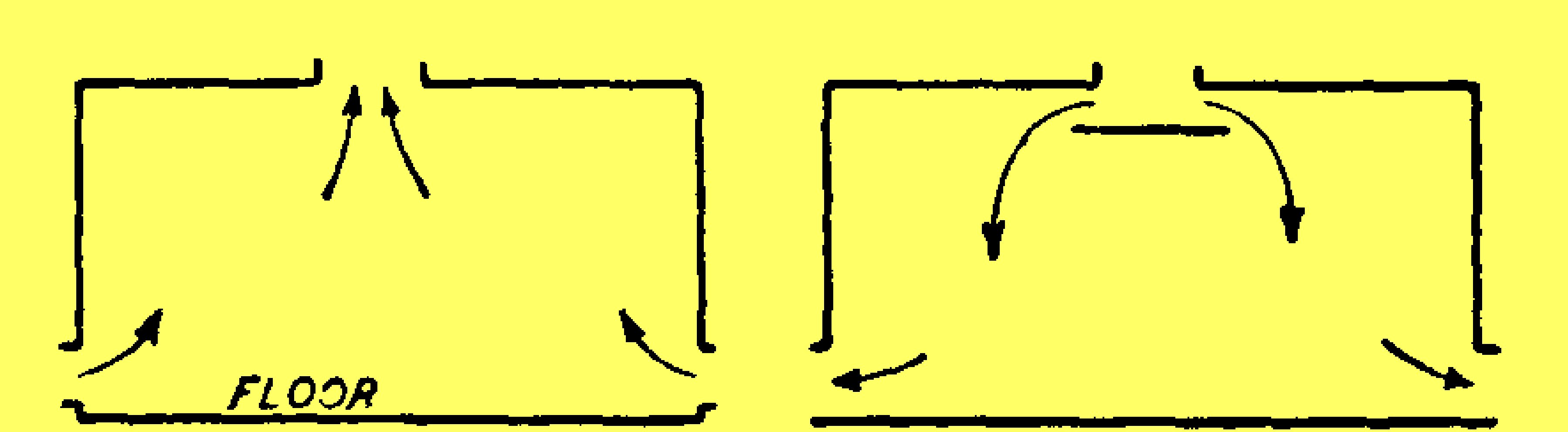
MEAN RADIANT TEMPERATURE

AIR TEMPERATURE

THE OTHER FACTORS THAT CAN INFLUENCE THERMAL COMFORT

- VERTICAL TEMPERATURE DIFFERENCE,
- ASYMETRIC FIELD OF TEMPERATURE,
- DRAFT,
- HEAT OR COLD FLOOR,
- COLOURS,
- AGE,
- SEX,
- ETHNIC DIFFERENCES,ETC.

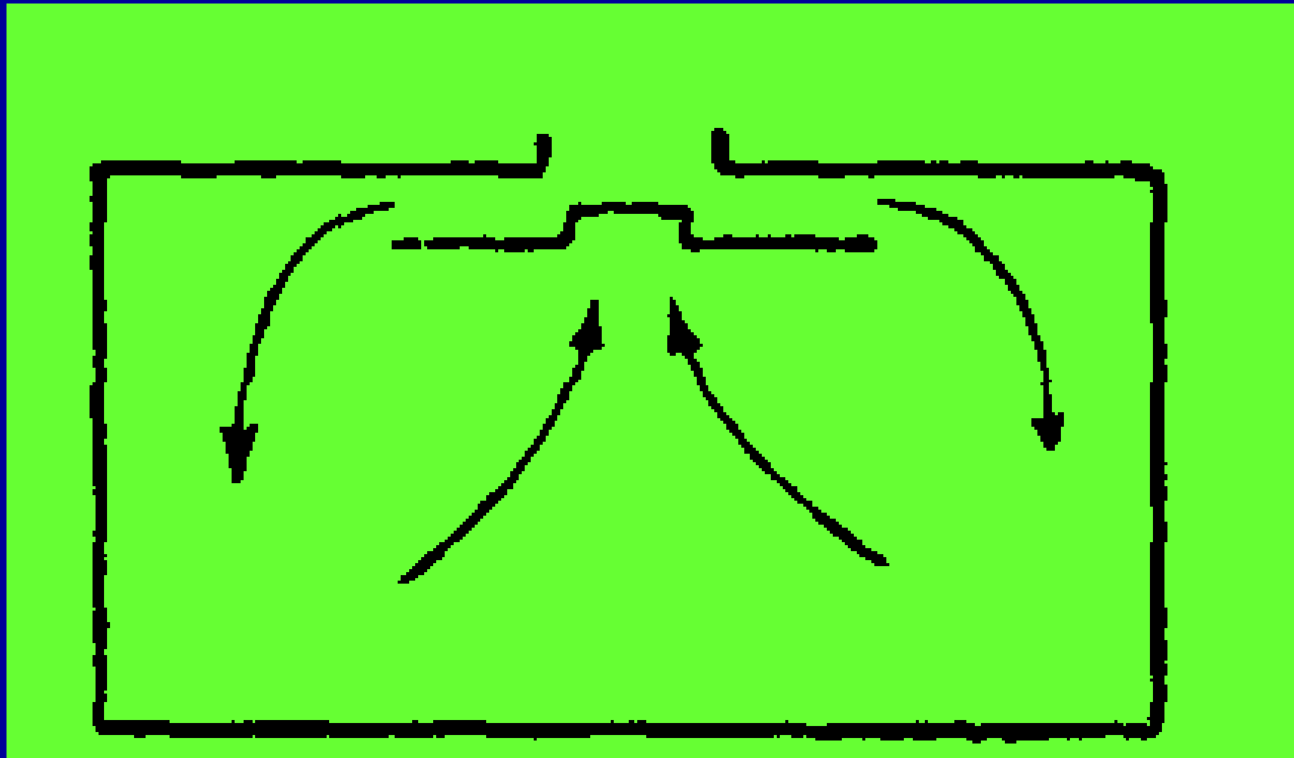
SYSTEMS OF AIR DISTRIBUTION IN BUILDINGS



UPWARD FLOW SYSTEM **DOWNWARD FLOW SYSTEM**

SOURCE: F.PORGES-HVAC ENGINEERS HANDBOOK, IX ED,1991

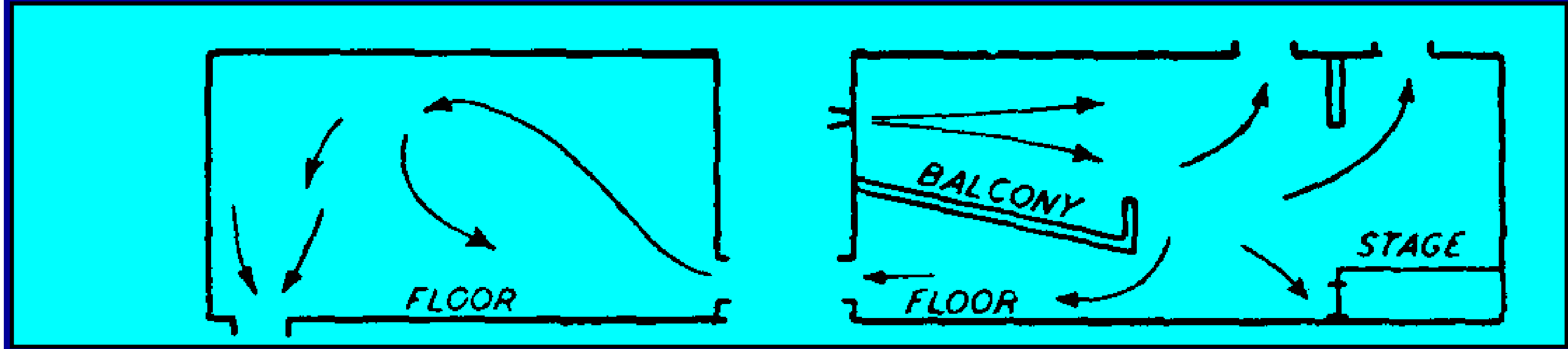
SYSTEMS OF AIR DISTRIBUTION IN BUILDINGS



HIGH LEVEL SUPPLY AND RETURN SYSTEM

SOURCE: F.PORGES-HVAC ENGINEERS HANDBOOK, IX ED,1991

SYSTEMS OF AIR DISTRIBUTION IN BUILDINGS

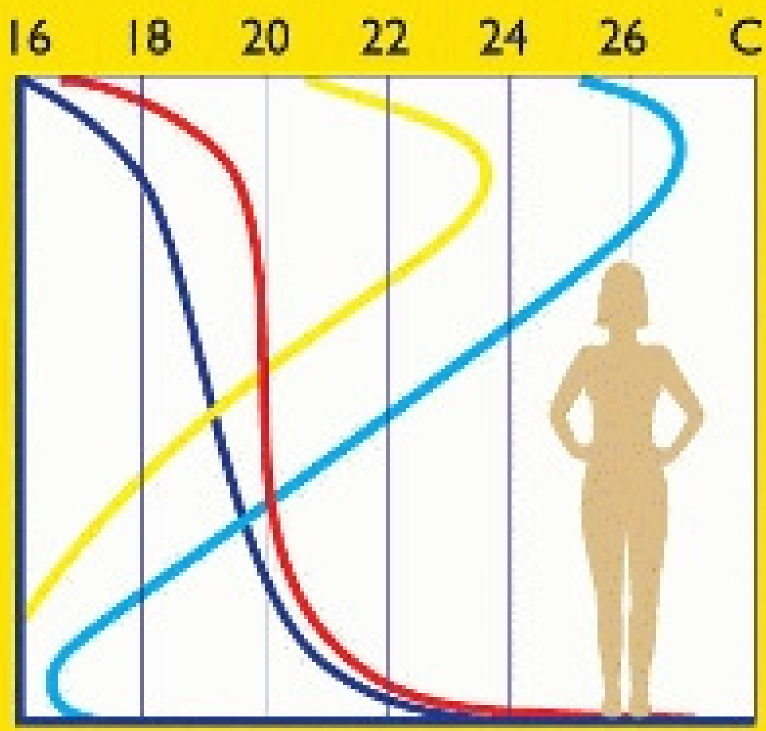


LOW LEVEL SUPPLY AND RETURN SYSTEM

EJECTOR SYSTEM

SOURCE: F.PORGES-HVAC ENGINEERS HANDBOOK, IX ED,1991

Vertical temperature distribution for the different types of heating

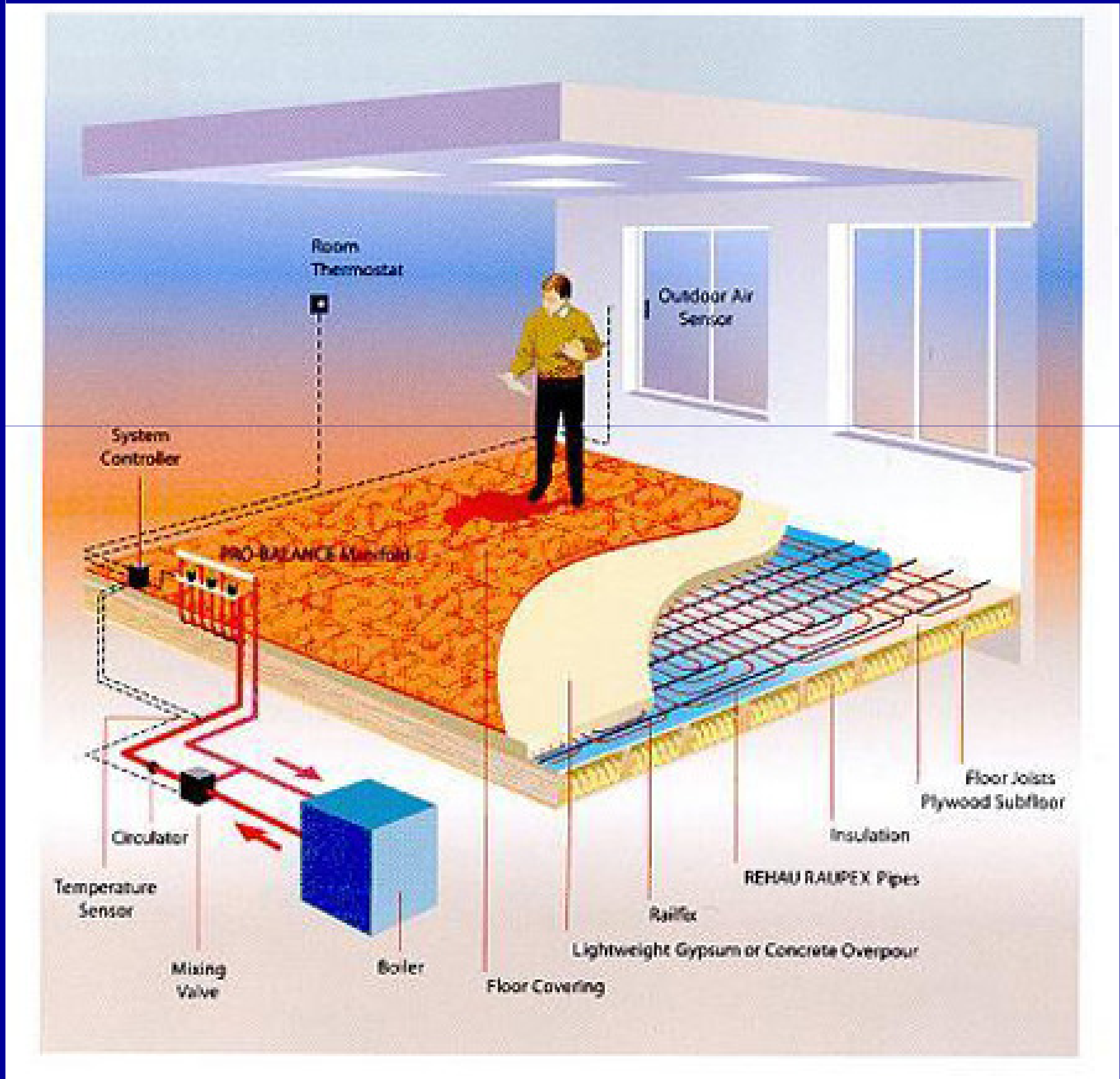


Floor electric heating

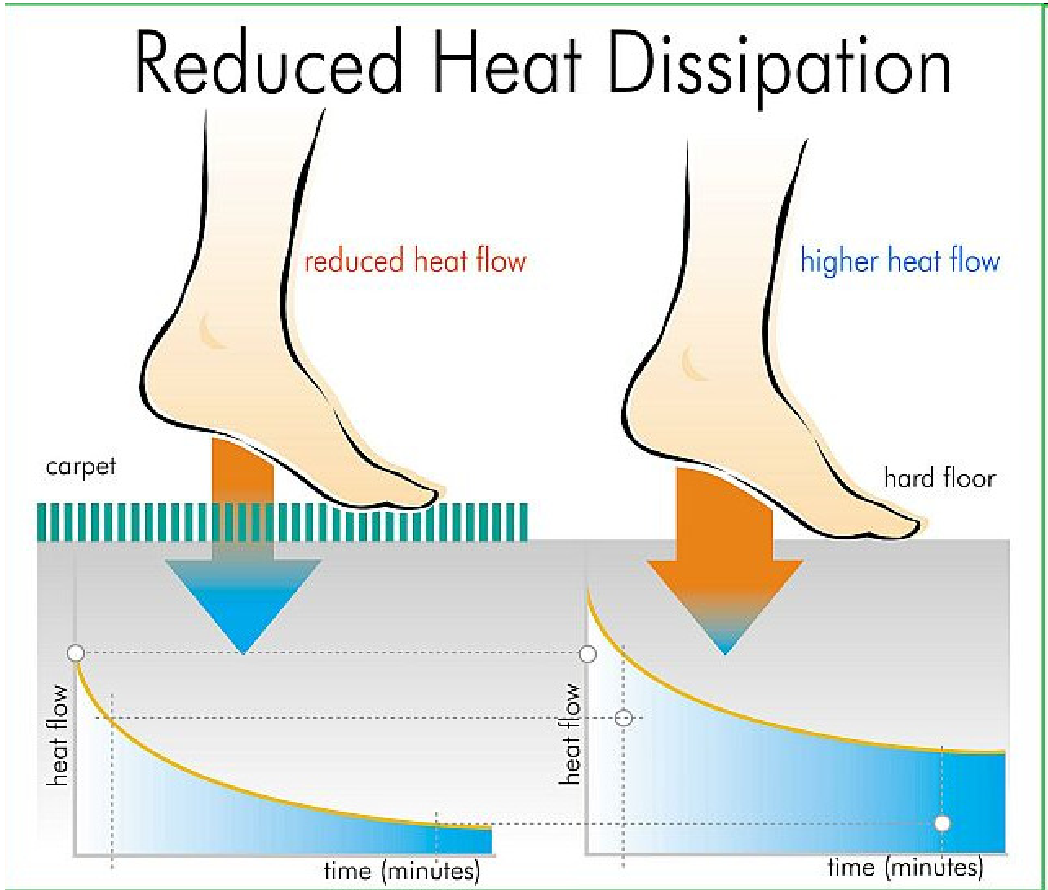
Ideal profil

Convective heaters placed on the internal walls

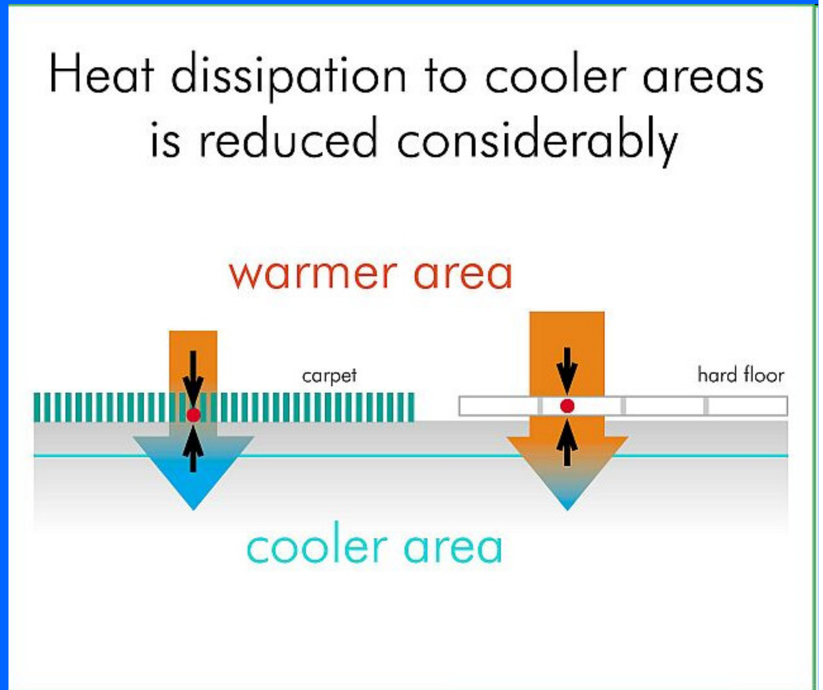
Air heating



https://reich-chemistry.wikispaces.com/file/view/radiant-floor-heating.jpg



Floor covering properties



The contact coefficient

$$b = \sqrt{\lambda \cdot \rho \cdot c}$$

http://www.gut-ev.de/en/Images/Reduced_Heat_Dissipation.jpg

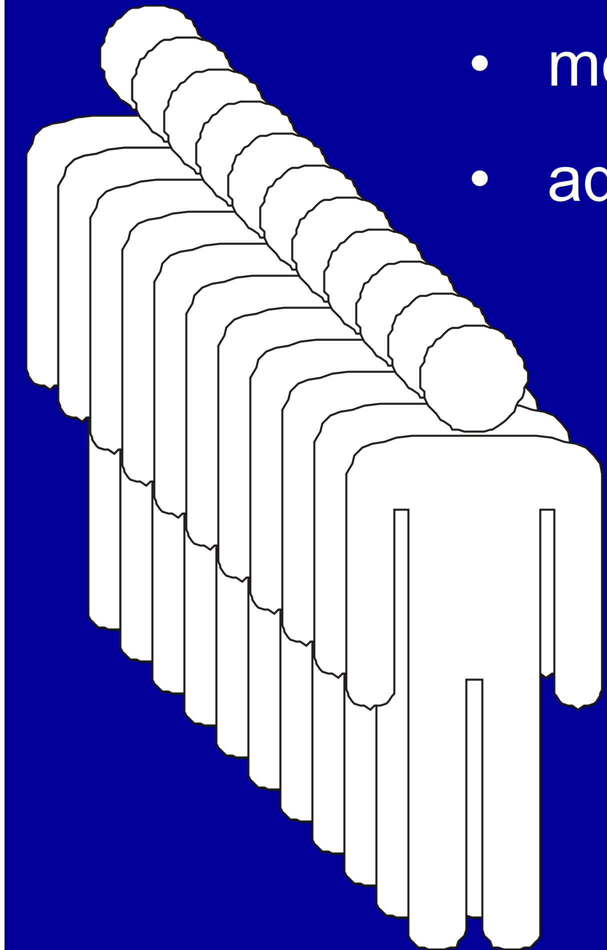
COLOUR

,,WARM " COLOURS

,,COOL " COLOURS

As colour has no thermal INFLUENCE ON MAN , any influence on thermal sensation must therefore be of a psychological nature.

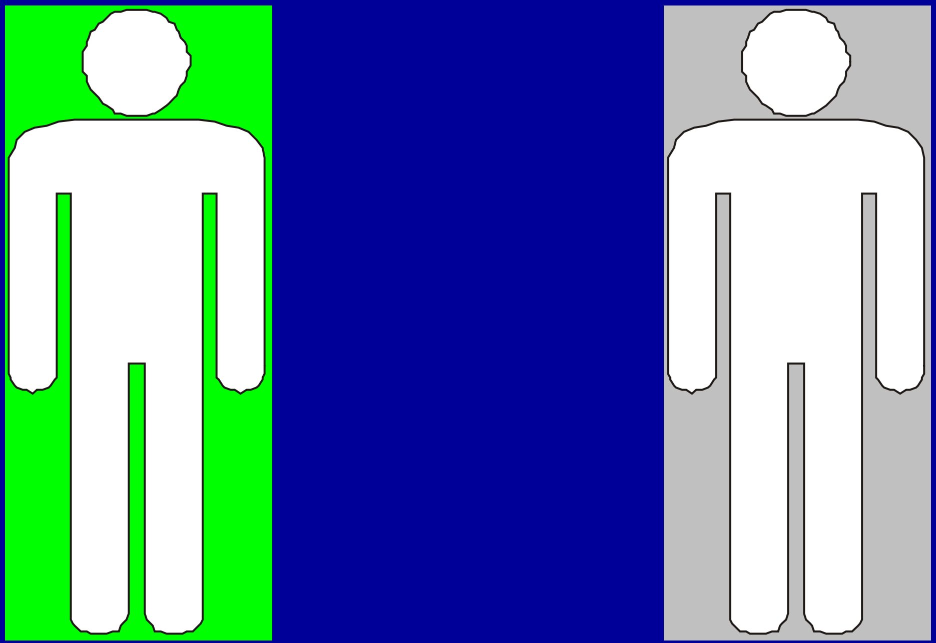
CROWDING



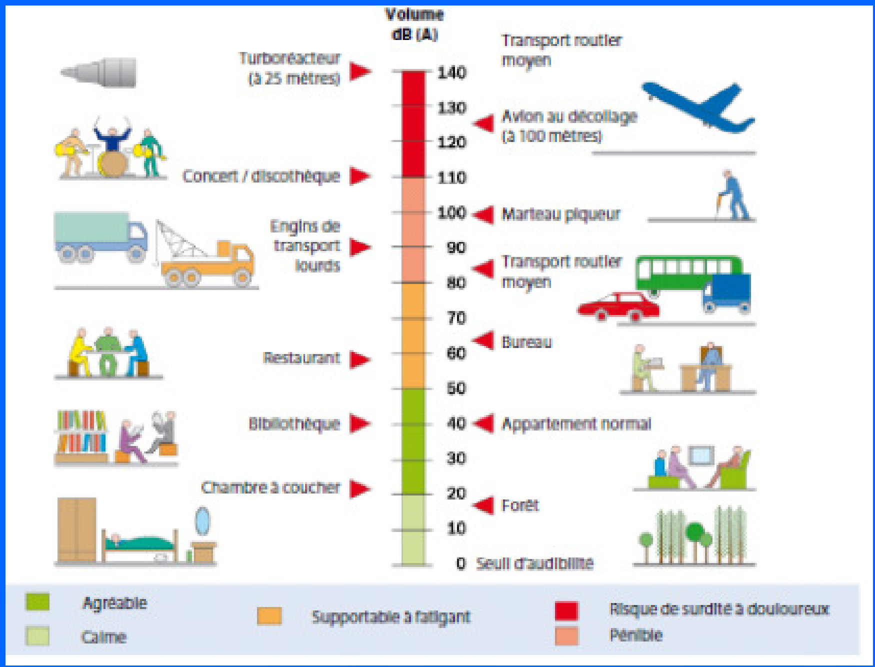
- the convective heat transfer will be impaired ,
- mean radiant temperature will increase,
- added heat sources of the occupants.

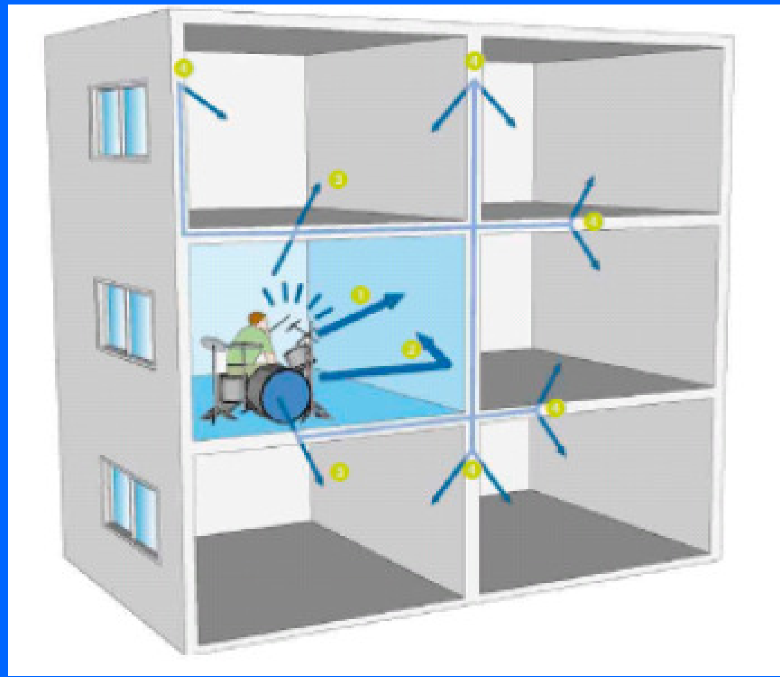
AGE

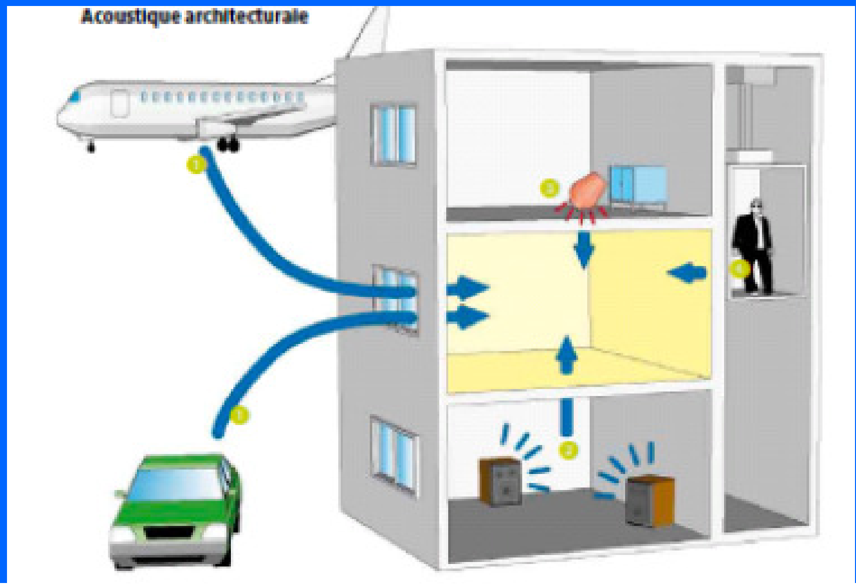
People over 40 years of agree prefere comfort temperature $\Delta t = 1K$ effective higher than that desired for persons below that age.



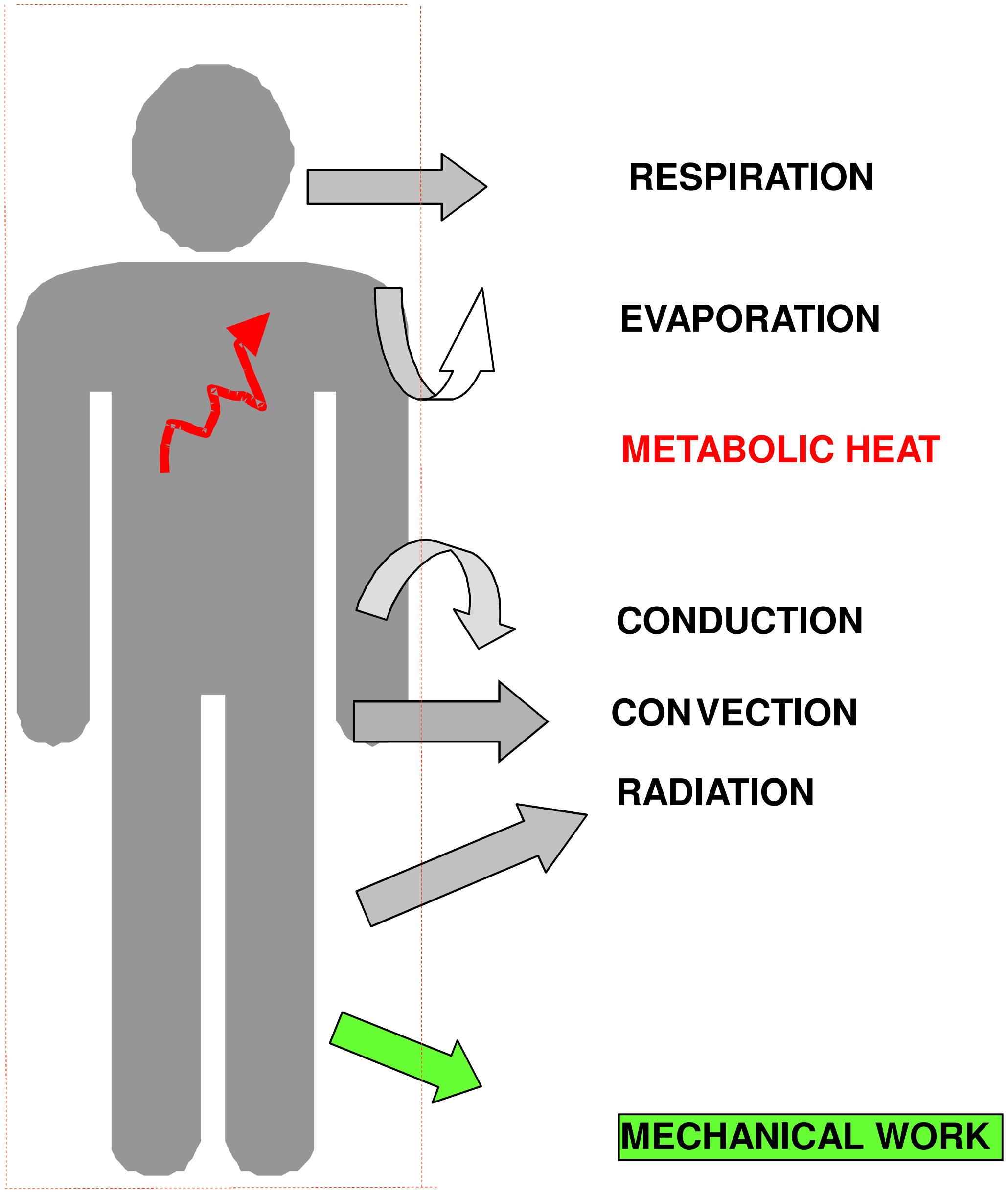
NOISE


Volume
dB (A)
Turboréacteur
(à 25 mètres)
Transport routier
moyen
140
130
120
110
100
90
80
70
60
50
40
30
20
10
0
Avion au décollage
(à 100 mètres)
Concert / discothèque
Marteau piqueur
Engins de
transport
lourds
Transport routier
moyen
Bureau
Restaurant
Bibliothèque
Appartement normal
Chambre à coucher
Forêt
Seuil d'audibilité
Agréable
Calme
Supportable à fatigant
Risque de surdité à douloureux
Pénible




Acoustique architecturale

$$E_d = E_w + \Delta E_u$$



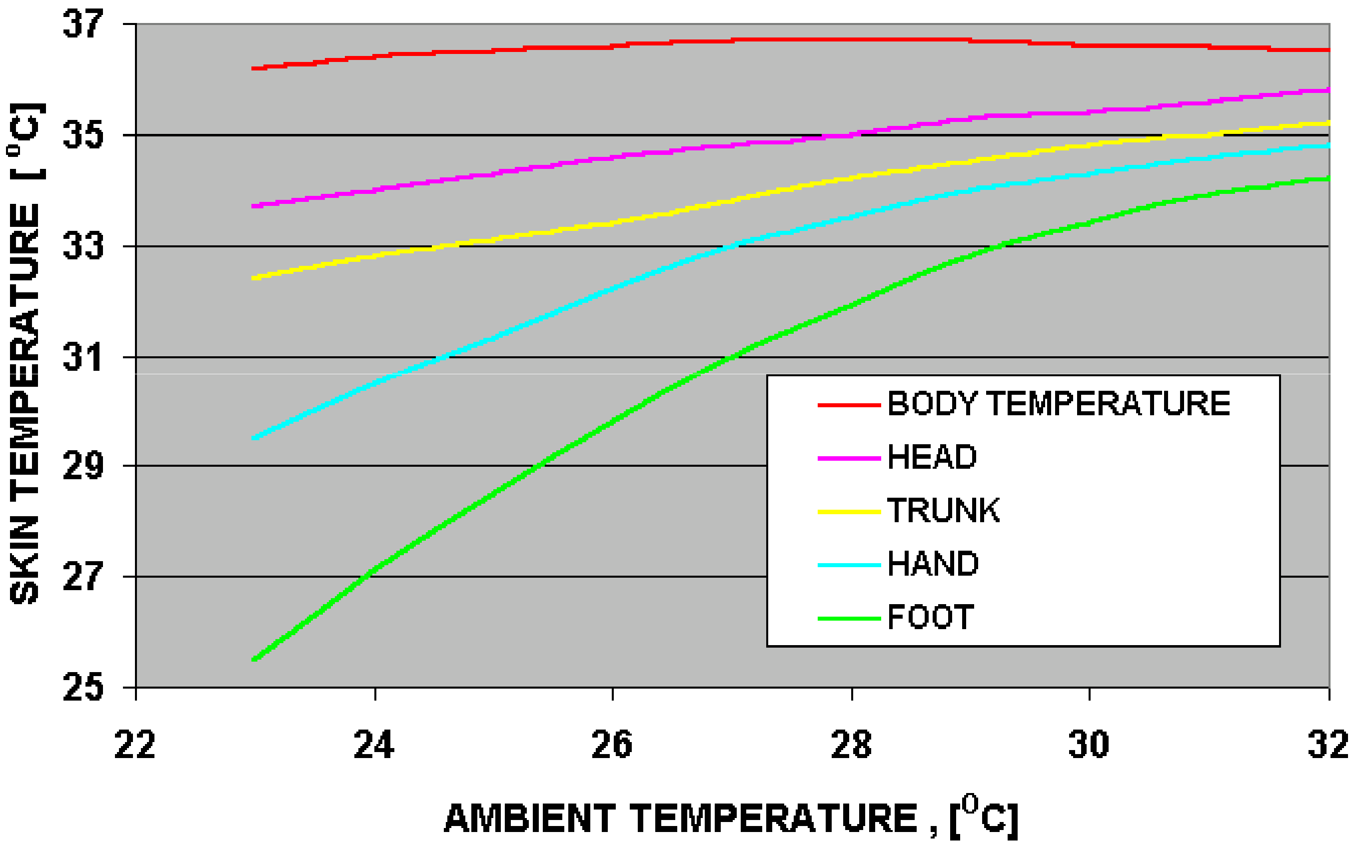
Du Bois area

$$A = 0.202\ m^{0.425}\ h^{0.725}$$

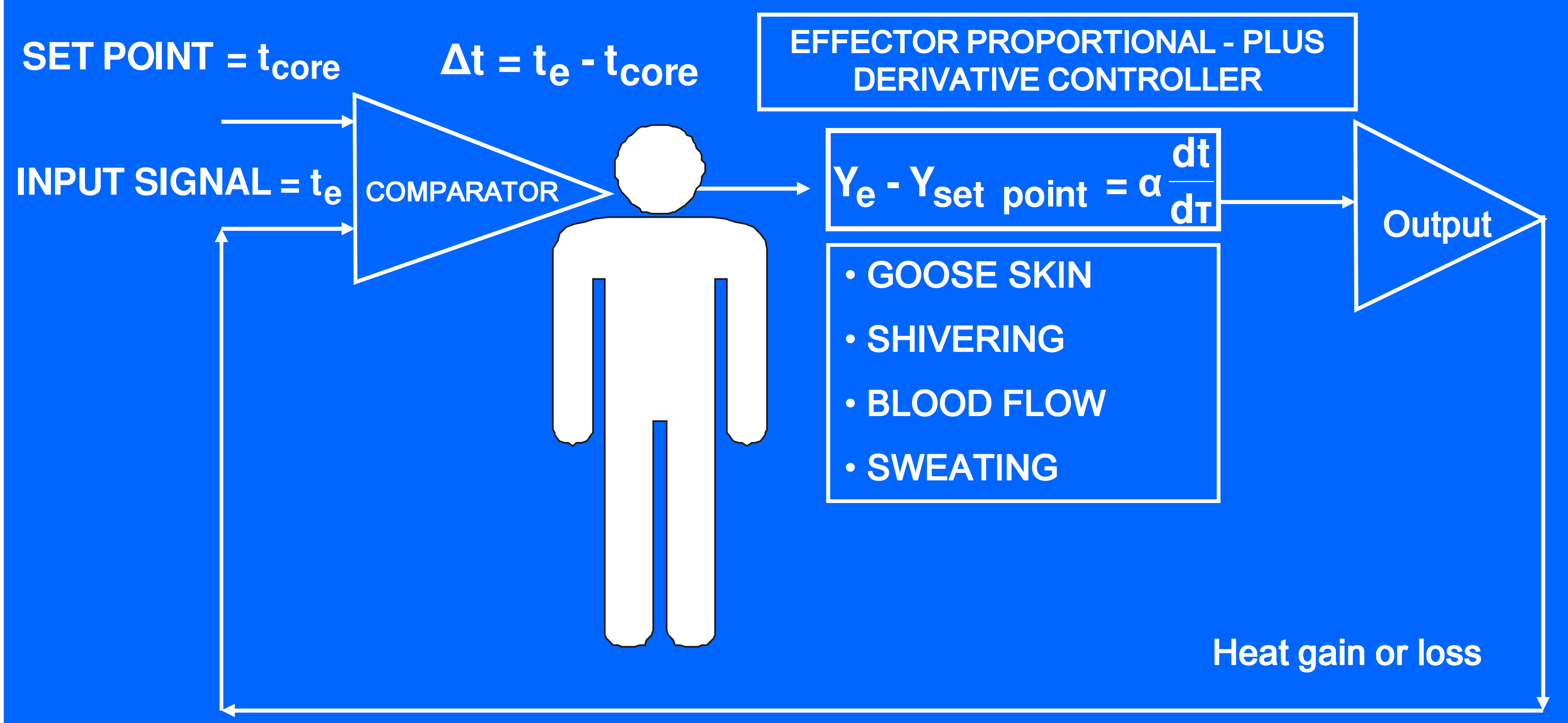
where:

m- body mass, kg

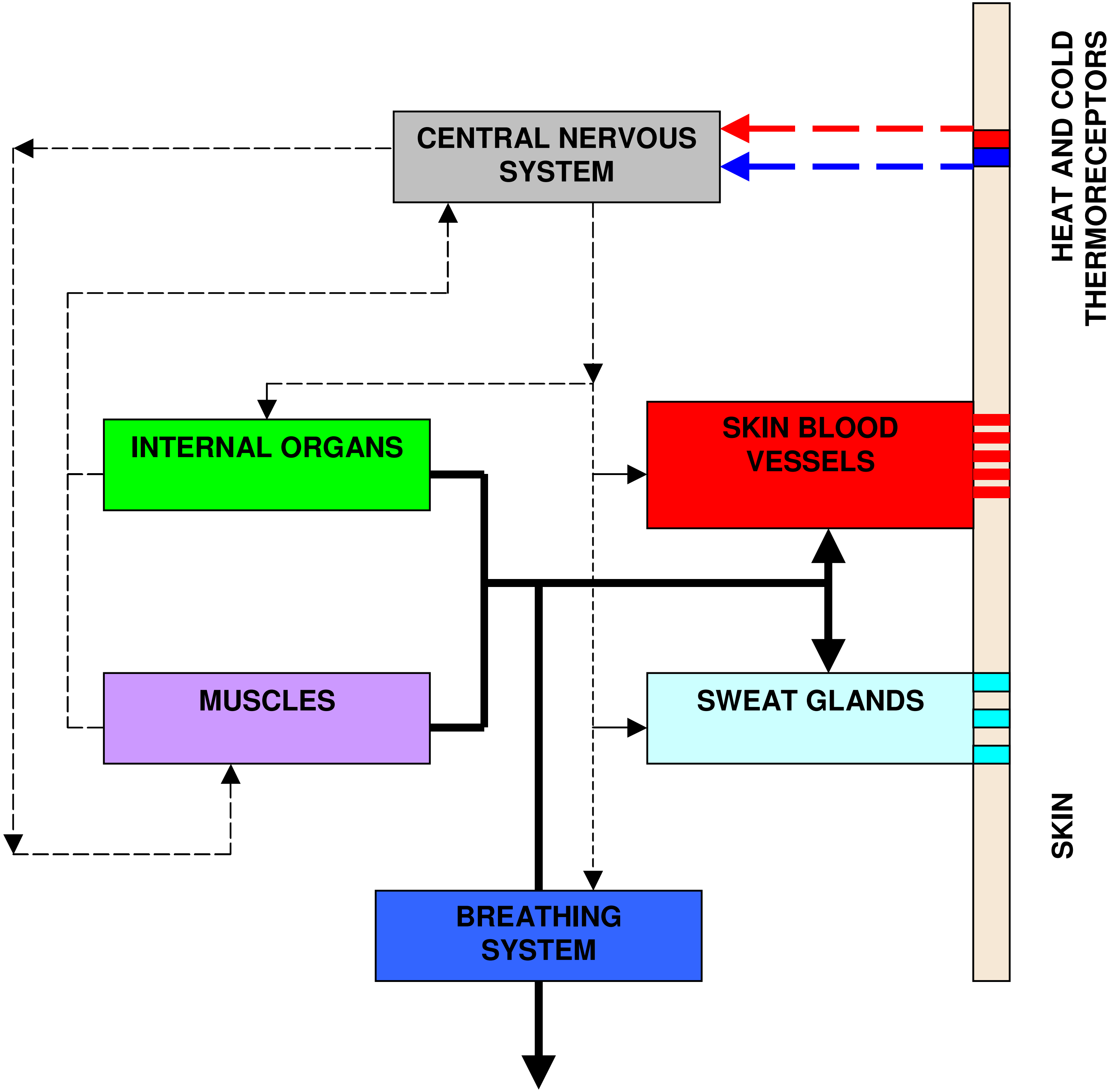
h-height, m

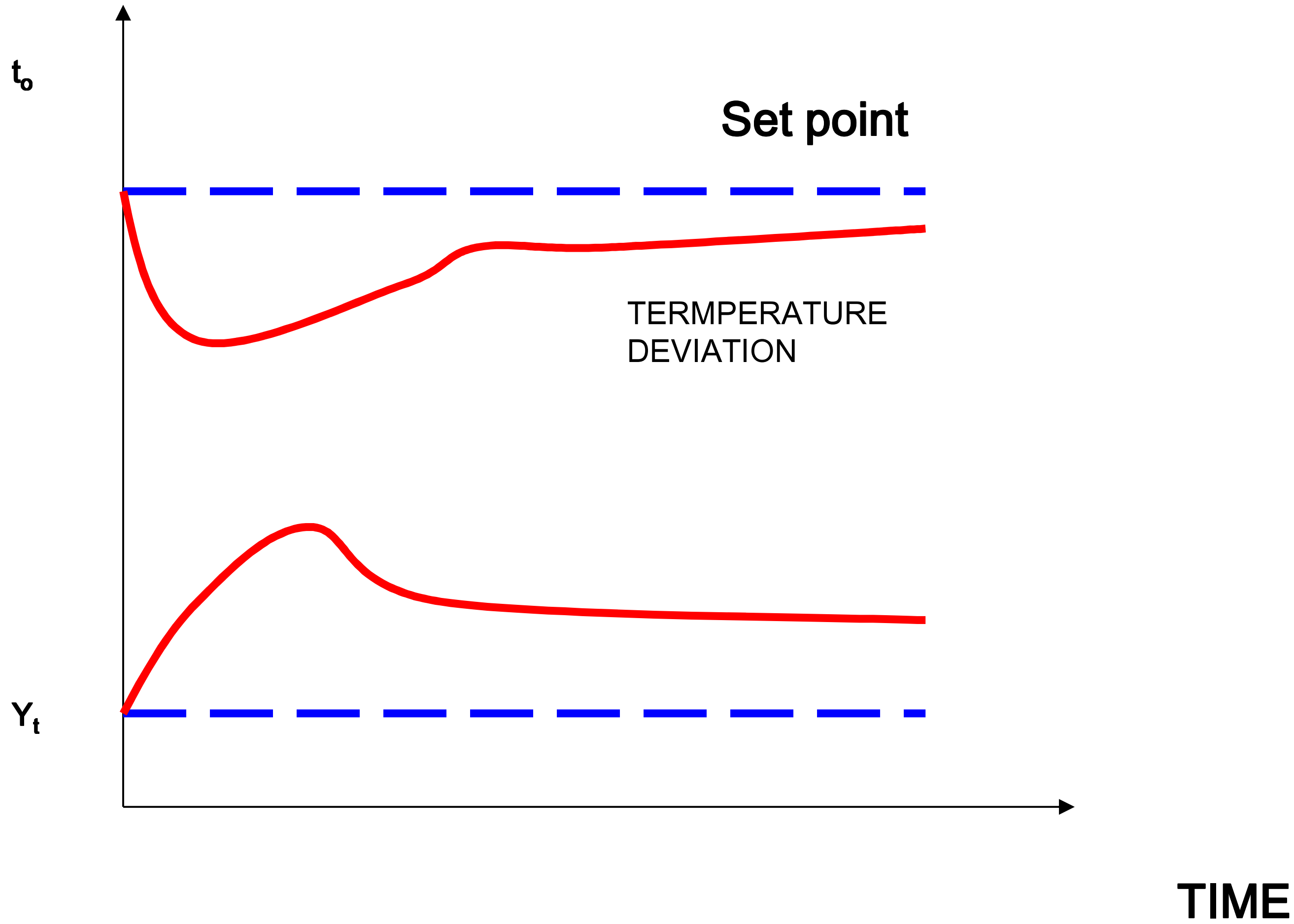


Relationship between skin temperature and ambient temperature

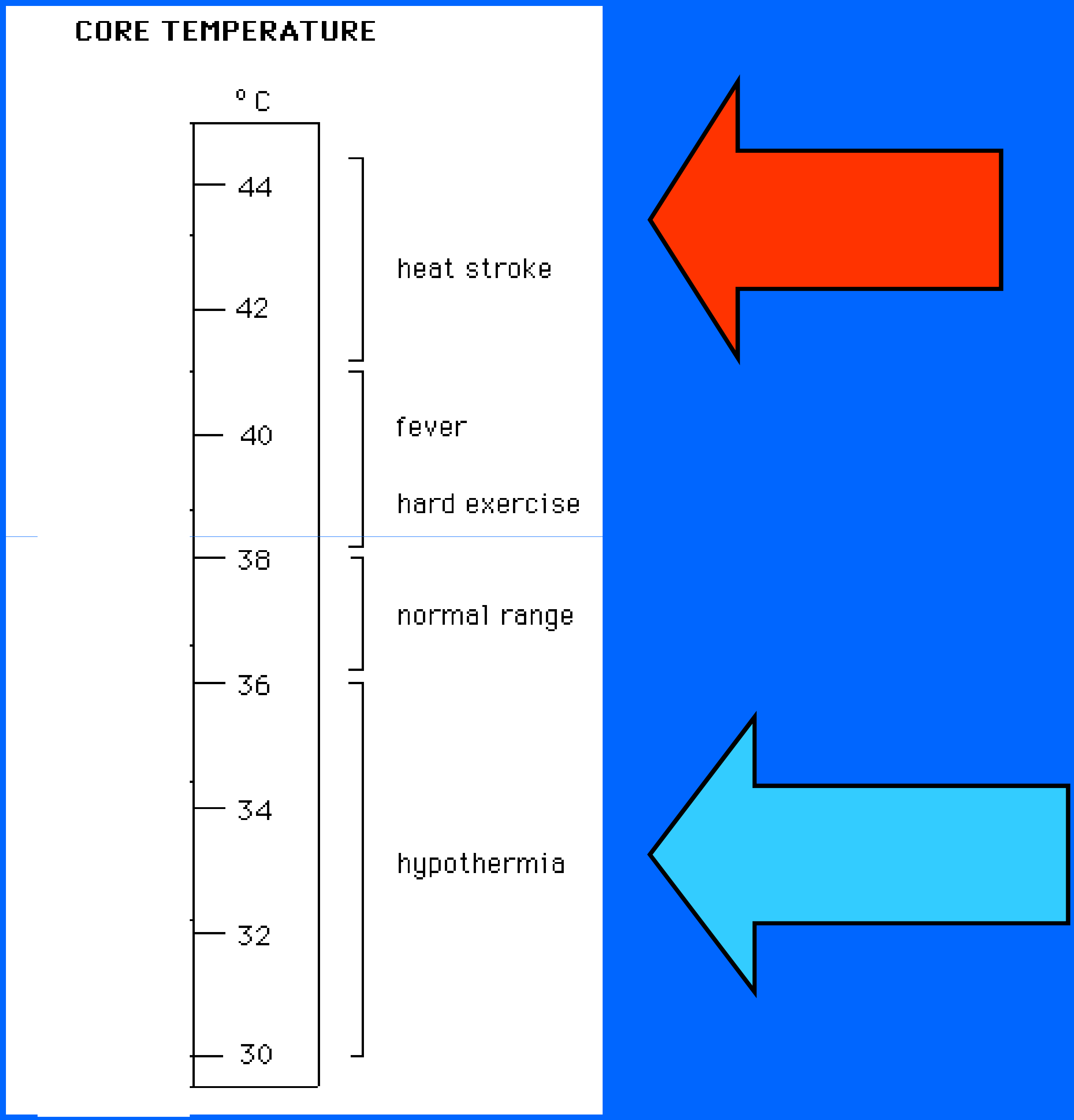


THERMORGULATORY SYSTEM OF HUMAN BODY


CENTRAL NERVOUS SYSTEM
HEAT AND COLD THERMORECEPTORS
INTERNAL ORGANS
SKIN BLOOD VESSELS
MUSCLES
SWEAT GLANDS
SKIN
BREATHING SYSTEM

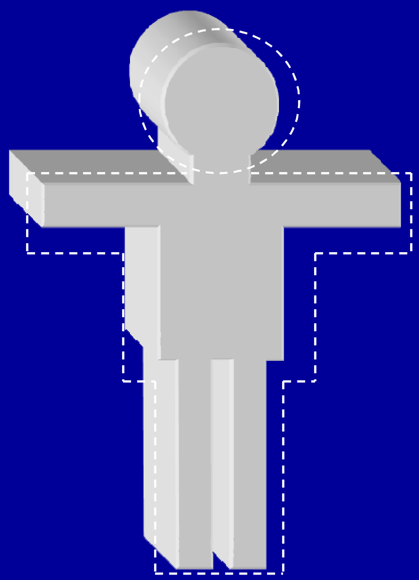


THERMOREGULATORY SYSTEM WITH PROPORTIONAL TEMPERATURE REGULATION



http://www.princeton.edu

THERMAL BALANCE OF THE HUMAN BODY



- METABOLIC HEAT - $\dot{Q}_m$
- HEAT LOSS BY SKIN DIFFUSION - $\dot{Q}_d$
- HEAT LOSS BY EVAPORATION OF SWEAT- $\dot{Q}_e$
- LATENT RESPIRATORY HEAT LOSS- $\dot{Q}_u$
- DRY RESPIRATORY HEAT LOSS - $\dot{Q}_j$
- HEAT CONDUCTION THROUGH CLOTHING- $\dot{Q}_p$
- HEAT LOSS BY RADIATION - $\dot{Q}_r$
- HEAT LOSS BY CONVECTION - $\dot{Q}_k$
- MECHANICAL POWER - $\dot{W}$

$$\dot{Q}_m - \dot{W} - (\dot{Q}_d + \dot{Q}_e + \dot{Q}_u + \dot{Q}_j + \dot{Q}_k + \dot{Q}_r) = \pm\dot{S} \qquad (1)$$

$$\dot{Q}_m = \dot{Q} + \dot{W} \qquad (2)$$

Source :Thermal Comfort, P. O. Fanger, McGraw-Hill, New York, 1970.

THERMAL HOMEOSTASIS , $t_{core} = 37^oC$

$$\dot{S} = 0 \qquad (3)$$

CONDITIONS FOR THERMAL COMFORT

$$\dot{Q} - (\dot{Q}_d + \dot{Q}_e + \dot{Q}_u + \dot{Q}_j) = \dot{Q}_P = \dot{Q}_k + \dot{Q}_r \qquad (4)$$

$$A < t_s < B \qquad (5)$$

$$C < \dot{Q}_e < D \qquad (6)$$

THE COMFORT TEMPERATURE

$$t = f(v, \dot{q}_m, \varphi, t_r, I_{cl}) \qquad [7]$$

- INTERNAL HEAT PRODUCTION

INTERNAL BODY HEAT

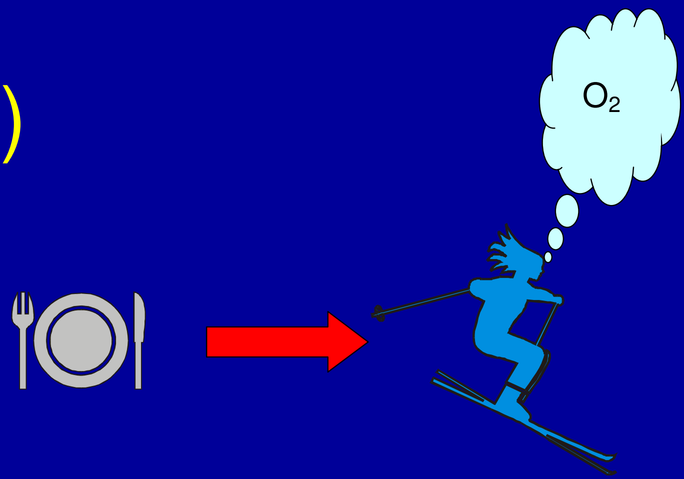
$$\dot{Q} = \dot{Q}_m - \dot{W} \qquad \textbf{(8)}$$

EXTERNAL MECHANICAL EFFICIENCY

$$\eta = \frac{\dot{W}}{\dot{Q}_m} \qquad \textbf{(9)}$$

METABOLIC HEAT

$$\dot{Q}_m = C\dot{V}_{O2} = \dot{q}_m \cdot A_{Du} \quad (10)$$



C= 18.85-20.95 J/cm^3

$$\dot{Q} = C \cdot (1 - \eta) \cdot \dot{V}_{O_2} = (1 - \eta) \cdot \dot{Q}_m = (1 - \eta) \cdot \dot{q}_m \cdot A_{Du} \quad (11)$$

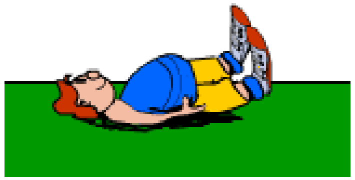


Table 1. Metabolic Heat Generation Rates

	Activity	$\dot{q}_m$ [W/m^2]
1	Lying down	47
2	Quietly seated	58
3	Sedentary activity (office, home, school)	58
4	Standing, relaxed	70
5	Light activity (shopping, laboratory, light work)	93
6	Medium activity (shop work, domestic work, machine work)	117
7	Heavy activity (heavy machine work, garage work	175
8	Heavy exercise (running)	525

Thermal Comfort, P. O. Fanger, McGraw-Hill, New York, 1970.

• HEAT LOSS BY PERSPIRATION (BY SKIN DIFFUSION)

$$\dot{Q}_d = \beta \cdot r \cdot A_{Du} \cdot (p_s - \varphi \cdot p_{sw}) \quad (12)$$

$$p_s = 256 \cdot t_s - 3360, (13)$$

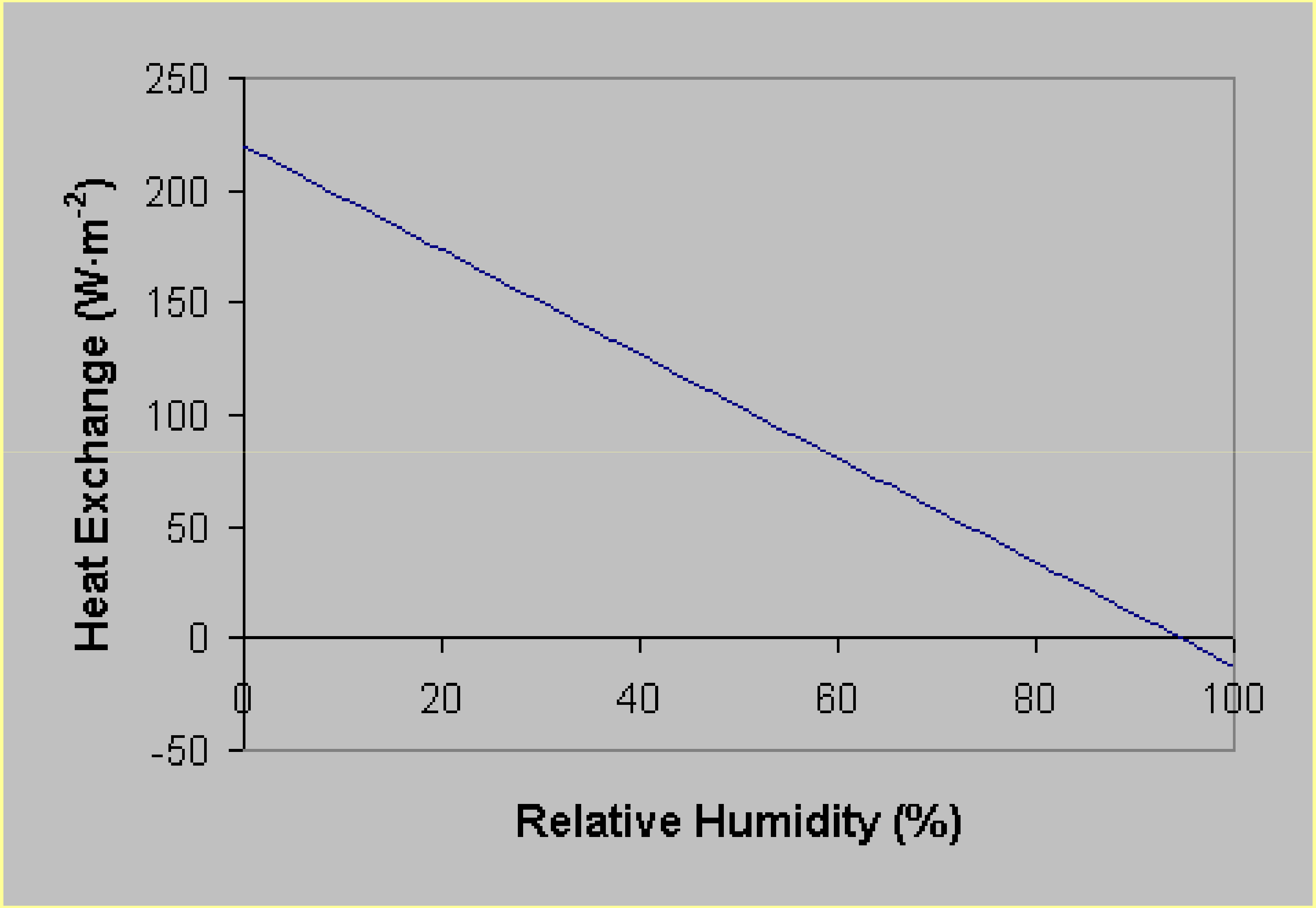
β - evaporation coefficient , [kg/m^{2}sPa]

A_{du} - Du Bois area ,[m^2]

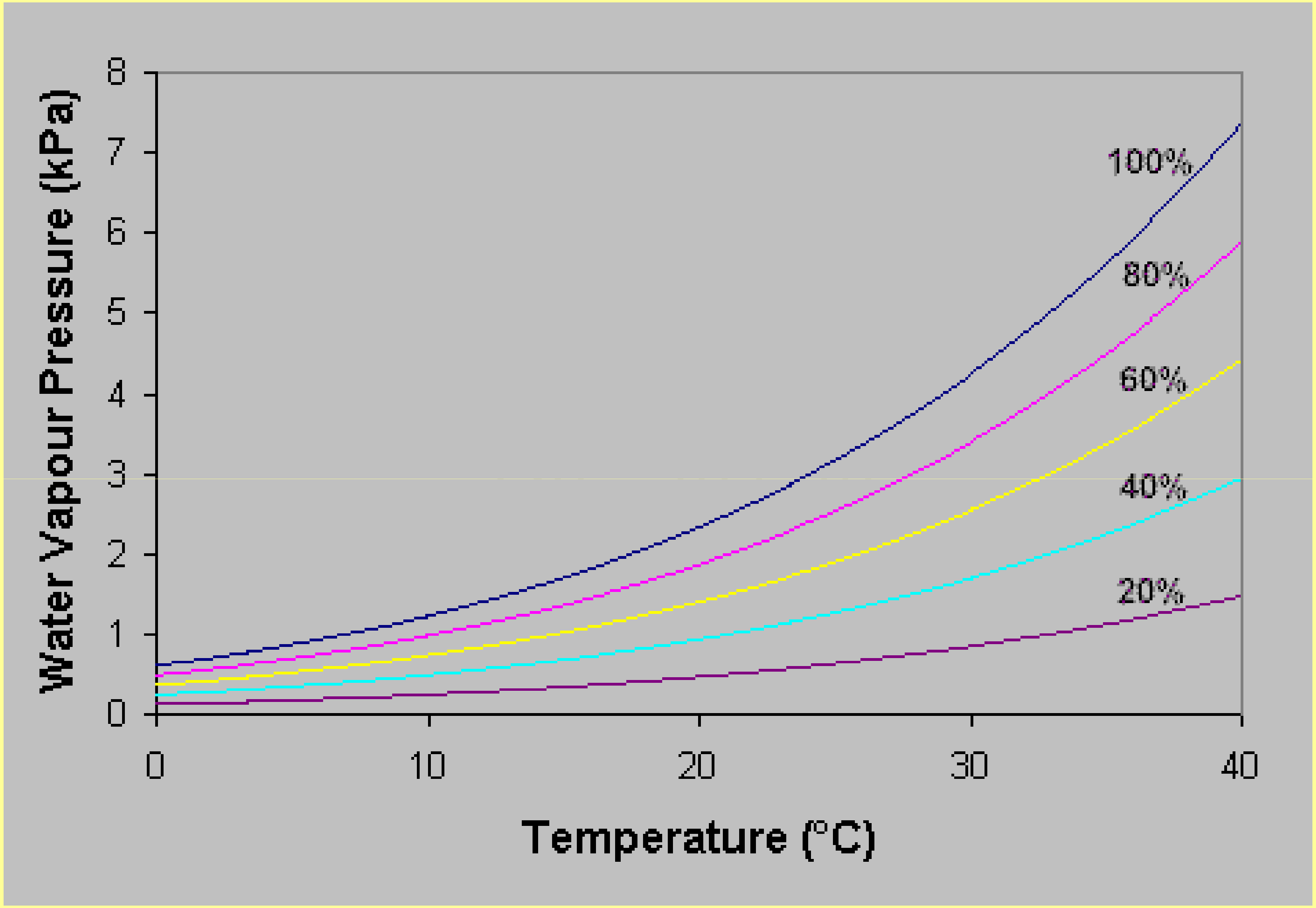
p_s, p_{sw} – pressure of saturated vapor, pressure of vapour at ambient temperature, [Pa]

r - heat of vaporization of water, [J/kg]

φ - the relative humidity of the gas



Heat Exchange (W·m-2)
250
200
150
100
50
0
-50
0
20
40
60
80
100
Relative Humidity (%)



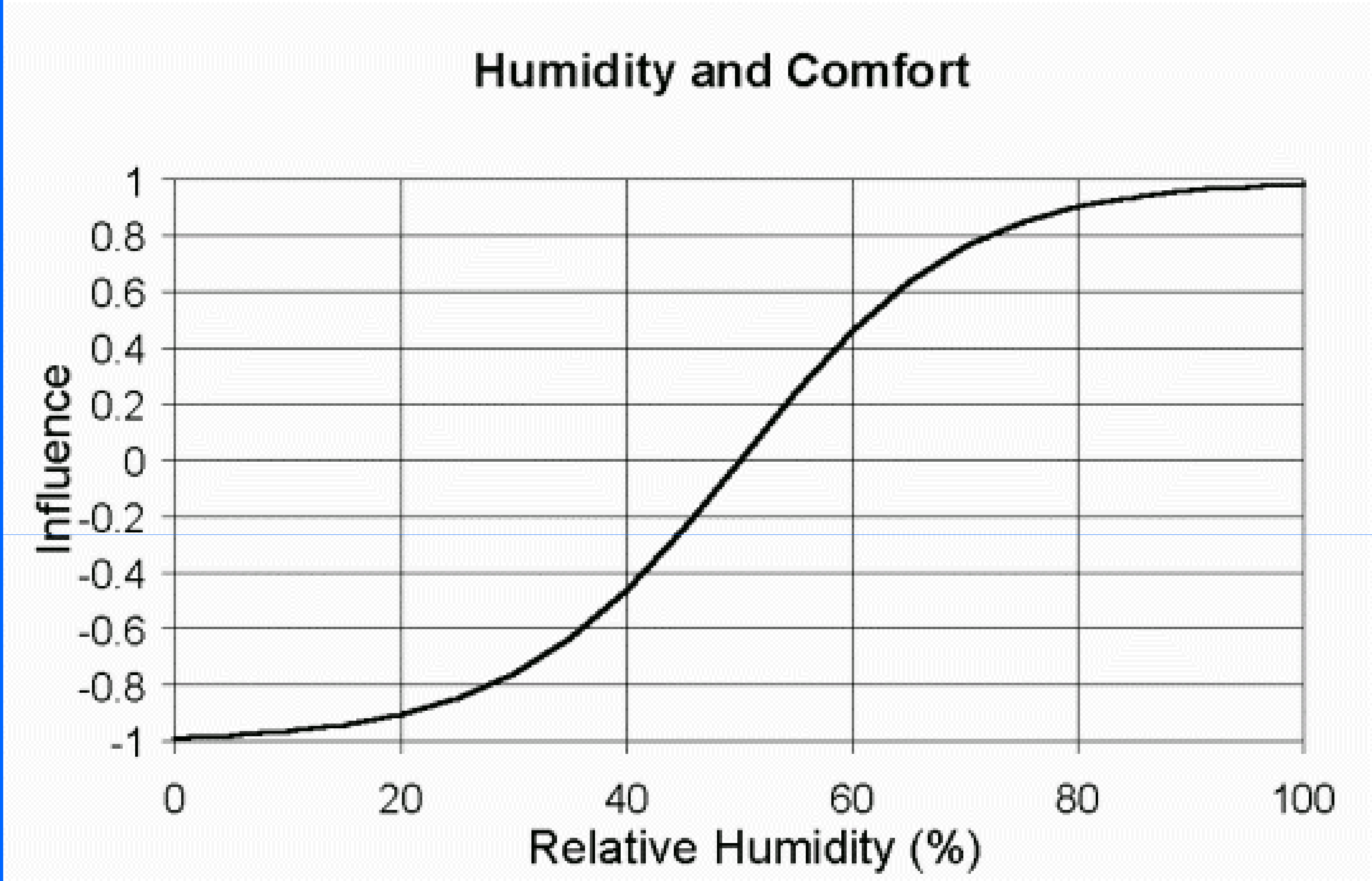
Water Vapour Pressure (kPa)
0
1
2
3
4
5
6
7
8
0
10
20
30
40
Temperature (°C)
100%
80%
60%
40%
20%

Relative Humidity φ, [%]

MILEDEW

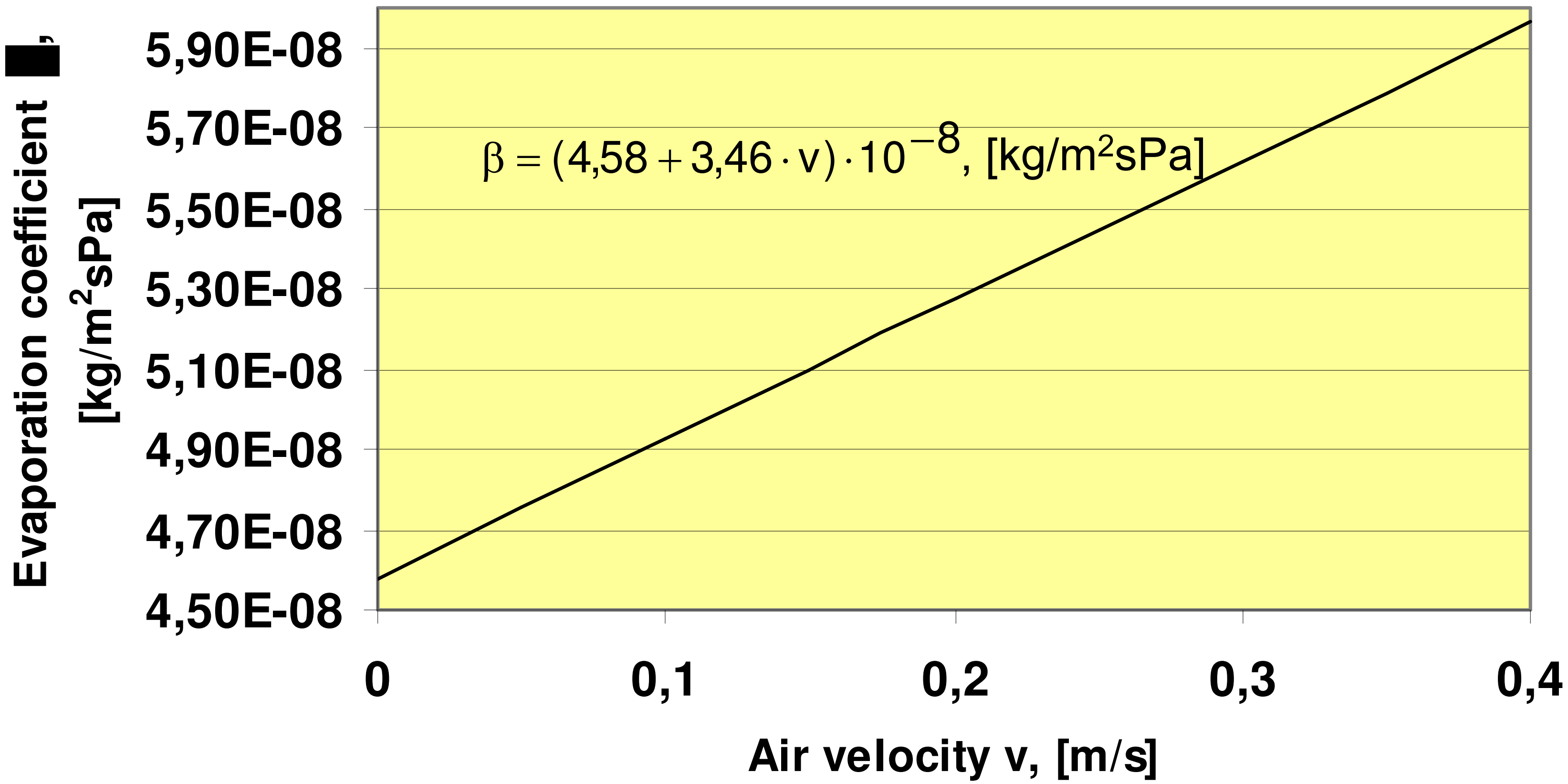
Optimum Relative Humidity φ= 45% - 55%

http://www.gasairconditioning.org/relative_humidity_chart.htm



http://www-fa.upc.es

LEARNING ABOUT HUMIDITY IN THERMAL COMFORT
ROSET, J.,MARINCIC, I.; OCHOA, J.M.; SERRA , R.



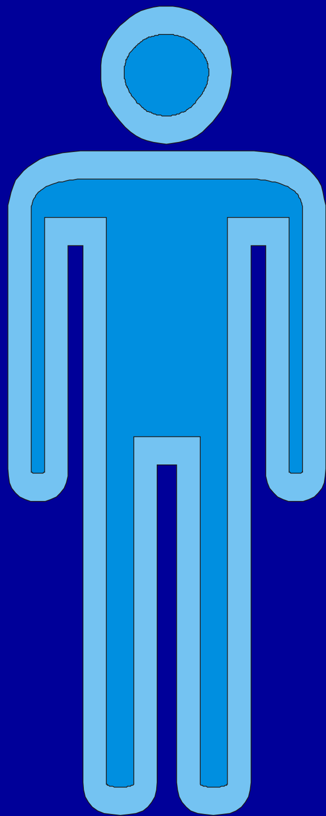
Source : Andjulovici A., Georgescu S.- Confortul termic in cladiri,Ed.Tehnica, Bucuresti 1966, Arkady , Warsaw 1971.

• HEAT LOSS BY EVAPORATION OF SWEAT

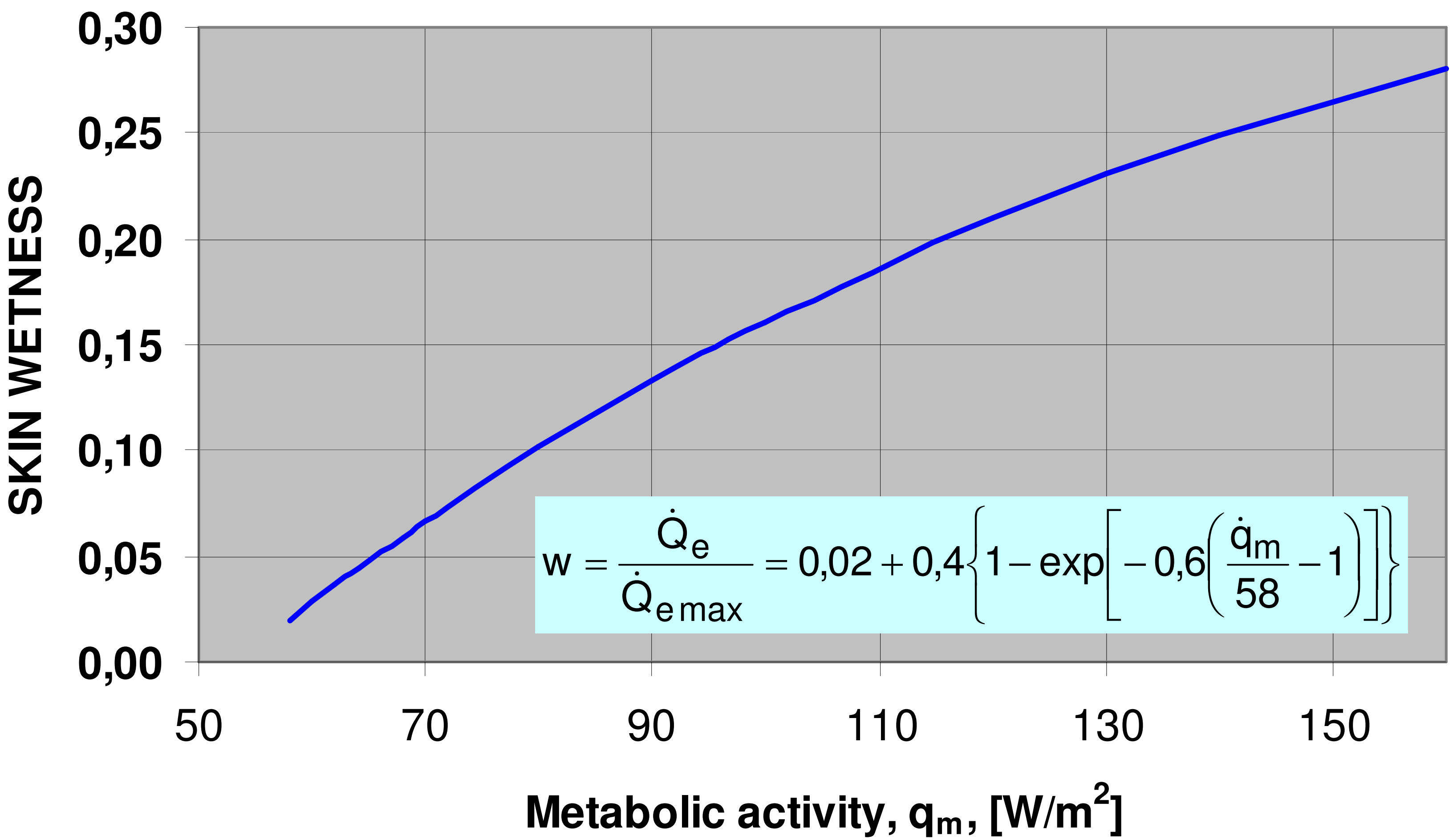
$$\dot{Q}_e = w \cdot \beta \cdot r \cdot A_{Du} \cdot (p_s - \varphi \cdot p_{sw}) \qquad (14)$$

Skin wetness is defined as:

$$w = \frac{\dot{Q}_e}{\dot{Q}_{e\,max}} = 0{,}02 + 0{,}4 \cdot \left\{1 - \exp\left[-0{,}6 \cdot \left(\left(\frac{\dot{q}_m}{58}\right) - 1\right)\right]\right\} \qquad (15)$$



Azer N.Z, Hsu S. –The use of modeling human response in the analysis of thermal comfort of indoor environmens. Proceedings of a Symposium held at the National Bureau of Standards, February, 1977.



SKIN WETNESS
0,30
0,25
0,20
0,15
0,10
0,05
0,00
50
70
90
110
130
150
Metabolic activity, q_m, $[W/m^2]$
$w = \frac{\dot{Q}_e}{\dot{Q}_{e\,max}} = 0,02 + 0,4\left\{1 - \exp\left[-0,6\left(\frac{\dot{q}_m}{58} - 1\right)\right]\right\}$

RESPIRATORY HEAT LOSS

- LATENT RESPIRATON HEAT LOSS
- DRY RESPIRATION HEAT LOSS

•DRY RESPIRATORY HEAT LOSS

Dry respiratory heat loss results from the difference between the expired and inspired gas temperatures:

$$\dot{Q}_j = \dot{m} \cdot c_p \cdot A_{Du} \cdot (t_{ex} - t_i) \quad (16)$$

c_p - specific heat at the constant pressure, J/kgK

$\dot{m}$ - mas rate of gas, kg/s

t_{ex}, t_i - the temperature of the expired, inspired gas, ^oC

• LATENT RESPIRATORY HEAT LOSS

$$\dot{Q}_u = \dot{m} \cdot r \cdot (X_{ex} - X_{in}) =$$

$$= 1{,}43 \cdot 10^{-6} \cdot \dot{q}_m \cdot A_{Du} \cdot (X_{ex} - X_{in}) \quad (17)$$

$\dot{m}$ - mas rate of gas, kg/s

r - heat of water evaporation , kJ/kg,

X_{ex}, X_{in} - humidity ratio of the expired, inspired gas

- **HEAT CONDUCTED THROUGH THE CLOTHING**

$$\dot{Q}_p = \frac{(t_s - t_{cl})A_{Du}}{R_{cl}} \quad (18)$$

$$I_{cl} = \frac{R_{cl}}{0{,}155} \quad (19)$$

$$\mathbf{1clo = 0{,}155\ \frac{m^2 K}{W}} \quad (20)$$

A_{Du} - Du Bois area, m^2
I_{cl} - thermal resistance from skin to the surface of the clothing, clo,
t_{cl} , t_s - the temperature of the outer surface of the clothing, skin ^oC,
R_{cl} - thermal resistance from skin to the surface of the clothing, m^2K/W

Thermal Insulation of Clothing

The addition of thermal resistance due to clothing affects heat transfer mechanisms between the human body and the environment.

1 clo maintains sedentary man (1 met) indefinitely comfortable at 21°C, 50% RH, 0.01 m/sec.

Clo- value is a numerical representation of a clothing ensemble's thermal resistance, 1 clo = 0.155 m^2K / W.

Thermal resistance of different clothyhing ensembles		
Clothing Combination	**I_{cl}, [clo]**	**R_{cl} [m2-K/W]**
Naked	0	0
Shorts	0.1	0.016
Tropical ensemble (briefs, shorts, open- necked shirt, light socks and sandals)	0.3	0.047
Light summer ensemble (briefs, long lightweight trousers, short-sleeved shirt, light socks and shoes)	0.5	0.078
Working attire (briefs, long-sleeved shirt, trousers, woolen socks and shoes)	0.8	0.124
Typical indoor winter attire (briefs, long-sleeved shirt, trousers, long-sleeveds weater, heavy socks and shoes)	1.0	0.155
Heavy indoor winter attire (long underwear, long-sleeved shirt, suit with vest, heavy socks and shoes)	1.5	0.233

Source:Thermal Comfort, P. O. Fanger, McGraw-Hill, New York, 1970.

• **HEAT LOSS BY RADIATION**

• **HEAT LOSS BY RADIATION**

$$\dot{Q}_R = 4 \cdot 10^{-8} \cdot f_{cl} \cdot A_{Du} \cdot [(t_{cl} + 273{,}15)^4 - (t_r + 273{,}15)^4] \quad (21)$$

f_{cl} - the ratio of the surface area of the clothed body to the surface of the nude body,

t , t_r - the comfort , mean radiant temperature, ^oC,

A_{Du} - Du Bois area , m^2

- **HEAT LOSS BY CONVECTION**

$$\dot{Q}_K = \alpha f_{cl} A_{Du} (t_{cl} - t) \qquad (22)$$

C_p - specific heat at the constant pressure, J/kgK

f_{cl} - clothing factor,

p - total pressure of the breathing mixture, Pa,

v - relative velocity of gas , m/s

t , t_r - the comfort , mean radiant temperature, ^oC,

λ - thermal conductivity of the breathing gas , W/mK,

x_i - molar fraction of the inert gas

Convective heat transfer coefficient:

$$\alpha = f(c_p, \lambda, \eta, \rho, t, v) \qquad (23)$$

- **FREE CONVECTION,**
- **MIXED CONVECTION,**
- **FORCED CONVECTION**

C_p - specific heat at the constant pressure, J/kgK

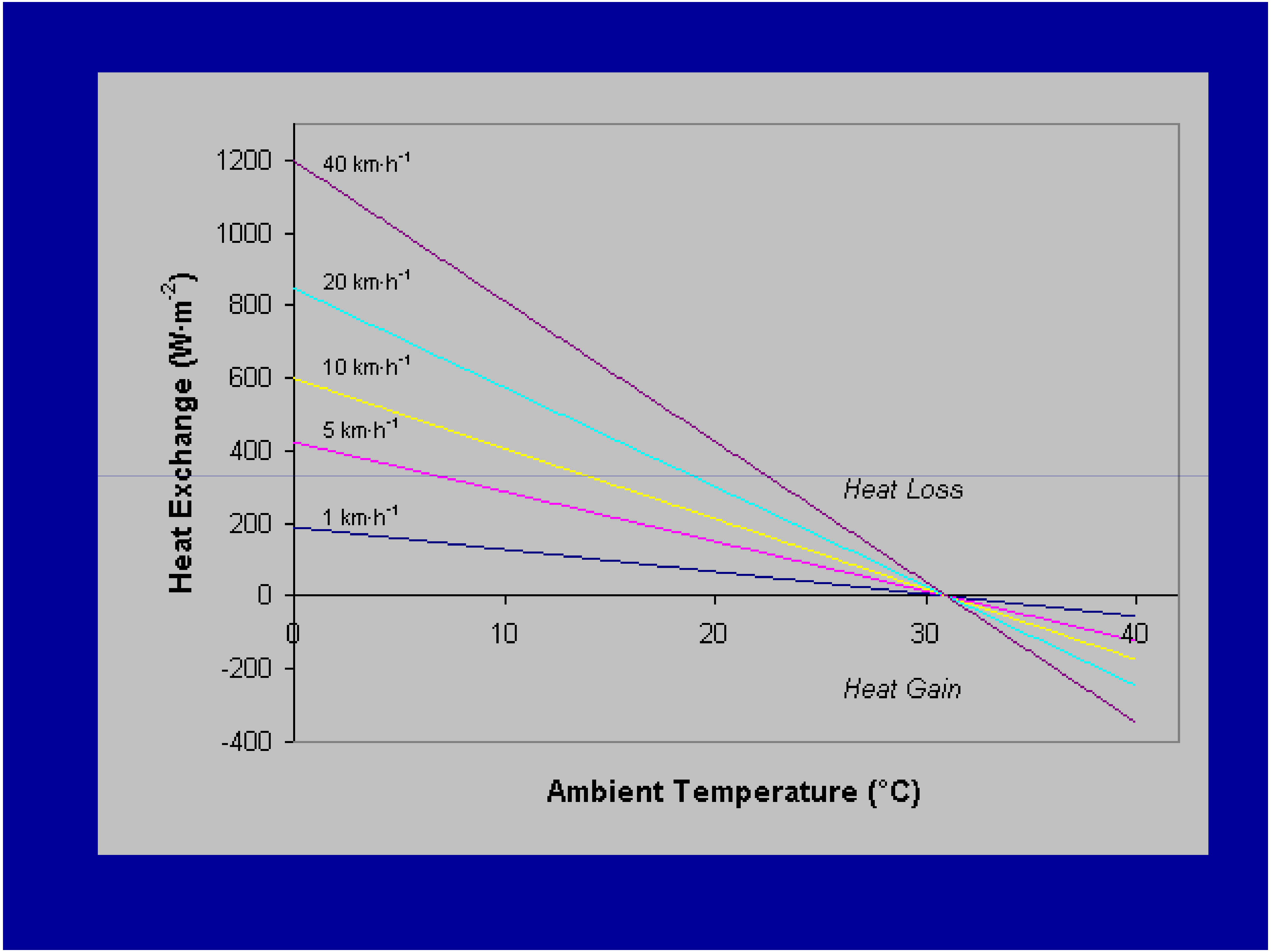
v - relative velocity of gas , m/s

t - the comfort , mean temperature, ^{o}C

λ - thermal conductivity of the breathing gas , W/mK,

v - air velocity, m/s,

η - dynamic viscosity coefficient , kg/ms



Heat Exchange (W·m-2)
1200
1000
800
600
400
200
0
-200
-400
40 km·h-1
20 km·h-1
10 km·h-1
5 km·h-1
1 km·h-1
Heat Loss
Heat Gain
0
10
20
30
40
Ambient Temperature (°C)

THE COMFORT EQUATION

$$(1-\eta)\cdot\dot{q}_m - \beta\cdot r\cdot(p_s - \varphi\cdot p_{sw}) +$$

$$- w\cdot\beta\cdot r\cdot(p_s - \varphi\cdot p_{sw}) -$$

$$+ 1{,}43\cdot 10^{-6}\cdot\dot{q}_m\cdot r\cdot(X_{ex} - X_{in}) - \dot{m}\cdot c_p\cdot(t_{ex} - t) =$$

$$= \frac{t_s - t_{cl}}{0{,}155\cdot I_{cl}} = \alpha\cdot f_{cl}\cdot A_{Du}\cdot(t_{cl} - t) +$$

$$+ 4\cdot 10^{-8}\cdot f_{cl}\cdot A_{Du}\cdot[(t_{cl} + 273{,}15)^4 - (t_r + 273{,}15)^4]$$

(23)

THE COMFORT TEMPERATURE

$$t = f(v, \dot{q}_m, \varphi, t_r, I_{cl})$$

ENVIRONMENTAL VARIABLES

-AIR TEMPERATURE: t

-MEAN RADIANT TEMPERATURE: t_r

-AIR RELATIVE VELOCITY: v

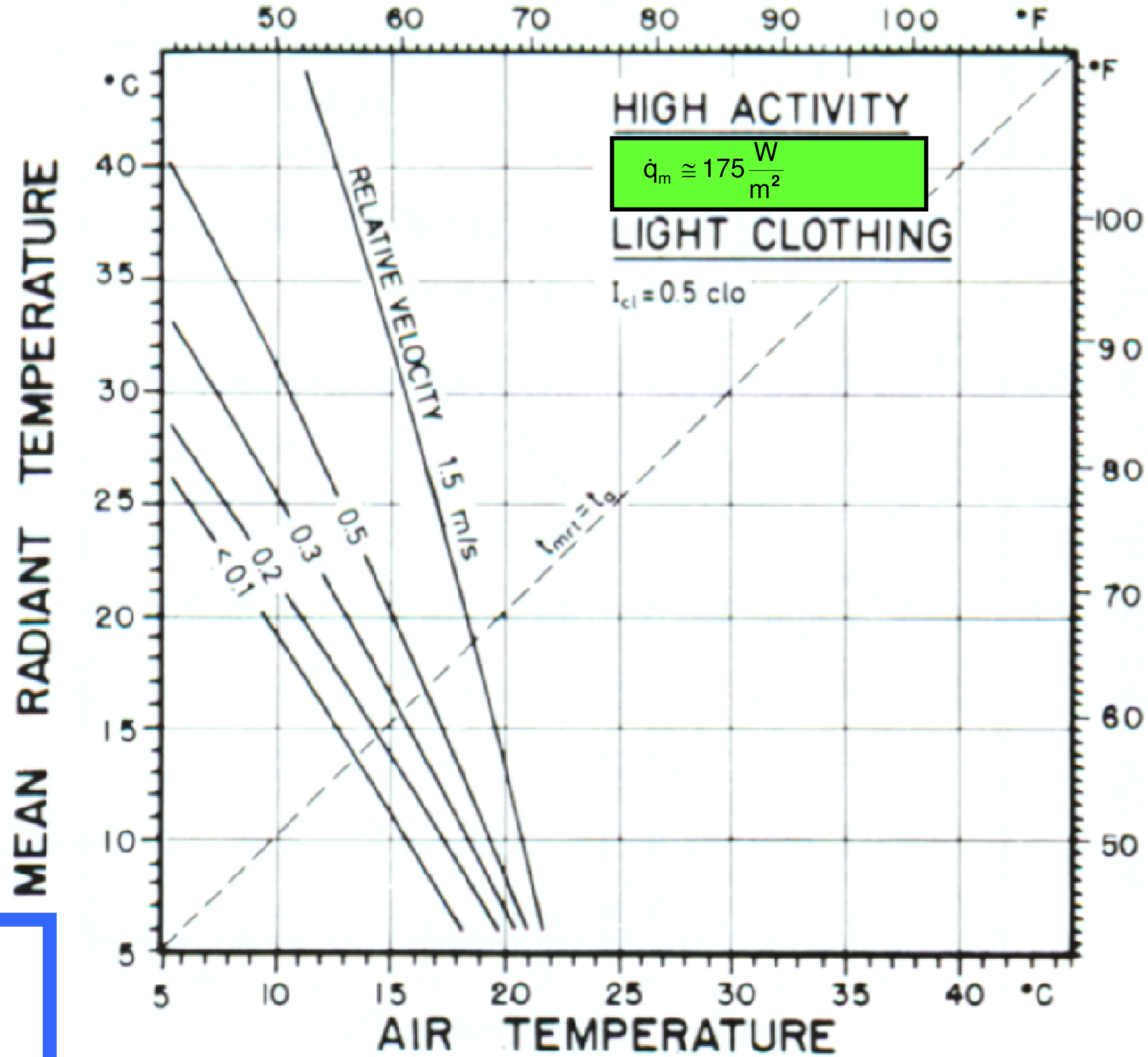
-HUMIDITY OF AIR: φ

HUMAN VARIABLES

- ACTIVITY OF MAN: $\dot{V}_{O_2}, \dot{q}_m$
- PARAMETERS OF THE BODY (MASS, HEIGHT): m, h, A_{Du}

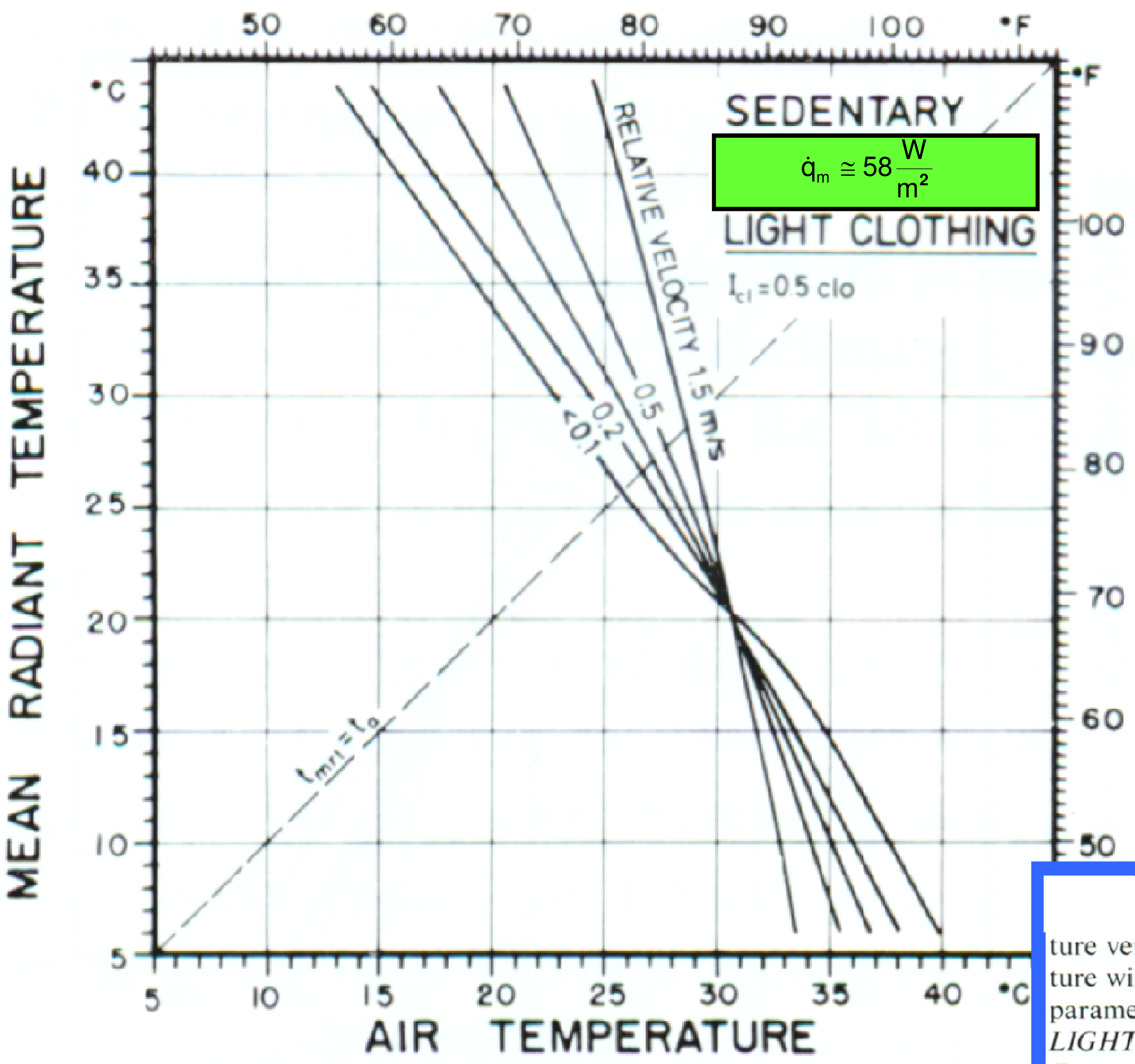
CLOTHING VARIABLES

- THERMAL RESISTANCE OF THE CLOTHING: I_{cl}
- MOISTURE PERMEATION FACTOR : F_{pcl}



Comfort lines (air temperature versus mean radiant temperature with relative air velocity as parameter) for persons with *LIGHT CLOTHING* (I_{cl} = 0.5 clo, f_{cl} = 1.1) at three different activity levels (rh = 50%).

Source: Thermal Comfort, P. O. Fanger, McGraw-Hill, New York, 1970.



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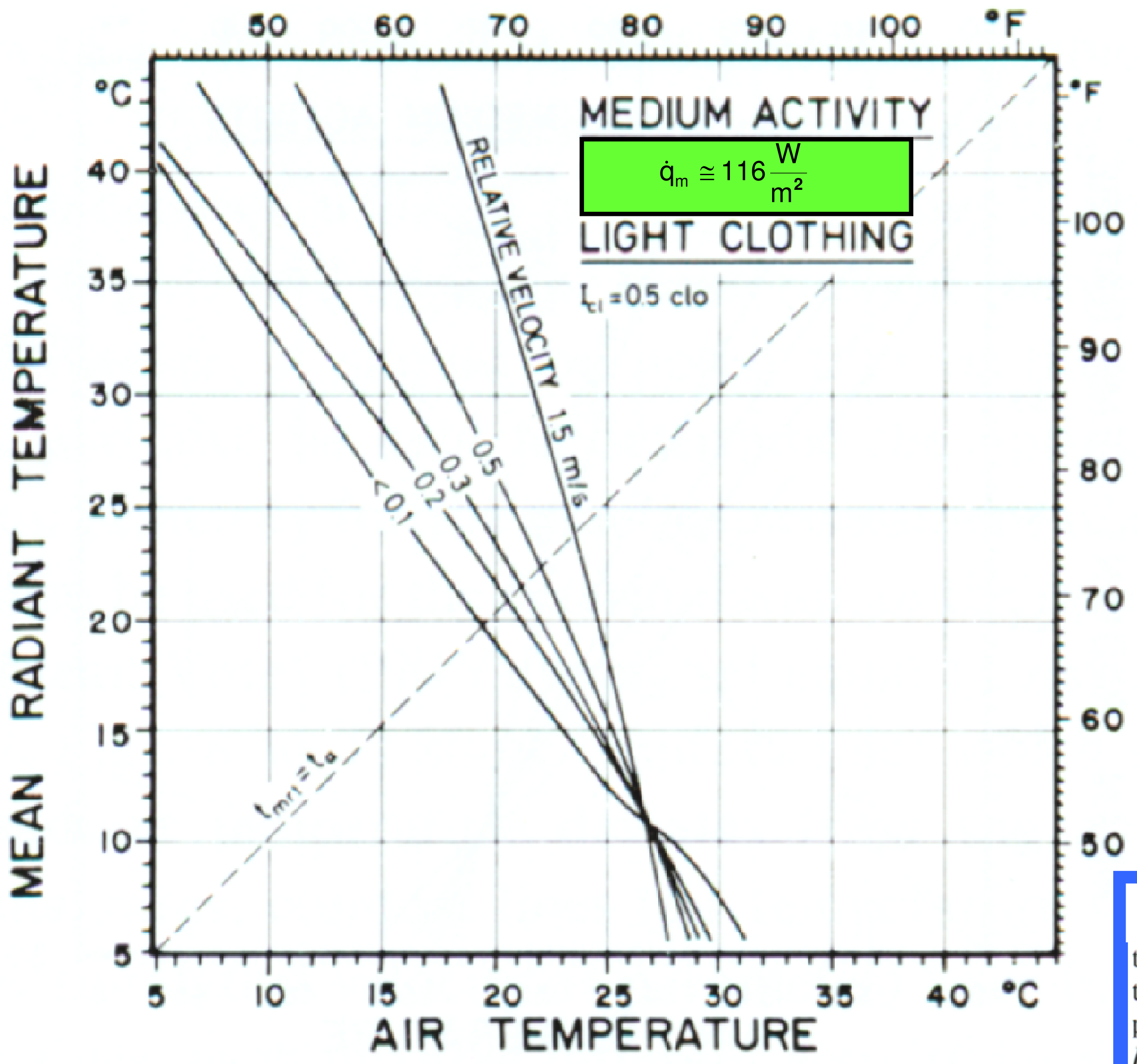

Comfort lines (air temperature versus mean radiant temperature with relative air velocity as parameter) for persons with *LIGHT CLOTHING* (I_{cl} = 0.5 clo, f_{cl} = 1.1) at three different activity levels (rh = 50%).

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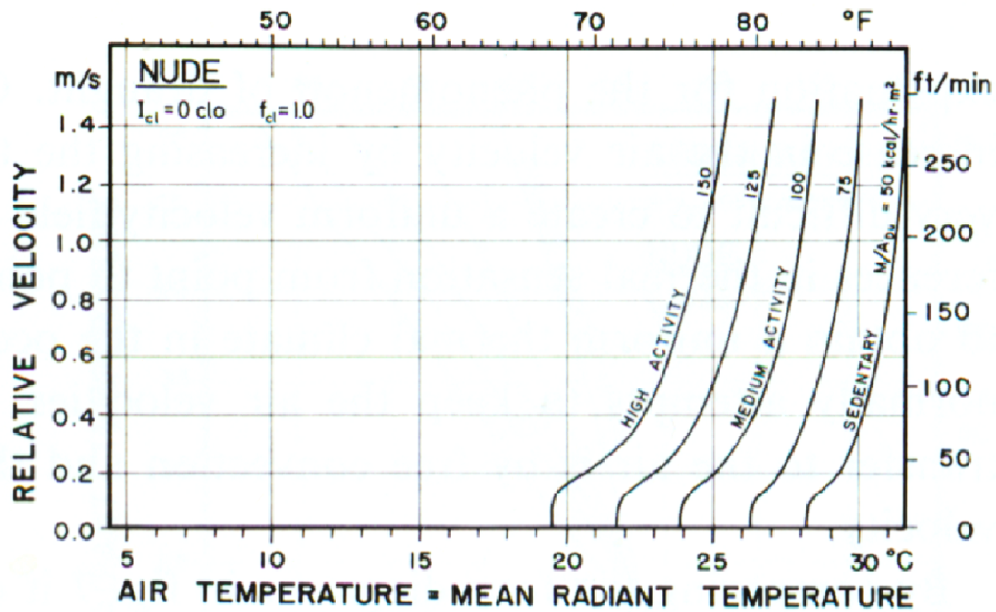
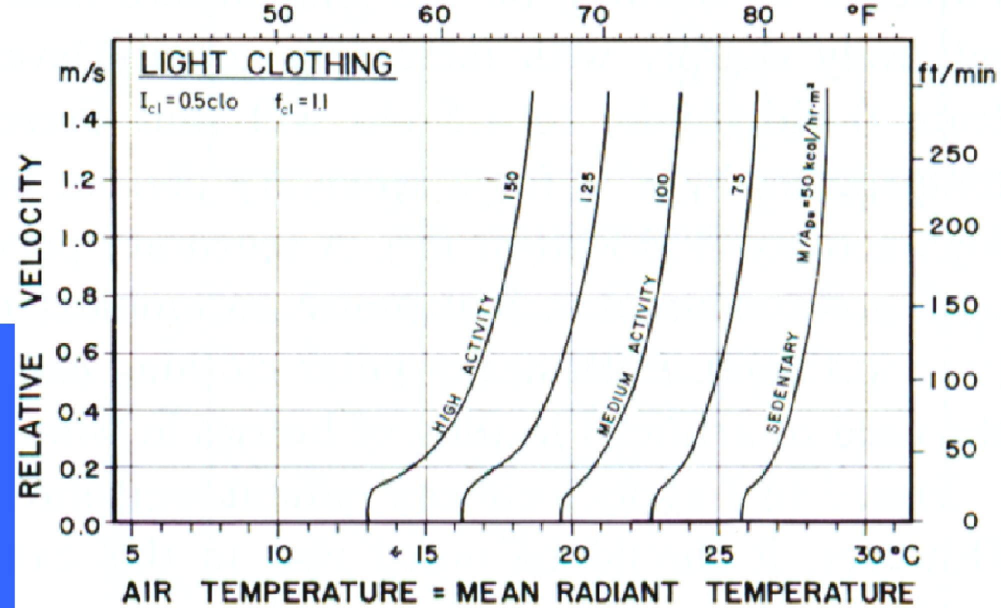
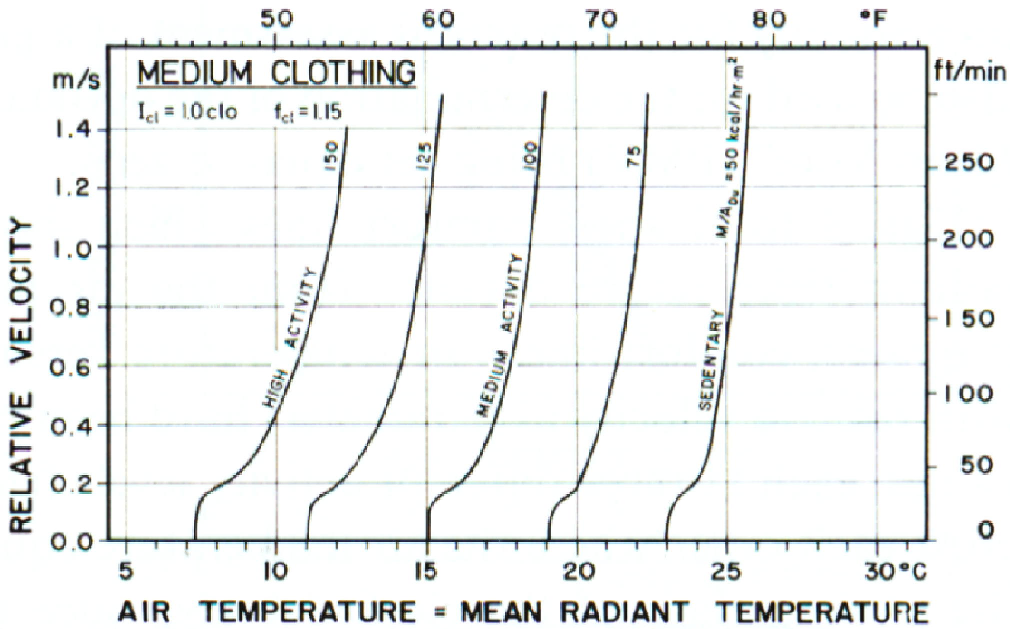
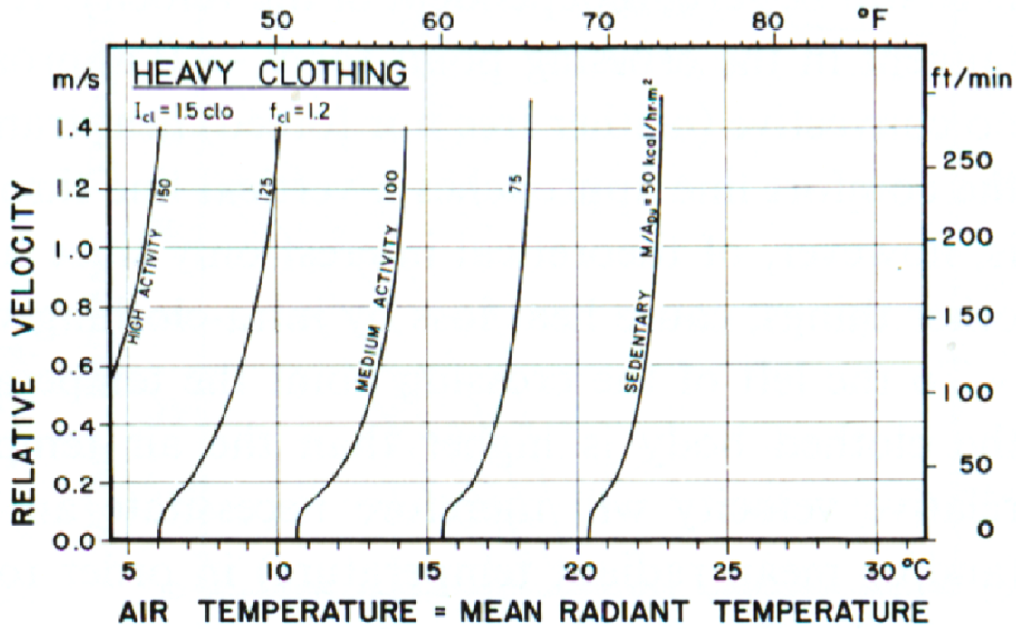




Comfort lines (ambient temperature versus relative air velocity with activity level as parameter) at 4 different clo-values (rh = 50 %).

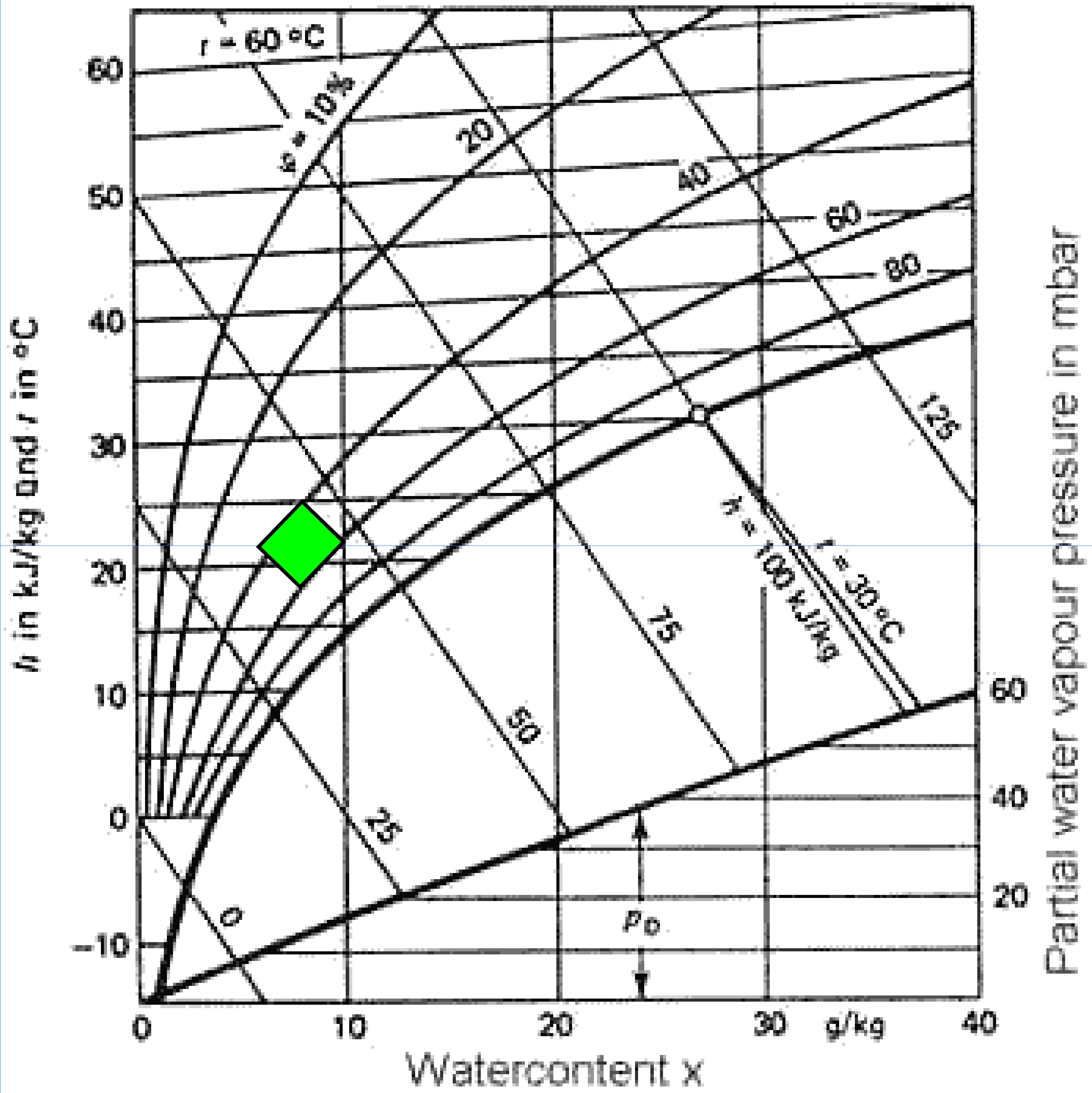
Source: Thermal Comfort, P. O. Fanger, McGraw-Hill, New York, 1970.





Comfort lines (ambient temperature versus relative air velocity with activity level as parameter) at 4 different clo-values (rh = 50 %).

Source: Thermal Comfort, P. O. Fanger, McGraw-Hill, New York, 1970.



COMFORT ZONE

ENVIRONMENTAL INDICES

- MEAN RADIANT TEMPERATURE
- PREDICTED MEAN VOTE (PMV) INDEX
- PREDICTED PERCENTAGE DISSATISFIED (PPD)
- INDEX LOWEST POSSIBLE PERCENTAGE DISSATISFIED (LPPD) INDEX
- THE OPERATIVE TEMPERATURE

- **MEAN RADIANT TEMPERATURE**

the uniform surface temperature of a black enclosure with which an individual exchanges the same heat by radiation as the actual environment considered.

PREDICTED MEAN VOTE (PMV) INDEX

The PMV index predicts the mean response of a large group of people according to the ASHRAE thermal sensation scale:

Table 1.1-The ASHRAE scale

+3	Hot
+2	Warm
+1	Slightly Warm
0	Neutral
-1	Slightly Cool
-2	Cool
-3	Cold

$$PMV = (0.303e^{-0.036M} + 0.028)L$$

where

M(q_m) - metabolic rate

L - thermal load defined as the difference between the internal heat production and the heat loss to the actual environment for a person hypothetically kept at comfort values of skin temperature and evaporative heat loss by sweating at the actual activity level.

PMV method can be described as a simple energy balance, in which a human comfort index is estimated from a difference in heat generated by the human body with the heat lost from the body to the surroundings.

PREDICTED PERCENTAGE DISSATISFIED (PPD) INDEX

- PPD is a quantitative measure of the thermal comfort of a group of people at a particular thermal environment.

- Fanger related the PPD to the PMV as follows:

$$PPD = 100 - 95e^{-(0.03353PMV^4 + 0.2179PMV^2)}$$

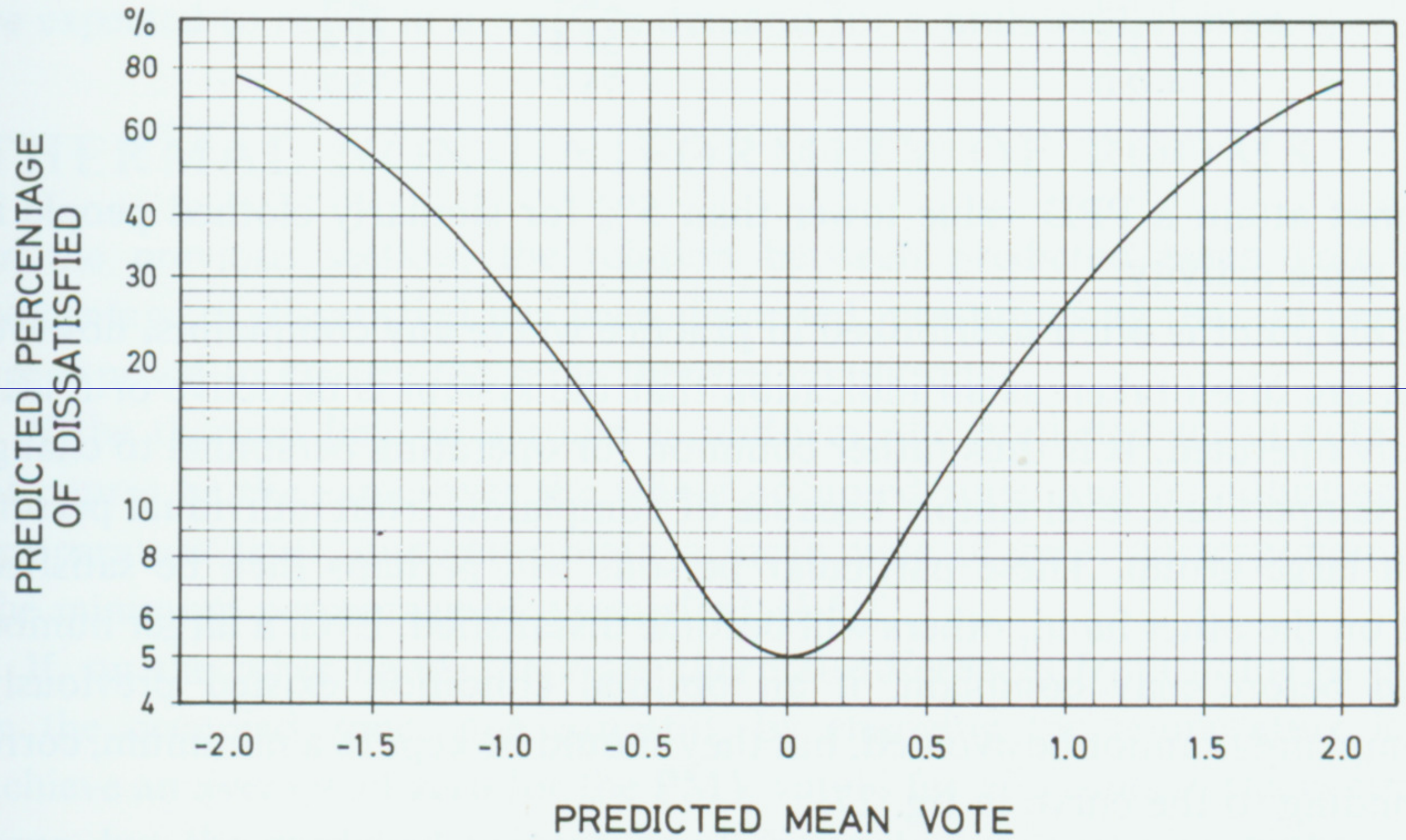


Fig. 27. Predicted Percentage of Dissatisfied (PPD) as a function of Predicted Mean Vote (PMV).

Source:Thermal Comfort, P. O. Fanger, McGraw-Hill, New York, 1970.

LOWEST POSSIBLE PERCENTAGE DISSATISFIED (LPPD) INDEX

- The LPPD is a quantitative measure of the thermal comfort of a room as a whole for a group of people in a thermally non-uniform environment.
- It is more useful for large rooms than for small one.
- As a recommended design target, LPPD is not to exceed 6%.

THE OPERATIVE TEMPERATURE

- Is one of several parameters devised to measure the air's cooling effect upon a human body

- It is equal to the dry-bulb temperature at which a specified hypothetical environment would support the same heat loss from an unclothed, reclining human body as the actual environment. In the hypothetical environment, the wall and air temperatures are equal and the air movement is 7.6 cm/s.

From experiment it has been found that the operative temperature

$$T_0 = 0.48t_r + 0.19[\sqrt{vt_a} - (\sqrt{v} - 2.76)t_s]$$

where:

- t_r is the mean radiant temperature; (° C),
- t_a is the mean air temperature; (° C),
- t_s is the mean skin temperature; (° C),

and v is the airspeed, cm/s.

http://en.wikipedia

Effective Temperature

Effective temperature is the uniform temperature of a radiantly black enclosure at 50% relative humidity, in which an occupant would experience the same comfort, physiological strain and heat exchange as in the actual environment with the same air motion.

http://www.healthyheating.com/solutions.htm