Impact of Ventilating and Asymmetrical Radiation on Human Beings in Hot Environment.

Ph.D. Dissertation

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A szellőzés és aszimmetrikus sugárzás hatása az emberekre meleg környezetben

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Az iskolavezető jóváhagyása **A témavezető jóváhagyása**

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CHAPTER 1

Introduction

In tropical countries such as Libya there are two basic problems with the thermal comfort:

- high indoor temperature in both traditional and modern buildings.
- unfavorable asymmetrical radiation of some of the external boundary walls heated up by the fierce sunshine.

Aware of these two problems I become interested as an Msc student at the Budapest University of Technology and Economics (BME) in comfort theory whose topics include research projects such as these. I applied for the PhD School of BME in 2003 Until 2007, my topic leader was Dr.László Bánhidi and during that time I carried out in-situ meeasurements in Libya and laboratory measurements with manikin in the BME Laboratory.

In 2007 the situation changed at BME (a merger of departments) so in agreement with BME I applied and was accepted at the PhD School of Szent István University. My topic leader is Professor Dr. István Barótfi. I passed my PhD exams and finished my dissertation here.

During my PhD studies I participated at several recognized international conferences and deliver the lectures. The number of my presentations (as shown by references) complies with the requirements.

1.1. The main topics of PhD work are:

- Generally about the thermal comfort
- Situation of the thermal comfort due regards in hot environment
- Impact of asymmetrical radiation and airflow
- Impact of airflow and asymmetrical radiation on human heat exchange
- Field studies in Libya
- Laboratory investigations

1.2. Determination of thermal comfort today

By the simplification:

comfort means more than maintaining the right temperature and it does not have to mean higher energy cost. In fact, with a perfect climate system, a comfortable indoor space can actually improve your family's health and well-being, cut energy bills, and save wear and tear non your heating and cooling equipment.

Thermal comfort has been defined by ASHRAE, 55-92 [1] as "that condition of mind which expresses satisfaction with the thermal environment" and as such will be influenced by personal differences in mood, culture and other individual, organisational and social factors.

About one third of the world's energy consumption is used to provide thermal comfort for man. It is no wonder, therefore, that efforts towards energy conservation have led to an increased interest in man's comfort condition order to assess the human response to different conservation strategies.

1.3. The theoretical basic of thermal comfort

Fanger P.O [2] elaborated the thermal comfort theory for example Fangar equation and diagrams but He made the theory of air quality (IAQ) with the (olf-decipol) system for the comfortable indoor space too Several factors affect the level of our personal comfort. In a comfortable space we have to take into consideration many factors never too damp. Because it well balanced, it is energy efficient, too. This conditions must be fulfilled to maintain comfort. One is that actual combination of skin temperature and the body's core temperature provide a sensation of thermal neutrality. The second is the fulfillment of the energy balance of the body the heat produced by the metabolism should be the amount of heat lost from the body. The relationship between the parameters, core body temperature and activity, which results in a thermally neutral sensation, are based on a large number of experiments. The thermal comfort is the primary aim of most heating and air-conditioning systems it is not surprising that over the years a considerable number of studies have been made with the purpose of investigating comfort conditions.

In modern industrial society man spends the greater part (70-80 %) of his life indoors. The main purpose of most heating air-conditioning system is to provide thermal comfort and acceptable air quality for human beings. For the design and operation of such systems and requirements for an acceptable thermal environment. Thermal comfort is defined as that condition of mind, which expresses satisfaction with the thermal environment (ASHRAE 55-92) [1] and CR 1752 [3]).

The first requirement for thermal comfort : thermal natural reality for the body .

The most important variables, which influence the condition of thermal comfort, are:

Personal factors:

- Activity level, M (met, W/m^2)
- Thermal resistance of the clothing, I_{cl} (clo, $m^2\ C^o/W$)

Environmental parameters:

- Air temperature, t_a
- Relative air velocity, v_r
- Air humidity, r_a (water vapour pressure)
- Mean radiant temperature, t_{mrt}

1.4. Heat balance comfort equation [3]

The thermal comfort equation for comfortable skin temperature and sweat production can be combined with the equation for the body's energy balance to derive the Comfort Equation. This equation describes the connection between the measurable physical parameters and thermally neutral sensation as experienced by the "average" person.

The comfort equation provides us with an operational tool which by measuring physical parameters enables us to evaluate under which conditions thermal comfort may be offered in home and workplace. The Comfort Equation derived by Fanger P.O [2] . is too complicated for manual arithmetic and is normally solved using a computer. The full equation can be seen in eq (1.4).

In the body has a chemicel burnong the so callect metabolism (M) which devided two main parts H and W

$$M = H + W \qquad (W/m^2) \qquad (1.1)$$

H – internal heat production in the human body

W- mechanical work

$$\eta = \frac{W}{M} \qquad (1.2)$$

$$H = M(1-\eta) \qquad (W/m^2) \qquad (1.3)$$

η -external mechanical efficiency (max 20 %)

M- Metabolic rate (W/m^2)

The equation reveals that the temperature of the surfaces in the enclosure where a person is has a huge influence on thermal sensation.

$H - Ed - E_{sw} - E_{re} - L = K = R + C$ (1.4)

E_d – Heat loss by water vapour diffusion through the skin

E_{sw} – Heat loss by evaporation of sweat from the surface of the skin

E_{re} – Latent respiration heat loss

L – Dry respiration heat loss

K – Heat transfer from the skin to the outer surface of the clothed body (conduction through the clothing)

R – Heat loss by radiation from the outer surface of the clothed body

C – Heat loss by convection from the outer surface of the clothed body

The calculation of these parameters are by the original in kcal/h:

$$E_{sw} = A_{Du} f\left(\frac{H}{A_{Du}}\right) \quad (1.5)$$

$E_d = \lambda_m A_{Du} (ps-pa)$ (kcal/hr) (1.6)

λ = 575 kcal/kg = heat of vaporization of water (at 35 ^oC)

m = permeance coefficient of the skin (kg/hr m^2 mmHg)

p_s = saturated vapour pressure at skin temperature (mmHg)

p_a = vapour pressure in ambient air (mmHg)

$E_{re} = V (W_{ex}-Wa)$ (1.7)

V = pulmonary ventilation (kg/hr)

W_{ex} = humidity ratio of the expiration air. (kg water/kg dry air)

W_a = humidity ratio of the inspiration air. (kg water/kg dry air)

Substituting the expressions for V and W_{ex}-Wa in eq (1.7), one obtains the following formula for the latent heat respiration heat loss:

$E_{re} = 0.0023 M (44-pa) \lambda$ (1.8)

$L = 0.0014 M (34 - t_a)$ (kcal/hr) (1.9)

$$R = A_{eff}\,\varepsilon\sigma\left[(t_{el}+273)^4 - (t_{mrt}+273)^4\right].\ \text{(W/m}^2\text{/hr)} \qquad (1.10)$$

A_{eff} – the effective radiation area of the clothed body. (m2)

ε – the emittance of the outer surface of the clothed body

σ - the Stefan- Boltzmann constant: 4.96 . 10^{-8} (Kcal / m^2 hr^o K^4)

t_{mrt} – the mean radiant temperature (C^o)

we could calculated the Aeff from the following equation

$A_{eff} = f_{eff}\, f_{cl}\, A_{Du}$ (m^2) (1.11)

f_{eff} – the effective radiation area factor

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

A_{Du}- DuBois area (the surface area of the nude body (m^2)

$A_{Du} = 0.203\ G^{0.425}\ H^{0.725}$

G- the weight of body

H- the height of body

$C = A_{Du}\, f_{cl}\, h_c\, (t_{cl} - t_a)$ (1.12)

h_c – the convection heat transfer coefficient (W / m^2 hr^o C)

$$hc = 2.05(t_{cl} - t_a)^{0.25} \qquad (1.13)$$

$$t_s = 35.7 - 0.032\frac{H}{A_{Du}} \qquad (1.14)$$

In a steady-state heat balance, the heat energy produced by metabolism equals the rate of heat transferred from the body by convection, radiation, evaporation, and respiration. If the metabolism rate is not balanced momentarily by the sum of the transfer of heat, the body temperature will change slightly, providing thermal storage in the body. The complete equation for the heat balance becomes as following in equation. 1.15

$$\frac{M}{A_{Du}}(1-\eta)-0.35\left[43-0.061\frac{M}{A_{Du}}(1-\eta)-p_a\right]-0.42\left[\frac{M}{A_{Du}}(1-\eta)-50\right]-0.0023\frac{M}{A_{Du}}(44-p_a)-$$

$$0.0014\frac{M}{A_{Du}}(34-t_a)=3.4\times10^{-8}f_{cl}\left[(t_{cl}+273)^4-(t_{mrt}+273)^4\right]+f_{cl}h_c(t_{cl}-t_a)$$

$$t_{cl}=35.7-0.032\frac{M}{A_{Du}}(1-\eta)-$$

$$0.181_{cl}\frac{M}{A_{Du}}(1-\eta)-0.35\left[43-0.061\frac{M}{A_{Du}}(1-\eta)-p_a\right]-0.42\left[\frac{M}{A_{Du}}(1-\eta)-50\right]-0.0023\frac{M}{A_{Du}}$$

$$(44-p_a)-0.0014\frac{M}{A_{Du}}(34-t_a)=3.4\times10^{-8}f_{cl}\left[(t_{cl}+273)^4-(t_{mrt}+273)^4\right]+f_{cl}h_c(t_{cl}-t_a)$$

(1.15)

A double equation (1.15) expresses the internal heat production H minus the heat loss by evaporation from the skin ($E_d + E_{sw}$) and by respiration (E_{re} + L) is equal to the heat conduced through the clothing (K) and dissipated at the outer surface of the clothing by radiation and convection (R + C).

1.5. AMV - PMV- PPD [3]

AMV = The Actual Mean Vote (AMV) of overall thermal sensation for people with physical disabilities compared with those of people without physical disabilities.

The overall thermal Actual Mean Vote (AMV) ranged between ‘slightly cool’ and ‘slightly warm’ with inclination towards the ‘slightly cool’. All these four cases fulfilled the less than 20% unacceptability criterion (ASHRAE Standard 55 [1],

For the qualification of the subjective thermal feeling Fanger P.O [2], elaborated the so called PMV – PPD theory. What it is means PMV and PPD, well

PMV = Predicted mean vote

PPD = Predicted Percentage of Dissatisfied

The PMV-index predicts the mean value of the subjective ratings of a group of people in a given

environment.

The PMV scale is a seven-point thermal-sensation scale ranging from -3 (cold) to +3 (hot), where 0 represents the thermally neutral sensation.

To predict how many percentage of people are dissatisfied in a given thermal environment, the PPD-index (Predicted Percentage of Dissatisfied) has been introduced.

Even when the PMV-index is 0, there will still be sprcentages of individuals are dissatisfied with the temperature level, regardless of the fact that they are all dressed similarly and have the same level of activity.

we never could realized comfort parameters which are optimal for everybody - comfort evaluation differs from person to person.

The mathemathical equations of PMV and PPD are:

PMV equation: $PMV = (0.33\, e^{-0.36\,M} + 0.028).\ [(M - W) - H - E_{res} - C_{res}]$ (1.16)

PPD equation: $PPD = (100 - 95\, e^{-(0.03353\,.\,PMV^{4} + 0.2179.PMV^{2})})$ (1.17)

By this equations and laboratory investigations were elaborated the PMV – PPD diagram (fig. 1.1)

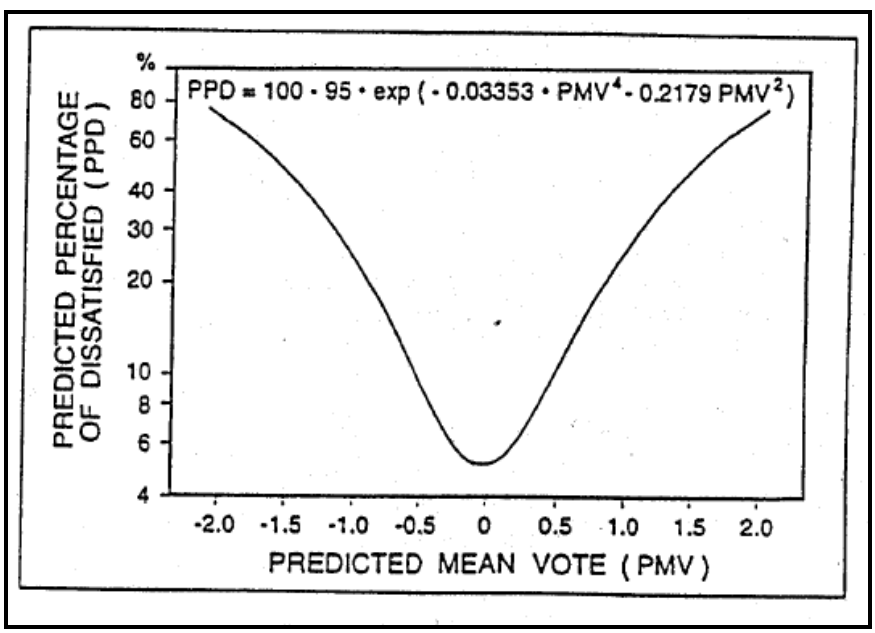


Fig 1.1 – Predicted percentage of dissatisfied (PPD) of predicted mean vote PMV

1.6. The standard of thermal comfort

1.6.1. The basic tables and figures

All the tables and figures from CR 1752 [3] collected the design criteria for spaces in different types of buildings for the occupancy listed in the table (1.1a) and (1.1b). For the thermal comfort dimensioning are elaborated the ASHRAE standard 55 [1] and CR 1752 [3] in Europe. We are working with European Standard. This Standard use two tables:

- the basic table (table 1.1a and 1.1b)
- the so called local discomfort table (1.2)

the explanation of the table 1.2 see in the 1.6.2 Chapter. May we call the attention the Table 1.1a and 1.1b using the operative temperature

$$top = \frac{t_a . h_c + t_{mrt} . h_r}{h_c + h_r} \qquad (1.18)$$

Where:

t_a- Air temperature (C^o)

h_c – the convection heat transfer coefficient ($W / m^2 hr^o C$).see equation 1.13

t_{mrt} – the mean radiant temperature (C^o)

h_r – the radiant heat transfer coefficient ($W / m^2 hr^o C$). see equation 1.12

Table. 1.1a. Design criteria for spaces in different types of buildings. This table applies for low-polluting building materials and furnishing, for the occupancy listed in the tale and for a ventilation effectiveness of one.

Type of building/ space	Clothing (clo)		Activity	Occupancy	Category	Operative temperature °C		Mean air velocity m/s		Sound pressure	Required ventilation rate	Additional ventilation when smoking is allowed[2]
	Summer	Winter	met	person/ (m^2 floor)		Summer	Winter	Summer	Winter	dB(A)	l/s(m^2 floor)	l/s(m^2 floor)
Single office (cellular office)	0,5	1,0	1,2	0,1	A	24,5±0,5	22,0±1,0	0,18	0,15	30	2,0	-
					B	24,5±1,5	22,0±2,0	0,22	0,18	35	1,4	-
					C	24,5±2,5	22,0±3,0	0,25	0,21	40	0,8	-
Landscaped office	0,5	1,0	1,2	0,07	A	24,5±0,5	22,0±1,0	0,18	0,15	35	1,7	0,7
					B	24,5±1,5	22,0±2,0	0,22	0,18	40	1,2	0,5
					C	24,5±2,5	22,0±3,0	0,25	0,21	45	0,7	0,3
Conference room	0,5	1,0	1,2	0,5	A	24,5±0,5	22,0±1,0	0,18	0,15	30	6,0	5,0
					B	24,5±1,5	22,0±2,0	0,22	0,18	35	4,3	3,6
					C	24,5±2,5	22,0±3,0	0,25	0,21	40	2,4	2,0
Auditorium	0,5	1,0	1,2	1,5	A	24,5±0,5	22,0±1,0	0,18	0,15	30	16[1]	-
					B	24,5±1,5	22,0±2,0	0,22	0,18	33	11	-
					C	24,5±2,5	22,0±3,0	0,25	0,21	35	6,4	-
Cafeteria/ Restaurant	0,5	1,0	1,4	0,7	A	23,5±1,0	20,0±1,0	0,16	0,13	35	8	-
					B	23,5±2,0	20,0±2,5	0,20	0,16	45	5,7	5,0
					C	23,5±2,5	20,0±3,5	0,24	0,19	50	3,2	2,8

1) It may be difficult to meet the Category A draught criteria.
2) Additional ventilation required for comfort when 20% of the occupants are smokers. The health risk of passive smoking should be considered separately.

Table 1.1b. Design criteria for spaces in different types of buildings. This table applies for low-polluting building materials and furnishing, for the occupancy listed in the tale and for a ventilation effectiveness of one.

Type of building/ space	Clothing (clo)		Activity	Occupancy	Category	Operative temperature °C		Mean air velocity m/s		Sound pressure	Required ventilation rate	Additional ventilation when smoking is allowed
	Summer	Winter	met	person/ (m^2 floor)		Summer	Winter	Summer	Winter	dB(A)	l/s(m^2 floor)	l/s(m^2 floor)
Classroom	0,5	1,0	1,2	0,5	A	24,5±0,5	22,0±1,0	0,18	0,15	30	6,0	.
					B	24,5±1,5	22,0±2,0	0,22	0,18	35	4,3	.
					C	24,5±2,5	22,0±3,0	0,25	0,21	40	2,4	.
Kindergarten	0,5	1,0	1,4	0,5	A	23,5±1,0	20,0±1,0	0,16	0,13	30	7,0	.
					B	23,5±2,0	20,0±2,5	0,20	0,16	40	5,0	.
					C	23,5±2,5	20,0±3,5	0,24	0,19	45	2,8	.
Department store	0,5	1,0	1,6	0,15	A	23,0±1,0	19,0±1,5	0,16	0,13	40	3,5	.
					B	23,0±2,0	19,0±3,0	0,20	0,15	45	2,5	.
					C	23,0±3,0	19,0±4,0	0,23	0,18	50	1,4	.

1.6.2. Local thermal Discomfort [4]

Even though a person has a sensation of thermal neutrality, parts of the body may be exposed to conditions that result in thermal discomfort. This local thermal discomfort can not be removed by raising or lowering the temperature of the enclosure. It is necessary to remove the cause of the localised over-heating or cooling.

Generally, local thermal discomfort can be grouped under one of the following four headings:

1. Local convective cooling of the body caused by draught

2. Vertical air temperature differences.

3. Hot or cold feet, caused by uncomfortable floor temperature.

4. Cool or warm of parts of the body by asymmetrical radiation. This is known as a radiation asymmetry problem.

Remember, only when both the local and general thermal comfort parameters have been generated ated, can the quality of the thermal environment be judged.

The basic calculation table of this parameters see table. 1.2

Table 1.2 Three categories of thermal state and local discomfort factors

Category	Thermal state of the body as a whole		Local discomfort			
	Predicted Percentage of Dissatisfied PPD %	Predicted Mean Vote PMV	Percentage of dissatisfied due to draught DR %	Percentage of dissatisfied due to vertical air temperature difference %	Percentage of dis-satisfied due to warm or cool floor %	Percentage of dissatisfied due to radiant assymmetry %
A	< 6	-0,2<PMV< +0,2	< 15	< 3	< 10	< 5
B	< 10	-0,5<PMV< +0,5	< 20	< 5	< 10	< 5
C	< 15	-0,7<PMV< +0,7	< 25	< 10	< 15	< 10

Using the CR 1752 Standard the so called "optimum operative temperature " diagrams (fig. 1.2)

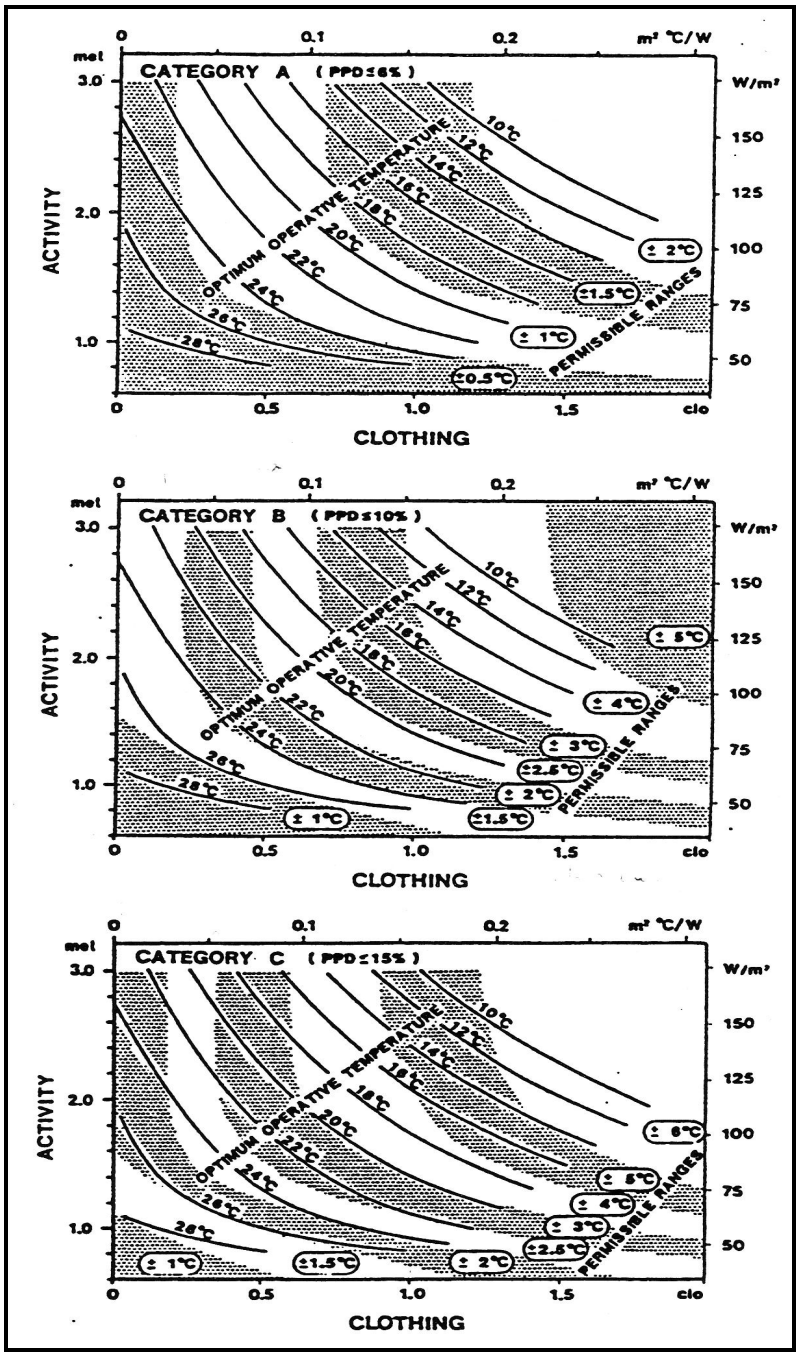


Fig. 1.2. The basic designing diagrams of (CR 1752)

Fig 1.2. The optimum operative temperature as a function of clothing and activity for the three categories of the thermal environment.

1.6.2.1. Draught criterion

The permissible mean air velocity and turbulence in the function of local air temperature is given in Fig 1.3 for the three categories. The mean air velocity is a function of local air temperature and turbulence intensity. The turbulence may vary between 30 and 60 % in conventionally ventilated spaces.

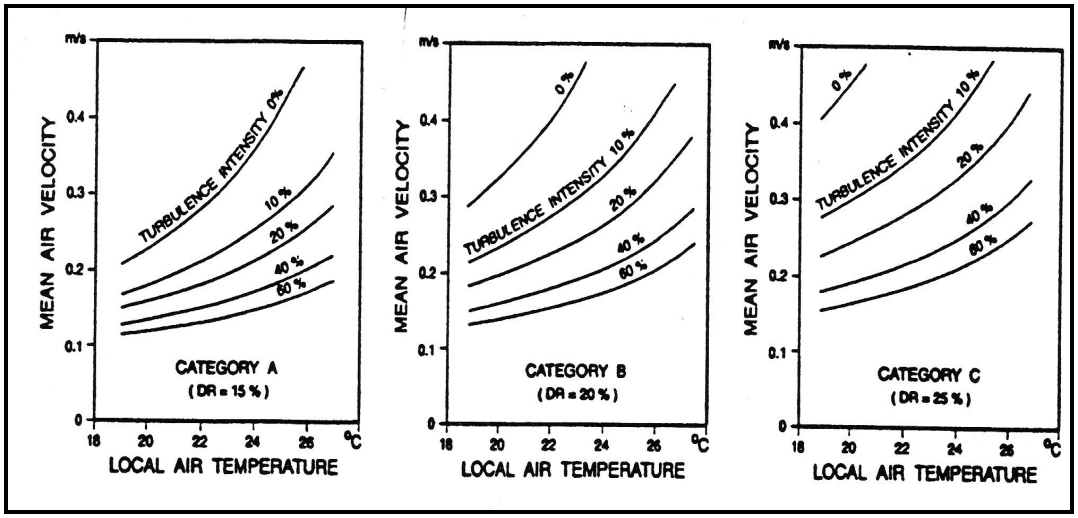


Fig 1.3. Permissible mean air velocity as a function of local air temperature and turbulence intensity for the three categories of the thermal environment.

1.6.2.2. Vertical air temperature

For the calculation of vertical air temperature, see Fig. 1.4

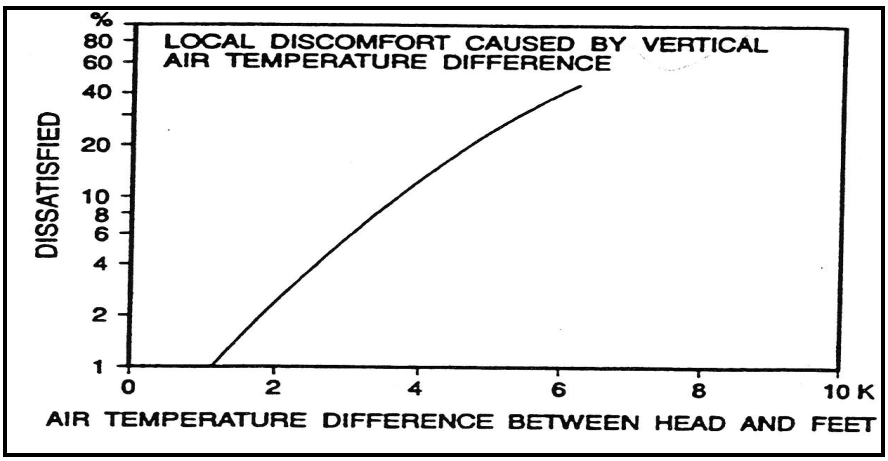


Fig 1.4. Local discomfort caused by vertical air temperature difference

Table 1.3. Permissible vertical air temperature difference.

Category	Vertical air temperature difference °C
A	< 2
B	< 3
C	< 4

1.6.2.3. Floor temperature

By the local discomfort of floor could take into consideration fig. 1.5

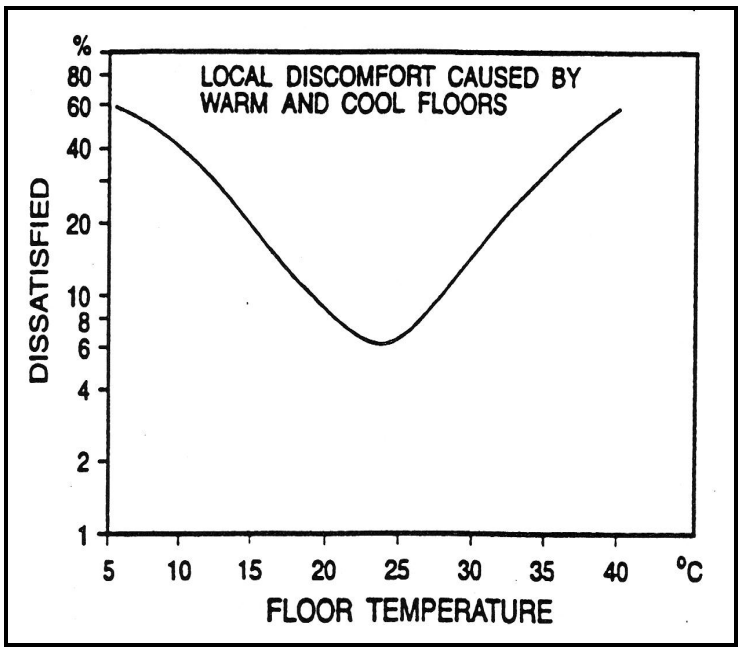


Fig 1.5. Local discomfort caused by warm and cool floors

Table 1.4. Permissible range of the floor temperature for the three categories of the thermal environment

Category	Range of surface temperature of the floor °C
A	19 - 29
B	19 - 29
C	17 - 31

1.6.2.4. Asymmetrical radiation

Effect of asymmetrical radiations see Fig. 1.6

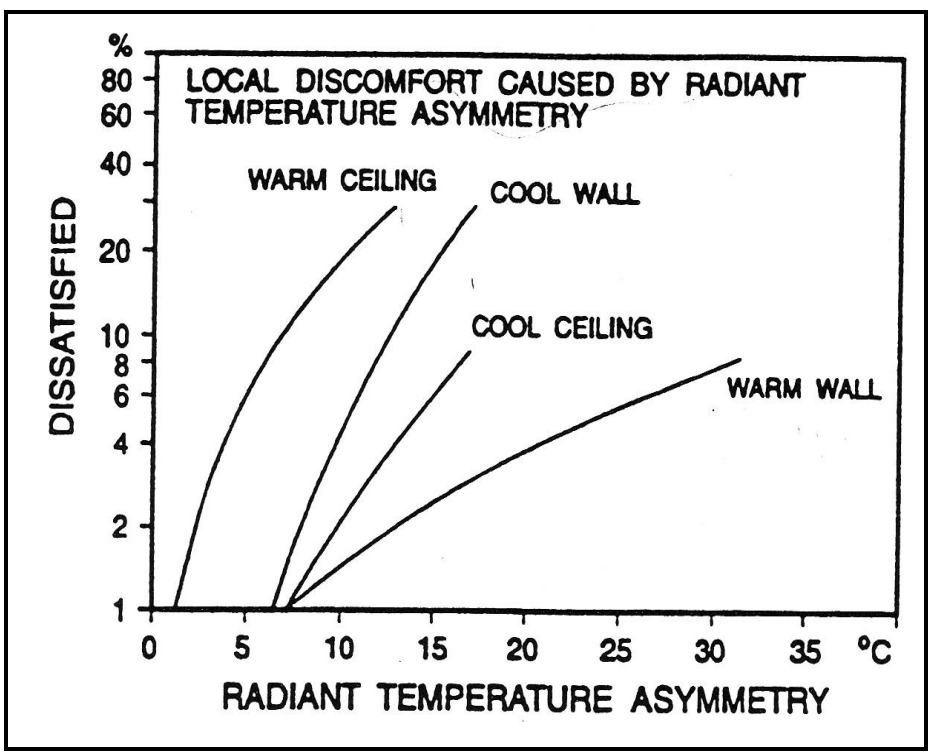


Fig 1.6. Local discomfort caused by radiant temperature asymmetry.

Table. 1.5. Permission radiant temperature asymmetry for the three categories of the thermal environment

Category	Radiant temperature asymmetry °C			
	Warm ceiling	Cool wall	Cool ceiling	Warm wall
A	< 5	< 10	< 14	< 23
B	< 5	< 10	< 14	< 23
C	< 7	< 13	< 18	< 35

1.6.3. Impact of airflow on human heat emission

The drught is an uneanred local cooling of the body caused by air movement and temperature. It is the most common cause for complaint in many ventilated spaces. A draught rating may be expressed as the percentage of people predicted to be othered by draught. The draught rating could calculate by the following equation (mode of draught) too.

$$DR = (34 - t_a)(v - 0.05)^{0.62}(0.37.v.Tu + 3.14) \quad (1.19)$$

DR - is the draught rating, the percentage of people dissatisfied due to draught

t_a - is the local air teperature, in degrees centigrade

v - is the local mean air velocity, in metres per second

Tu - is the local turbulence intensity, in per cent

The risk of draught is lower for people feeling warmer than neutral and higher for people feeling cooler than neutral for the whol body.

For people feeling warm for the whole body an increased air movement will normally be felt decreasing the warm discomfort (as calculated by the PMV .

CHAPTER 2

Ventilation and Asymmetrical Radiation

2.1. Airflow and air temperature distribution [5]

The office-type experimental room ventilated by a floor return-type under floor ventilation system to investigate the distributions of airflow velocity and air temperature. the fan-powered floor air unit (FAU) with rectangular supply and return air outlets covered by straight-profile linear bar-type air diffusers was installed to deliver the conditioned air in the experimental room. Turbulence intensity and draught rate distributions inside the room were also calculated by using the measured data. From the experimental results, it is found that undesirable high air velocities and high draught rates were created within a small region near the supply outlet of the FAU. Temperature differences between different height levels were maintained within an acceptable comfort level under the tested supply air conditions and heat loads. The results indicated that the temperature stratification could be maintained at an acceptable comfort level by designing the supply air conditions properly. A clearance zone is suggested as a design consideration for locating the FAUs and occupants to avoid undesirable draught discomfort to the occupants. Air distribution plays an important role in achieving a satisfactory level of air quality and thermal comfort in mechanically ventilated rooms. The airflow velocity, temperature, and turbulence intensity distribution inside the room are mainly affected by the characteristics of the air distribution system. These are the key

parameters that affect the mixing of the supply air to the entrained room air and the efficiency of aged air replacement. These parameters also determine the human's thermal comfort sensation according to various comfort models that have been developed.

Traditionally, centralized HVAC systems are designed based on the fully mixed principle where locating both the supply and return air vents at the ceiling level is a common configuration, especially in office-type environments. However, in modern office environments, it is difficult for this traditional HVAC system design to satisfy the increasing demand for individual workers preferences on their microclimate. In addition, with the increased use of these office equipments, installation and management of the huge amount of cables that come with these equipments have been a challenging issue in modern building designs. In many modern office buildings, raised floors were built to create floor voids for flexible and convenient cable management. The popularity of utilizing raised floors inside the office buildings to tackle this problem has been increasing in the past decades. With the installation of the raised floor, under floor ventilation systems, which make use of the floor void to distribute conditioned air, has been gaining concern among the industry and the number of

applications have been growing. It is because it provides various advantages and convenience during the construction phase and in the later maintenance works. For under floor ventilation systems performed better than the traditional ceiling-based systems both in the energy aspect and in the indoor air quality aspect. Much research has been conducted to further explore the characteristics and advantages of the under floor ventilation system.

Under floor ventilation, systems are found in two major configurations, the top return (TR) type and the floor return (FR) type. In recent years, extensive studies have been conducted on the TR-type system, also known as floor-supply displacement ventilation system. A major reason for the under floor ventilation system to achieve energy saving could be the higher supply air temperature associated with the under floor ventilation system. That the typical supply air temperature of the traditional overhead-type system is

13–14°C while the under floor system can be as high as between 14° and 17°C. Increasing the supply air temperature leaded to the increase in chillers coefficient of performance (COP). They reported that with the implementation of the floor-based ventilation system, the supply air temperature could be kept at approximately 4°C higher than that in the ceiling-based system while a comfortable environment in the occupied zone was maintained. This increase in supply air temperature allowed the chilled water supply temperature to be increased from 5°C to about 9–10°C, leading to an improvement of the chiller's COP from 3.6 to 5. During their experiment period the monthly average outdoor air temperature was 28.9°C and the monthly average relative humidity was 68% in Tokyo.

2.2. Impact of asymmetrical radiation and air movement [4]

Thermal comfort is determined by air temperature, relative humidity, air movement, mean radiant temperature, the presence of direct solar radiation (insulation), and occupants' clothing and activity levels. Windows affect human comfort in several ways. During cold periods, exterior temperatures drive interior glass surface temperatures down below the room air temperature; how low the glass temperature drops depends on the window's insulating quality. The closer they are to a window, the more they will feel its influence. The fact that this heat loss occurs on one side of the body more than the other is called radiant asymmetry, and this leads to further discomfort. A familiar example of radiant asymmetry is the experience of sitting around a campfire on a winter night. The side of the body facing the fire is hot, while the side facing away is cold. In the case of a cold window, a person may be cold in warm clothes in a 70 degrees Fahrenheit room air temperature if part of the body is losing heat to a cold window.

Drafts near windows are the second major source of winter discomfort. Many people mistakenly attribute drafts to leaky windows when in fact they are the result of cold air patterns initiated by cold window surfaces. Air next to the window is cooled and drops to the floor. It is then replaced by warmer air from the ceiling, which in turn is cooled. This

sets up an air movement pattern that feels drafty and accelerates heat loss (Fig 2.1). Cold-temperature-induced drafts occur at the same time as radiant discomfort. This emphasizes the need for insulating windows that maximize interior glass surface temperatures under cold environmental conditions.

Drafts can also be caused by windows with significant air leakage. These leaks can be a result of poor installation and/or ineffective weather stripping. Such drafts correlate directly to air infiltration levels (see energy-related issues in this chapter). Radiant heat loss, convective currents from cold window surfaces, and drafts from air infiltration leaks all cause people to turn up thermostats during cold periods. Because this action may have little effect on increasing comfort levels, it can be wasteful and costly.

Direct sun has fairly obvious impacts on thermal comfort as well. During cold periods, solar radiation (within limits) can be a pleasant sensation. During warm weather, however, it invariably causes discomfort (Fig 2.2). People often close shades or blinds to block sunlight even though this means they can no longer enjoy the view. Just as people turn up the heat to compensate for cold windows in winter, they may use air-conditioning to counter the effects of warm window surfaces and sunlight in summer. If air conditioners are not sized or installed properly, some areas of a room may become comfortable while others are not, causing significant waste of energy.

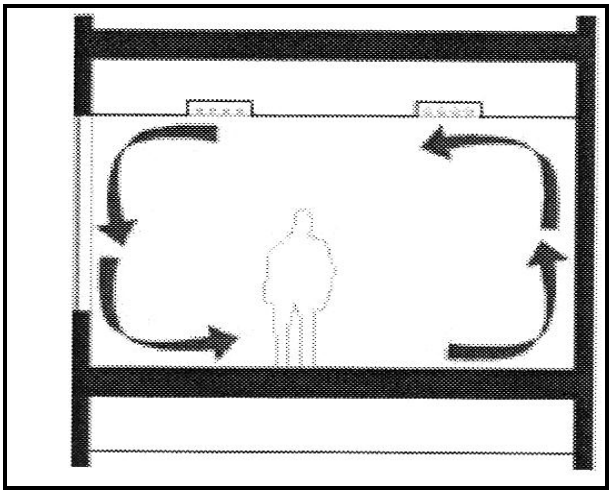


Fig 2.1. Air movement pattern that feels drafty and accelerates heat loss

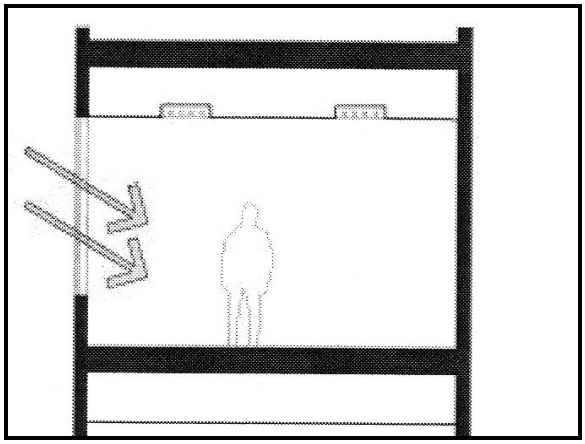


Fig. 2.2. During warm weather, causes discomfort

The glazing surface temperature increase due to solar radiation depends on the absorption of the glass and environmental conditions. Typical clear glass windows do not absorb enough solar radiation to cause a significant difference in surface temperature. With tinted glass, surface temperature increases can be significant. While poorly insulated tinted glass may actually feel quite comfortable on a cold sunny day, this practice is not recommended-the comfort consequences on hot summer days can be disastrous. During warm periods, the interior surface temperatures of poorly insulated tinted glass and clear glass with tinted film can get hot, as high as 140 degrees Fahrenheit. These surfaces radiate heat to building occupants and can also create uncomfortable convection currents of warm air.

In this analysis, it is assumed that the two major effects of windows on local thermal discomfort are asymmetric radiation from hot and cold window surfaces and from direct sunlight. Hourly values of predicted percentage dissatisfied (PPD) were computed based on the mean radiant temperature given these two effects, room air temperature, humidity, air speed, and ASHRAE Standard 55-92 [3]. Values for clothing and activity levels. This definition of PPD is more stringent than that defined for ASHRAE 55-92 [3] because of the inclusion of direct sun effects on local discomfort. Nevertheless, it is assumed in this book that the PPD must be less than 20 percent to comply with the Standard.

2.3. About the Asymmetry of Thermal Radiation [4]

If you stand in front of a blazing bonfire on a cold day, after a period of time your back will begin to feel uncomfortably cold. This discomfort can not be remedied by moving closer to the fire, resulting in an increased body temperature. This is an example of how non-uniform thermal radiation can result in the body feeling uncomfortable. To describe this non uniformity in the thermal radiation field, the

parameter Radiant Temperature Asymmetry is used. This parameter is defined as the difference between the Plane Radiant Temperature of the two opposite sides of a small plane element.

Experiments exposing people to changing degrees of radiant temperature asymmetry have proved that warm ceilings and cold windows cause the greatest discomfort, while cold ceilings and warm walls cause the least discomfort. During these experiments all the other surfaces in the room and the air were kept at an equal temperature.

The parameter Radiant Temperature Asymmetry can be obtained in two ways. One, by measuring t_{pr} in two opposite directions using a transducer that integrates the incoming radiation on to a small plane element from the hemisphere about it. The other is, to measure the temperatures of all the surrounding surfaces and then calculate Radiant Temperature Asymmetry. In fig. (2.3) the procedure to be used for such a calculation can be seen.

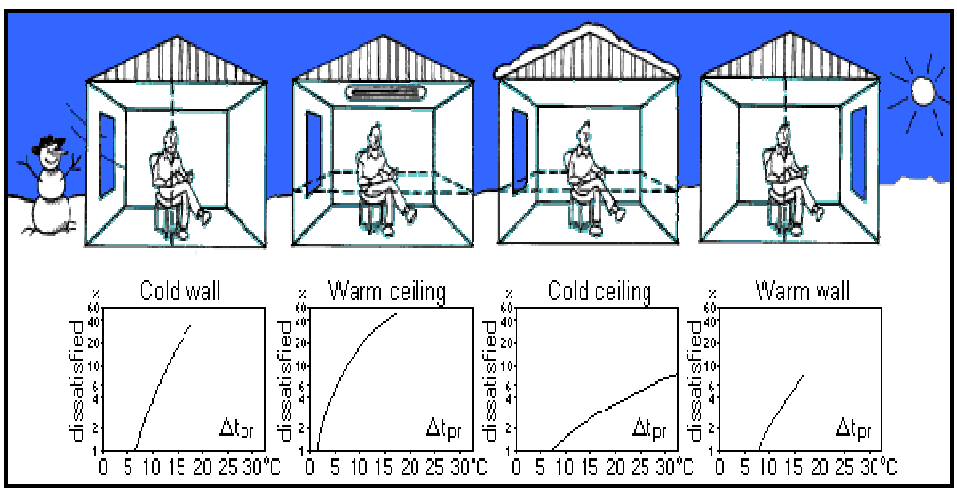


Fig. 2.3. Procedure to be used for such a calculation

2.4. Vertical Air Temperature Difference [4]

Generally it is unpleasant to be warm around the head whilst at the same time being cold around the feet, regardless of this being caused by radiation or convection. In the last section we looked at the acceptance limits of Radiant Temperature Asymmetry. Here we will look at what air temperature difference is acceptable between the head and feet.

Experiments were carried out with people in a state of thermal neutrality. The results, displayed in the diagram, showed that a 3°C air temperature difference between head and feet gave a 5% dissatisfaction level. The 3°C have been chosen as the ISO 7730 [7] acceptance level for a sitting person at sedentary activity.

When measuring air temperature differences it is important to use a transducer which is shielded against thermal radiation. This ensures that the air temperature is measured and not an undefined combination of air and radiant temperature.
The Vertical Air Temperature difference is expressed as the difference between the Air Temperature at ankle level and the Air Temperature at neck level.

2.5. Measuring Mean Radiant Temperature [4]

This consists of a thin-walled copper sphere painted black containing a thermometer with its bulb at the center of the sphere (typically of diameter 150 mm). The globe thermometer is suspended and allowed to reach thermal equilibrium with its surroundings (usually 20 minutes). With a far-inside globe, equilibrium time is 6 minutes, and using a thermocouple instead of a mercury thermometer, the time is 10 minutes. The equilibrium temperature depends on both convection and radiation transfer, however by effectively increasing the size of the thermometer bulb the convection transfer coefficient is reduced and the effect of radiation is proportionally increased. In equilibrium the net heat exchange is zero. Because of local convective air currents the globe temperature (tg) typically lies between the air temperature (ta) and the true mean radiant temperature (tr). The faster the air moves over the globe thermometer the closer tg approaches t_a.

The Mean Radiant Temperature of an environment is defined as that uniform temperature of an imaginary black enclosure which would result in the same heat loss by radiation from the person as the actual enclosure.
The equation for the calculation of Mean Radiant Temperature is eq (2.1)

$$\bar{t}_r = \sqrt[4]{\sum_n F_{p-i}\left(t_i + 273\right)^4} - 273 \qquad (2.1)$$

t_i Surface temperature of surface i [°C]

F_{p-i} Angle factor between the person and surface i $\sum F_{p-i} = 1$

Measuring the temperature of all surfaces in the room is very time consuming, and even more time consuming is the calculation of the corresponding angle factors. That is why the use of the Mean Radiant Temperature is avoided if possible. The Globe Temperature, the Air Temperature and the Air Velocity at a point can be used as input for a Mean Radiant Temperature calculation. Use of the Globe Temperature for calculation of Mean

Radiant Temperature and a procedure for calculation of Mean Radiant Temperature on the basis of Plane Radiant Temperatures can be seen in fig. (2.4).

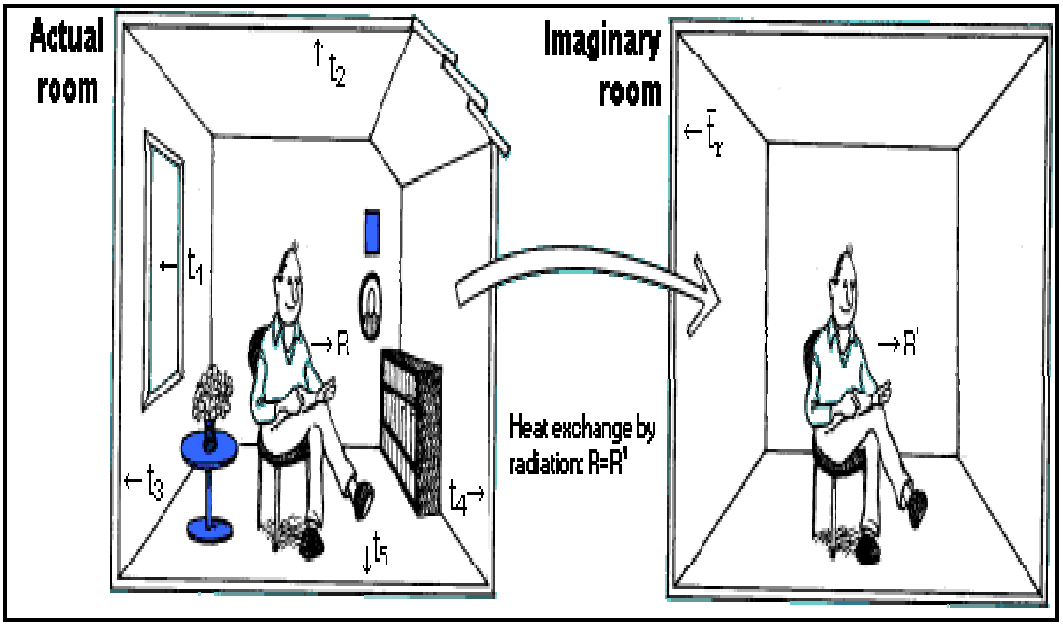


Fig. 2.4. Mean Radiant Temperature on the basis of Plane Radiant Temperatures

2.6. Measuring surface temperature (thermal conductivity) [4]

All surfaces are made of materials which conduct heat at varying rates (thermal conductivity). Our thermal sensations are not good indicators of surface temperature but rather we sense the rate of heat loss or gain e.g. in a thermally stable setting a tile floor will feel colder than a carpeted floor even though they have the same surface temperature because tile has a higher thermal conductivity than carpet. Surface temperatures can be measured by thermometers placed in direct contact with the surface of interest. Surfaces can be a significant source of discomfort.

2.6.1. Humidity [5]

The absolute humidity refers to the dampness/wetness in the air in the form of water vapor, that is, the mass of water vapor present in a unit volume of air (moisture content).

In S.I. units it is expressed in grams of water per cubic metre of air or space. (nb 454 grams = 1 lb/ 1m3 = 1.308 yd3 = .027 oz/yd3

CHAPTER 3

The purpose of the study

3.1. Situation of thermal comfort due regards in hot environment of Libya

In the hot environment countries as Libya the climate is mild, but the needs to heating in winter and cooling in summer are certain, especially in coast areas where the humidity is high and hot the climate. The traditional buildings provided a relatively acceptable thermal comfort in winter and particularly in summer. These buildings are naturally ventilated and have a high thermal capacity. They are equipped with a courtyard; in those houses, adapted shadowing devices and appropriate orientations of the openings facilitate the control of the solar gains. During summer, when the outdoor air is warm, people enjoy opening the windows and doors, the building is then cooled by natural ventilation and coolness is stored at night in the building thermal mass for the next day. This traditional way of construction is not anymore possible because of its cost and the change of people standard of life and behaviour; courtyard houses are replaced with flat or terrace houses. The development of "modern" construction was very fast and no care was paid to the thermal quality of the buildings. Most of the recent buildings are not equipped with thermal insulation and their tightness to air very poor; this is due to the lack of appropriate standards. While the traditional buildings provide an acceptable thermal comfort, the recent buildings are very poor from the thermal point of view, and the thermal comfort is very low.

3.1.1. General problems

- The use of thermal insulation, multiple glazing and controlled ventilation systems are necessary in the North African countries, the climate is mild, but the needs to heating in winter and cooling in summer are certain, especially in coast areas where the humidity is high.
- During summer, when the outdoor air is cool, people enjoy opening the windows and doors, the building is then cooled by natural ventilation and coolness is stored in the building thermal mass for the next day. This traditional way of construction is not anymore possible because of its cost and the change of people standard of life and behaviour; courtyard houses are replaced with flat or terrace houses.
- The development of "modern" construction was very fast and no care was paid to the thermal quality of the buildings. Most of the recent buildings are not equipped with thermal insulation and their tightness to air is very poor., this is due to the lack of appropriate standards. While the

traditional buildings provide an acceptable thermal comfort, the recent buildings are very poor from the thermal point of view, and the thermal comfort is very low.

- During the last few years, the standard of life has risen, so people started to equip their houses and offices with heating systems in winter and with small air conditioning systems (mainly windows and split air conditioning systems) for the summer conditions. Therefore the energy consumption became very high because of the mediocre thermal quality of the buildings.
- In Libya, even if the buildings are heated and cooled, they are thermally controlled according to people sensation.
- The heating or air conditioning systems, if they exist, are put on or off manually when people feel uncomfortable.
- There is no set temperature for heating or cooling, and so the indoor temperatures are free running. Appropriate thermal standards for the North African countries, taking in consideration the particularities of the climate, the people culture and behaviour, and the method of occupation of the buildings, becomes necessary. Such standard is now under preparation. In Fig 3.1, show Monthly maximum temperature of Tripoli.

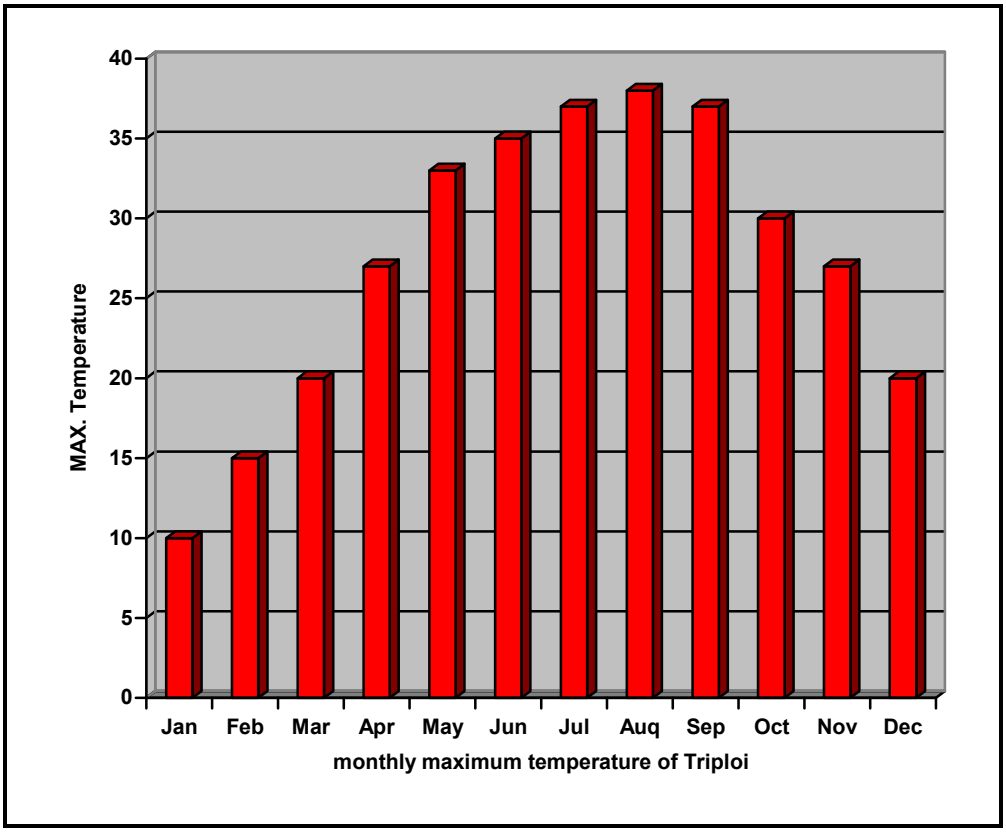


Fig 3.1. Monthly maximum temperature of Libya

3.2. Improving thermal comfort by developed layers of the buildings.

A wealth of information and data with respect to thermal properties and analyses are offered in previous studies and presented in various literatures. who assessed insulation materials on the basis of time lag, decrement factor, cost, and R-value by Al-Nafeez et al [6]., Zaki et al [7]. predicted thermal performance of a two-layer composite wall with periodic change of the outside air temperature and solar insulation. Zaki et al [7] .Eben Saleh, M.A. [8]. evaluated the performance of different arrangements and thicknesses of building insulation within the outer side of the building envelope. Eben Saleh, Vol-7 [9].

The heat conduction equation to the frequency domain, Lindfors et al., had formulated a technique for estimating thermal properties of building components from in situ measurements, Lindfors et al.,[10]. Later, a socalled state-space model was derived by Norlen for estimating the thermal properties that were assumed to be constant. However, this model could be applied only to a single homogeneous slab with negligible radiative and convective fluxes, Norlen, [11]. Keeping constant the overall thickness and thermal transmittance of a three layered building envelope, , Bojic et al., [12]. Using a whole building dynamic modeling performed in a continuously utilized house, Kossecka et al.[13]. proved that walls with massive internal layers have better annual thermal performance than those with inside insulation. Kossecka et al. [13]. Al-Sanea et al, 2002 [14]. had also shown that the wall orientation has a significant effect on the heat transfer characteristics, but a relatively smaller effect on the total cost and optimum thickness for a given insulation material. In the same literature, they referred to the total cost and optimum insulation thickness and their sensitivity to changes in the economic parameters.. Al-Sanea ea al,vol 24 [15] and Al-Sanea ea al, vol 22 [16]. investigated using a finite volume implicit method, the effect of the location of the insulation layer in building walls on the initial transient heat-transfer response. They studied the effect of insulation location on the thermal performance of building walls under steady periodic conditions.

Using the climatic data of Riyadh, the results showed that the insulation layer location had a minimal effect on the daily mean heat transmission load, with a slight advantage for the outside insulation in summer and the inside insulation in winter. The outside insulation gave smaller amplitude of load fluctuation and smaller peak loads in both summer and winter for all wall orientations. Al-Sanea ea al, vol 22 [16].

Throughout this preview, it could be noted that an extensive research work has been recorded into the influence of wall and roof design and insulation to energy behavior. However, the corresponding

performance and influences of other factors such as building shape and area, U-values, air volumes, and temperature differentials are not much considered in any of these studies. Correspondingly, an optimized overall thermal design of buildings to promote a combined design performance prediction is the issue to be particularly stressed in this study.

Almost of Libya buildings with out insulation, therefore the buildings are not comfortable There are three main variables, which affect the rate of heat loss or gain from a building: 1- total area and U value; 2- the volume of air in the building; and 3- temperature difference. ASHRAE, [17]. Recognizing those variables will help the designer to understand the influence of the overall building component with respect to construction form. A designer of a building can simply affect the first variable by making changes in the choice of materials and the form of construction. These changes can significantly affect both the U-value and the method of construction.

However, a designer is able to manipulate and emphasize those variables altogether. For any enclosed volume, there are many ways in which dimension of height, length and width can vary. Resulting in different total surface area without changing the volume of the enclosure. Thus, in case of two buildings, both having the same volume and built of the same material, quite different surface area may be encountered, and hence, different rates of fabric heat loss will develop. ASHRAE, [17].

3.3. Thermal comfort model of ASHRAE

We were working about the European CR1752 [3], but we have to mentioned the ASHRAE ,55 [1] thermal comfort model briefly. The usual thermal comfort standards, ASHRAE,55, [1], during these experiments, the person posture, his activity and his clothing are fixed, thus, the person sensation is taken under steady state conditions. In real conditions, these parameters are never fixed. Moreover, a person changes one or some of these parameters to adapt himself to his climatic conditions and to seasons. In Libya, three different climatic zones have been identified in the country. There are wide seasonal changes in the weather between summer and winter. In summer, midday temperature can exceed 42 °C in some regions in the south. In winter, temperature can fall down to around 5 °C in the Mountains. In the desert, at the country south, the freezing winter nights are also very common. The indoor design temperatures as described by international standards . ISO 7730 [18] and ASHRAE 55, [1] take no account of climatic variations and people adaptive behaviors. For any task and use of the building, there is a recommended temperature that is assumed to apply irrespective of climate, way of life and kind of clothing, though with some recognition of difference between summer and winter. Analysis of thermal comfort field studies have shown that indoor comfort temperature as felt by the occupants is function of mean outdoor temperature M.A. Humphreys [19], A. Auliciem and R. de Dear

[20], J.F. Nicol, A. Auliciem [21], J.F. Nicol, I.A. Raja [22], J.F. Nicol and S. Roaf [23], R. de Dear, A. Auliciem [24], R.J. de Dear and G.S. Brager [25] and G.S. Brager and R.J. de Dear [26]. This means that we can relate indoor comfort temperature to climate, region and seasons. For free running buildings and according to different surveys held under different climatic conditions, M.A. Humphreys, [27] has found that the comfort temperature can be obtained from the mean outdoor temperature with eq. (3.1)

$$T_c = 0.534T_o + 11.9 \tag{3.1}$$

Humphreys equation by deleting some fields studies such those with children as the subjects, and adding more information from other studies not included by M.A.Humphreys, UK [28]. These revisions increased the database to 53 separate field studies in various climatic zones covering more countries and more climates. After combining the data for naturally ventilated buildings and air-conditioned buildings, the analysis led to an equation involving the outdoors air temperature (T_o) and the indoor air temperature (T_i), this resulting equation is eq. (3.2):

$$T_c = 0.48T_i + 0.14\ T_o + 9.22 \tag{3.2}$$

A. Auliciems has also proposed a single line for all buildings, which covered the naturally ventilated buildings and air-conditioned buildings [29]. This relation is given by eq. (3.3)

$$T_c = 0.31\ T_o + 17.6 \tag{3.3}$$

J.F. Nicol has conducted several surveys under different climatic conditions. In a first survey in Pakistan [30], he has established a relation between comfort temperature and outdoor temperature given by eq. (3.4)

$$T_c = 0.38\ T_o + 17.0 \tag{3.4}$$

In a second survey in Pakistan [31], he has found a second regression given by eq. (3.5)

$$T_c = 0.36\ T_o + 18.5 \tag{3.5}$$

Those relations show clearly that the comfort temperature is related to the outdoor temperature and so to the climate. The difference between those relations confirms that there is no universal comfort temperature.

CAPTER 4

Field studies

4.1. Thermal comfort field study with subject, Libya

This field study of thermal comfort within two types of buildings; old (traditional) and new (contemporary), in Ghadames city, Libya. The survey was undertaken in the summer seasons 2005 and 2006, which were typical of the hot–dry climate of North Africa. It shows how the 237 residents responded to the environmental conditions. Questionnaires were collected from the residents of 51 buildings: 24 old buildings that employ natural ventilation systems with courtyards and 27 new buildings that employ air-conditioning systems. In addition the environmental parameters were measured in 11 buildings (5 old, 6 new) representing 50 subjects, to calculate the predicted mean vote value of the subject using Fanger's model as presented in ISO 7730 [18]. The survey has shown that the measurements of predicted mean vote (PMV) and Predicted Percentage of Dissatisfied (PPD) in new air-conditioned buildings provide satisfactory comfort conditions according to ISO 7730 [18], and the occupants agree by indicating a satisfactory actual mean vote (AMV). The equivalent measurements and survey results in old traditional buildings indicated that although the PMV and PPD based on measurements and ISO 7730 [18], implied discomfort (hot), the occupants expressed their thermal satisfaction with the indoor comfort conditions. The field study also investigated occupants' overall impression of the indoor thermal environments; the results suggest that people have an overall impression of higher standard of thermal comfort in old buildings than in new buildings.

This field survey seeks to determine the extent to which existing research findings, and the ISO 7730 [18], which is based on the Fanger model [2], could be applied when designing for thermal comfort in a hot–dry climate. Since 1987, UNESCO has listed the old town of Ghadames city as a historic site in the World Heritage List. As such this

makes Ghadames an important area and to be chosen as a case study in shaping the values of traditional dwellings in Libyan towns and cities. Ghadames lies 630 km south-west of Tripoli, close to the border

of Algeria and Tunisia (see Fig. 4.1.1). It is located in the Libyan Sahara Desert, and situated at an altitude of 350 m above sea level.

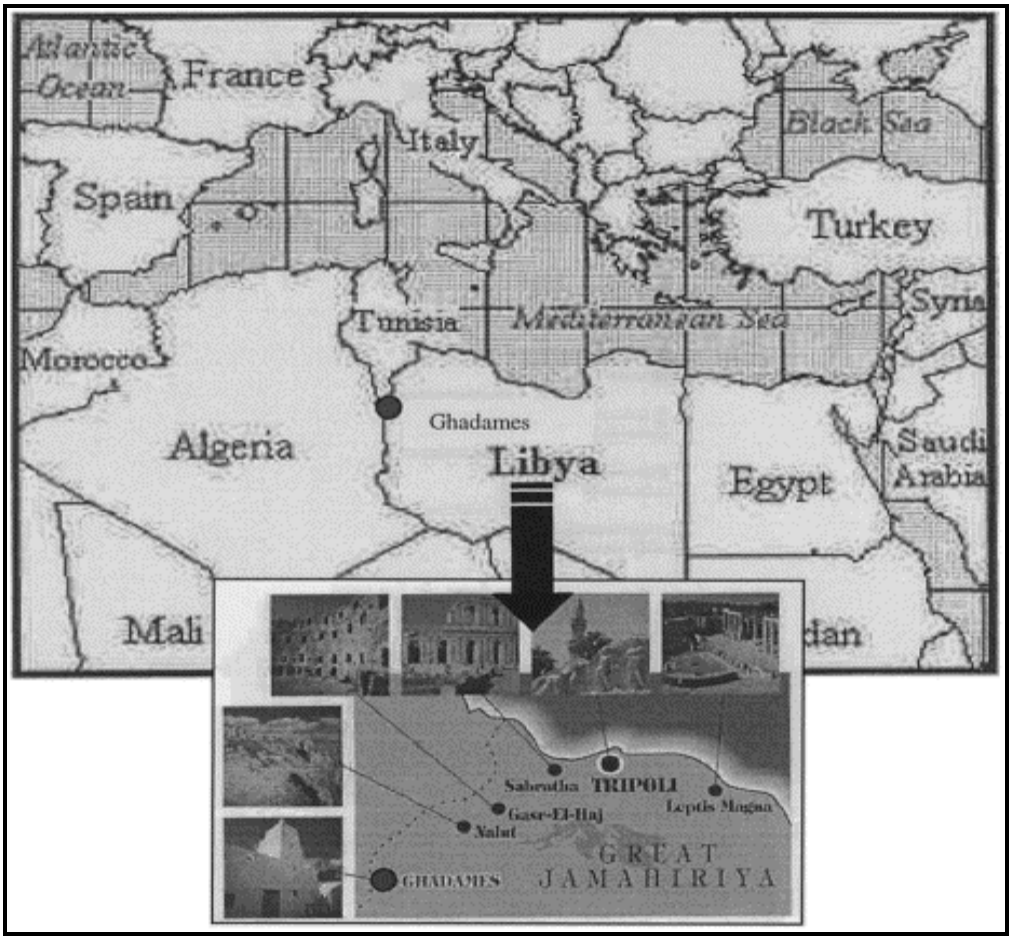


Fig. 4.1.1 Location maps.

The population by 1990 had increased to 14,700 as a result of the improved economic condition of the country. The Ghadames soil is clay and stone, which is not suitable for agricultural use, but good use has been made of existing local materials, such as palm trees, clay and stones (i.e. limestone, gypsum, etc.) to construct the traditional buildings. The old town was originally protected from the drifting sand and from the high air temperature of the surrounding desert by date palm trees. The desert climate of Ghadames is one of the most extreme climatic conditions within the whole country. The climate is characterised by high air temperature, high solar radiation, low rainfall, low humidity and many sandstorms. The weather in summer is hot throughout the day, the air temperature sometimes rising to more than 47°C, falling to 30°C during the nights. In winter, the weather is cold at night, falling to 0°C

temperature in the night. The old town of Ghadames provides a unique opportunity to investigate the differences between traditional and contemporary architecture.

4.1.1. Parameters of construction

The construction in Libya and Europe and that radiant asymmetry may also cause discomfort. The percentage of dissatisfied as a function of the radiant temperature asymmetry caused by a warm ceiling, a cool wall, a cool ceiling or by a warm wall. Radiant asymmetry is rarely a problem in ventilated or air-conditioned building construction and in figures 4.1.1 and 4.1.2 shown the heat transfer and asymmetrical radiant through the surfaces of building and for improving thermal comfort the first step improves the building construction. We show two types of constructions and outdoor-indoor temperature:

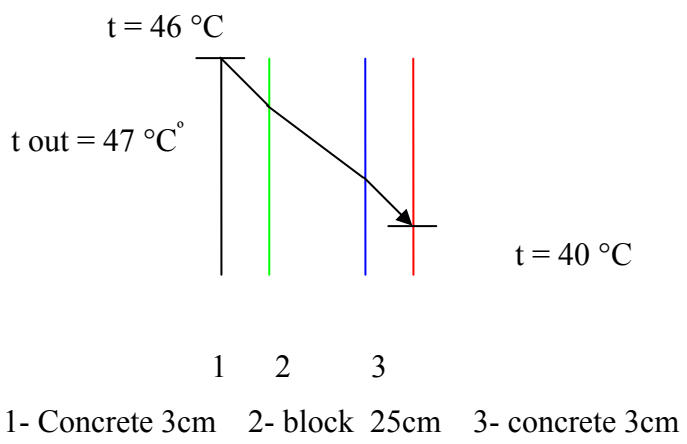
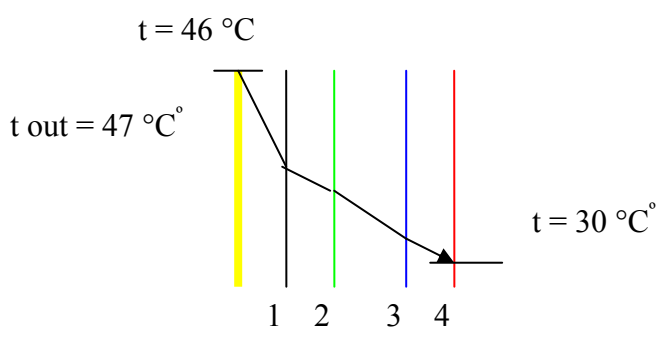


Fig 4.1.2 Without isolation



2- concrete 3cm 3-block 25cm 4-concrete 3cm

Fig 4.1.3 With isolation

4.1.2. Overview of thermal comfort

Thermal comfort has been defined by ASHRAE [1] as "that condition of mind which expresses satisfaction with the thermal environment" and as such will be influenced by personal differences in mood, culture and other individual, organisational and social

factors. Thus thermal discomfort within building environments is a prevalent and significant issue throughout the developed and developing countries. The thermal comfort standards prescribed by ISO 7730 [18] are the first that have been used on a world-wide basis. They are based on Fanger's work in climate chamber experiments on young Danish students and to the PMV model. There have been extensive studies to measure thermal comfort using test chambers; e.g. Fanger [2] I and S. Tanabe et al [32] in Japan, T.M. Chung and Tong [33] in Hong Kong. Other studies in thermal comfort have used field surveys such as A. Auliciems and De Dear [34], J.F. Nicol [35] in Pakistan, and T.H. Karyono [36]. In addition, D.A. McIntyre [37] presented a comparison of Fanger's climate chamber work with field studies reviewed by Humphreys [38], suspecting that certain intervening variables that occur in the "real" world might not be reproducible in the climatic chamber. N.A. Oseland [39] has reported on significant discrepancies occurring between predicted mean votes (PMV) and actual mean votes (AMV) values obtained in offices and homes as compared with climate chamber studies, attributing the differences to contextual and adaptation effects. Thus, the field studies closer to the "real" world may be preferable to climate chamber ones. A field study on thermal comfort of the above type has not previously been attempted in Libya specifically or North Africa generally. The present paper discusses the practical application of ISO 7730 [18], Standard based on field work carried out at Ghadames city in Libya. It also presents human thermal sensation votes in both naturally ventilated and air-conditioned buildings.

4.1.3. Detailed overview of field survey

A thermal comfort field survey was carried out in 51 buildings in the Ghadames, Libya in summer 2006. These buildings employ natural ventilation systems (old traditional buildings), and air conditioning systems (new contemporary buildings). Fig. 4.1.4 and Fig. 4.1.5 show typical examples of these two types. It was impossible to cover all the buildings in Ghadames, due to the limited time, but the survey was planned to select

buildings that represent: (a) different locations in Ghadames; and (b) typical types and sizes (i.e. private and public, one story building, flats or two story building, etc.). All 237 responding subjects (male and female), who lived in the buildings surveyed, were Ghadamesian tribe's people in age from 19 to 70 years, and with an average occupancy of 4.5 persons for an old building and 4.8 persons for a new building. The work was divided into two kinds of surveys: (i) subjective survey, which involved collecting data from a total of 24 old buildings, effectively representing 108 (24×4.5) subjects and 27 new buildings representing 129 (27×4.8) subjects, in order to represent the actual mean vote (AMV) of the resident in Ghadames. (ii) Subjective and objective survey, which involved measuring the environmental variables in 11 buildings, representing 22 (5×4.5) subjects in old buildings and 28 (6×4.8) subjects in new buildings, in order to calculate the PMV of the subjects using Fanger's model as presented in ISO 7730 [18].

Fig. 4.1.4. Old type of buildings in Ghadames, 2005.

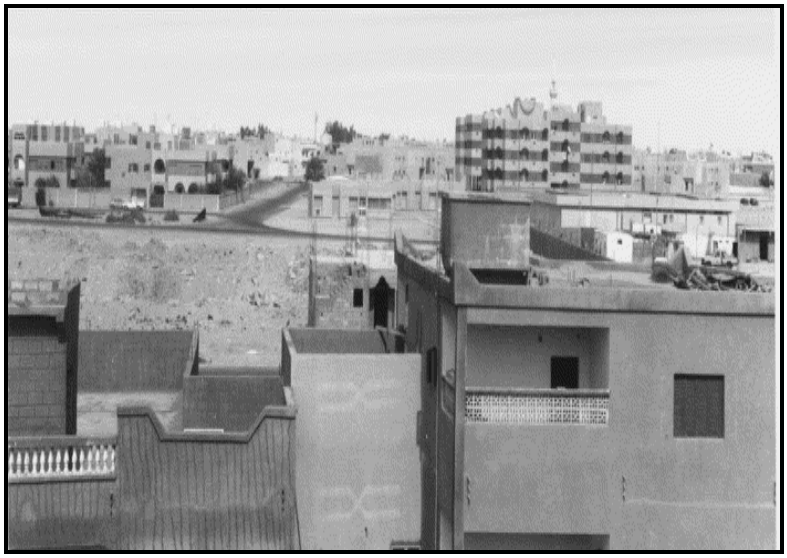


Fig. 4.1.5. New type of buildings in Ghadames, 2005.

4.1.4. The equipment

In order to calculate the PMV and PPD values as presented in the ISO 7730 [18], four basic environmental parameters were measured, and two personal parameters were estimated. Air temperatures were recorded using radiation-shielded thermocouples (Type T, copper/constantan). These values were logged every 15 min and average values were calculated every hour. Air velocities were measured using an omni-directional anemometer. The mean globe temperatures were measured, using a standard globe thermometer and mean radiant temperatures were then calculated. The equipment used in this study complied with the criteria given in ISO 7726 [40].

4.1.5. The questionnaire

Subjects were asked to complete a questionnaire at the same time as the environmental variables (air temperature, globe temperature, surface temperature, air velocity and relative humidity) were being recorded. Details of clothing and activities were noted for each subject. The subjective study involved collecting data using questionnaires. The questionnaire is based on six sections: background and personal information; social interaction; thermal environment and personal influences; occupants' perceptions of the environmental conditions in the whole building; occupants' thermal comfort; and people's general feeling and personal well being. For the purpose of this field study, only the personal information and occupants' thermal sensation data have been presented including data about the age and gender of the subjects and his/her family; hours/days spent inside the building including sleeping

time; presence of air-conditioning units; etc. The occupants' thermal comfort has been tested using 7-point ASHRAE sensation scale, ranges from −3 as cold to +3 as hot and 0 as neutral. In addition, preference and satisfaction scales have been used. The subjects were selected randomly from different groups of people in Ghadames (i.e. Educated, Administrative, Architects and Elite groups) to represent typical range of samples.

4.2. Findings

4.2.1. Subjective votes

Questionnaires were collected from 51 buildings representing 237 subjects, from both old and new buildings, as mentioned earlier. Fig.4.2.1 shows the overall thermal comfort sensation for a summer season of the respondents in old naturally ventilated buildings and new air-conditioned buildings. These results convey the occupants' overall impressions of the building types in terms of their comfort conditions.

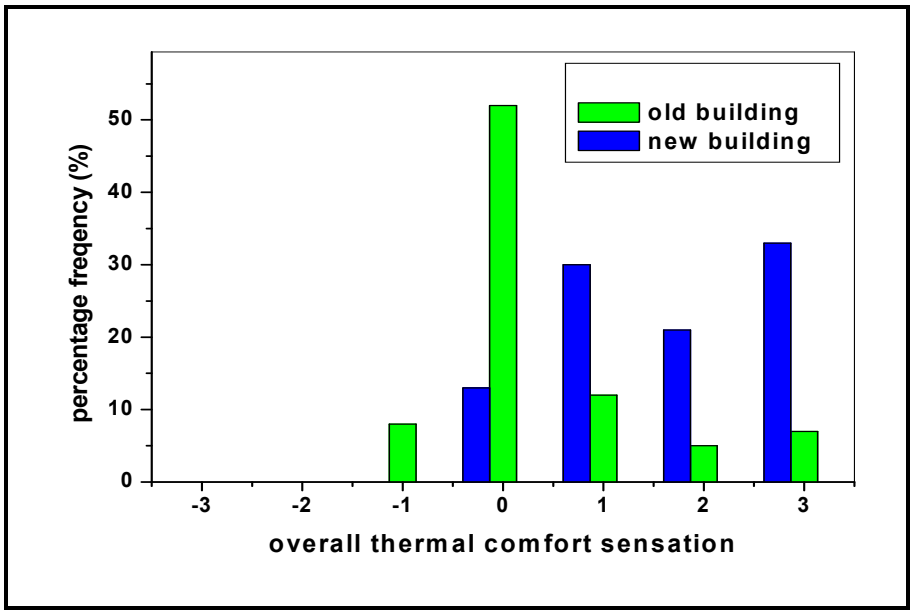


Fig. 4.2.1 overall thermal comfort sensation for old buildings and new buildings.

It illustrates that 54% of the respondents are feeling neutral (0) in the old buildings and only 15% of the respondents in new buildings are feeling neutral. In addition, 13% of people reported as being slightly cool (−1) in the old buildings compared with 0% in the new buildings, with 8% of them feeling hot (3) in the old buildings, and 33% feeling hot in new buildings. This results therefore suggest that the

occupants have an overall impression of higher standards of thermal comfort in old buildings than in new buildings.

Furthermore, the survey showed, from the preference scale, that 62% of the residents in old buildings did not want a change in their indoor environment, while 38% wanted to be cooler. By comparison, only 41% of new building occupants voted for no change with 59% who wanted to be cooler. At the same time there were 96% who were generally satisfied with their environment of the old buildings, compared with 77% of occupants of the new buildings.

4.2.2. Physical measurements

In addition to the subjective study, an objective survey was carried out in 11 buildings (5 old, 6 new), representing 50 subjects. All subjects were sitting on the floor, and they were wearing their traditional uniform of 0.6 clo. The sensors to measure the four basic environmental parameters were placed at a height of 0.3 m above the floor, representing the centre of gravity of the subject. The measurements were made at the same time as the respondents were completing the questionnaires. A sample of the results of the measurements of each type of building is illustrated in Table.4.2.1 Only 11 buildings (5 old and 6 new). The study was undertaken in four weeks in each summer season, therefore it was only possible to survey 11 buildings, and (iii) nature of the research-had to make prior-arrangements with local authorities in Ghadames city introduce and interview residents, and finally, select buildings whose residents agreed to participate in the survey. All of these issues together with the usual technical and logistic problems of site-work limited the sample size. Nevertheless the sample was selected in such a way that a reasonable cross section of house-holds was involved.

Table 4.2.1. Sample of the measurement results of old and new buildings in Ghadames, 2006

Building-Type.No	t-out (C^o)	ta (C^o)	Va (m/s)	Activity (met)	Rcl (Clo)	PMV	PPD
3 – Old	43.0	33.6	0.04	1.1	0.6	2.9	85
5 – Old	40.0	33.8	0.05	1.1	0.6	2.7	80
7 – New	36.2	28.8	0.20	1.1	0.6	0.7	20
10– New	39.1	33.0	0.19	1.1	0.6	1.8	60

4.2.3. Applicability of CR 1752

The basic and detailed parts of CR 1752 [3] . But Libya are working generally by ISO 7730 [18] therefore in the text we refer for this. The thermal comfort index PMV and PPD are considered to be applicable in the Ghadames environment, for the following main reasons; (a) Fanger's work investigated about 1296 subjects, with wide range of environmental conditions, including conditions when air temperature >30°C, and relative humidity up to 70%; and (b) a moderate thermal environment is deemed to have been achieved when PMV values range from −3 as cold to +3 as hot. Therefore, the Ghadames' environment is considered as a moderate thermal environment and the ISO 7730 [18], is expected to be applicable in such environments. In the present survey 50 subjects in good health (22 residents from 5 old buildings and 28 residents from 6 new buildings), in the age range 19–70 years, were asked to complete the questionnaire to determine the AMV. In the objective tests, measurements of environmental variables were used to calculate the PMV (Advanced Mean Vote) of the subjects using Fanger's model as presented in ISO 7730 [18], The thermal comfort programmer developed by D.L. Loveday et al. [41] has been used to calculate PMV and PPD values. The measurements in the new buildings were conducted with the air-conditioning system operating, whilst the old buildings depended on natural ventilation. The metabolic rate was viewed to be 1.1 met (63.8 W/m^2) to represent a sedentary activity, and clothing value, Rcl, was estimated to be 0.6 clo representing the thermal resistance of a traditional uniform.

Fig. 4.2.2(a) and (b) show the comparison of PMV and AMV for both old buildings and new buildings show in the Fig 4.2.2. (a) and (b). Each point in the figures represents an average vote of 4.5 and 4.8 subjects respectively. Fig. 4.2.2 (a) shows that the subjects were feeling neutral to slightly warm in old buildings, even when the indoor air-temperature ranged from 30 to 35°C. It can be seen that there are clear discrepancies between PMV and AMV for the old buildings. This shows that Fanger's model is invalid for predicting the thermal comfort in such environments. Adaptation effects could explain this.

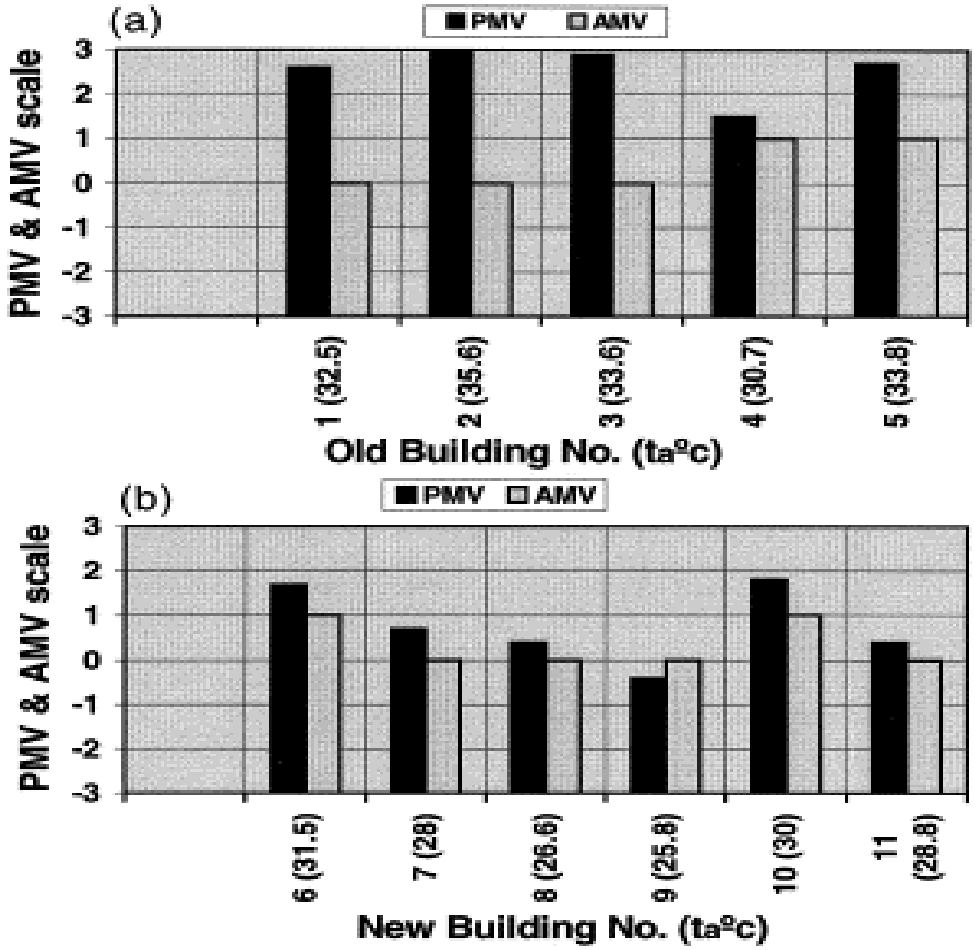


Fig. 4.2.2 (a) Comparison of PMV and AMV at 18:45 p.m. in five old buildings, 2004. (b) Comparison of PMV and AMV at 18:45 p.m. in six new buildings, 2004 (air-conditioning systems turned on).

Fig. 4.2.2 (b) shows good agreement between the PMV values and the AMV of the occupants in new air-conditioned buildings. It shows that 67% of subjects in four new buildings were feeling neutral, while 33% were feeling slightly warm.

In addition to the physical measurements, two personal parameters need to be estimated; clothing value and metabolic rate. For such estimation, it is important to take account of the uncertainties that are associated with these two parameters within the ranges normally found in the conditions of the field measurement. Thus, clothing values between 0.6–0.7 clo would be normally found on the population in Ghadames, with metabolic rates between 1.0 and 1.2 met in the sedentary posture adopted during the survey. Thus, a recalculation of the PMV and PPD values was carried out for the additional values of clothing and metabolic rate. These new values of PMV and PPD as shown in Table (4.2.2), suggested

that those possible different values of Rcl and metabolic rates had no significant effect. Thermal resistance of the clothing is not important when the temperature differences between the body surface and air is small, and heat loss is mainly by evaporation because the climate is dry.

Table 4.4.2. Effect of different Rcl and met values on the PMV and PPD

Building No	Rcl value (m2K/W) (Clo)	Metabolic (W/m2) (met)	PMV value	Metabolic (W/m2) (met)	PMV value	PPD value
Building (5)	0.095 (0.7)	63.8 (1.1)	2.7	58.15 (1.0)	2.8	84
	0.085 (0.6)	63.8 (1.1)	2.7	63.8 (1.1)	2.7	80
	0.08 (0.5)	63.8 (1.1)	2.7	70.0 (1.2)	2.7	80
Building (11)	0.095 (0.7)	63.8 (1.1)	0.5	58.15 (1.0)	0.1	6
	0.085 (0.7)	63.8 (1.1)	0.4	63.8 (1.1)	0.4	8
	0.095 (0.7)	63.8 (1.1)	0.3	70.15 (1.2)	0.5	12

It is concluded that the PMV values from ISO 7730 [18], still show disagreement with AMV values reported by the occupants in old buildings. However, if in new air-conditioned buildings, PMV and PPD values would change by the maximum +0.4 scale value for all conditions. The new values of PMV and PPD still show agreement with the AMV values reported by the occupants in new buildings.

CHAPTER 5

Thermal comfort investigation

1. **Manikin with ventilation and Subjects**

2. Manikin with ventilation

3. Manikin without ventilation

5.1. Manikin with ventilation and Subjects [42]

This investigation with manikin and subjects made by. K.W.D. Cheong, W.J. Yu' ,R. Kosonen, K.W. Tham and S.C. Sekhar [42], a thermal comfort study using a thermal manikin in a field environment chamber served by the Displacement Ventilation (DV) system. The manikin has a female body with 26 individually heated and controlled body segments. The manikin together with subjects was exposed to 3 levels of vertical air temperature gradients, nominally 1, 3 & 5 K/m, between 0.1 and 1.1 m heights at 3 room air temperatures of 20, 23 and 26 °C at 0.6 m height. Relative humidity at 0.6 m height and air velocity near the manikin and the subjects were maintained at 50% and less than 0.2 m/s, respectively. The aims of this study are to assess thermally non-uniform environment served by DV system using the manikin and correlate the subjective responses with measurements from the manikin. The main findings indicate that room air temperature had greater influence on overall and local thermal sensations and comfort than temperature gradient. Local thermal discomfort decreased with increase of room air temperature at overall thermally neutral state. The local discomfort was affected by overall thermal sensation and was lower at overall thermally neutral state than at overall cold and cool sensations.

In a space served by Displacement Ventilation (DV) system, air temperature, air velocity, turbulence level, etc. at ankle level are always different from those at head level [43] and [44]. In a thermally non-uniform condition, overall thermal neutrality is not always sufficient to provide thermal comfort. Overall thermal comfort (OTC) is experienced by the body in general and it is usually associated with a person's energy balance in a given environment. On the contrary, local thermal comfort takes into consideration the comfort level of different body segments. Therefore, it is possible to be in thermal comfort at a global level but still feel some local discomfort [45], [46] and [47]. This may be caused by an asymmetric radiant field, a local convective cooling of the body (draught), contact with a warm or cool floor, or by a vertical air temperature gradient. There are several variables of thermal stimulation that influence the magnitude of thermal sensation. Among the more important of variables are skin temperature, rate of change of the temperature, and the size of the area of skin over which the temperature change is applied, H. Hensel [48], suggested that warm and cold sensations can be expressed as a function of the temperature of the skin, the rate of change of skin temperature and the stimulation area. With the core temperature at steady state, changes in the average skin temperature are the cause for all changes in whole body thermal sensation. B.W. Olesen et al. [49] showed that the greatest non-uniformity of the skin temperature distribution over the body was partially due to lower

temperature at the person's feet, who was clothed in standard clothing (0.6 clo.) and cotton sweat socks (no shoes). When the subjects felt optimally comfortable, the trunk was the warmest part of the body and extremities the coolest.

In a thermally non-uniform environment, it is inaccurate to assess the environment using the conventional way by measuring several physical parameters including air temperature, air velocity, etc. Although thermal manikins are originally developed to measure thermal insulation of clothing, they have also been used for evaluation of microclimate conditions served by different ventilation systems. The most significant performance feature of thermal manikins is their capability of providing rapid, accurate and reproducible simulation of human body heat exchanges. They do so by measuring convective, radiative and conductive heat losses over surface of the whole body in all directions. This distinctive performance feature makes thermal manikins particularly useful in assessing thermally non-uniform environment such as in the space served by DV system.

There are a few researches on the evaluation of thermally non-uniform environment using a manikin. D.P. Wyon and Sandberg [50] carried out a thermal comfort experiment in a test room equipped with DV system by using a thermal manikin. It was found that thermal conditions above the table height were generally acceptable but cold discomfort was observed at the legs. Local discomfort was identified and mostly due to cold legs, ankles and feet. In addition, the results indicated that Equivalent Homogeneous Temperature (EHT) of 25.1 °C was preferred for whole-body condition. An optimum sectional air-temperature of 24.4 °C was suggested for average thermal sensation to be 'neutral' and a range, $20.9\,^{\circ}\mathrm{C} < T < 28.0\,^{\circ}\mathrm{C}$, based on 80% acceptability criterion was proposed. S. Tanabe et al. [51] developed a method for measuring thermally non-uniform environments using a thermal manikin with controlled skin surface temperature for an under-floor air distribution system. Equivalent temperature based on a thermal manikin was proposed to measure and evaluate the thermal environment. A method to calculate the PMV index from manikin heat loss was also given. P.V. Nielsen et al. [52] used a thermal manikin with 16 individually heated and controlled segments to assess the state of thermal comfort and local discomfort. The manikin was placed at different locations in a room served by both Mixing Ventilation (MV) and DV systems. The EHT expressed the global thermal comfort at the location of the manikin. The average value of EHT for DV was 23.9 °C, which is slightly above the requirement for thermal comfort. The difference between the largest and the smallest EHT (ΔEHT) for the elements of the manikin could be used as an indication of local discomfort due to the air temperature gradient, draft, turbulence, and un-symmetric radiation.

It was found that temperature gradient seemed to be a large source of local discomfort in the space served by DV system in comparison to MV system as following :

(1) To investigate the distribution of skin surface temperature and sensible heat loss of a manikin in the space served by DV system for different temperature gradients at different room temperatures at 0.6 m height.

(2) To correlate the subjective responses with the measurements from the thermal manikin.

5.1.2. Experimental facilities

This study was conducted in a Field Environment Chamber (FEC) as shown in Fig.5.1.1 at the National University of Singapore between July and September 2004.The chamber, 11.12 m (L)×7.53 m (W)×2.60 m (H), has an east-facing wall comprising of large glass panels which are insulated with aluminium foil externally and furnished with blinds internally to reduce heat conduction and solar radiation. The chamber is equipped with an Air-Conditioning and Mechanical Ventilation (ACMV) system that is capable of switching between DV and MV modes.

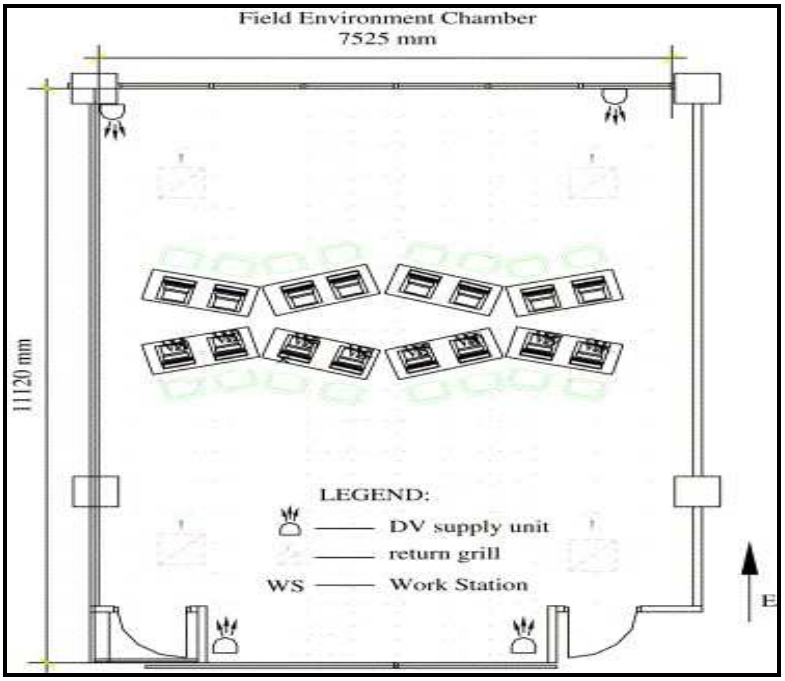


Fig. 5.1.1. Layout of FEC.

The manikin used in the experiment is an average-sized female with 1.68 m standing height as shown in Fig. 5.1.2.

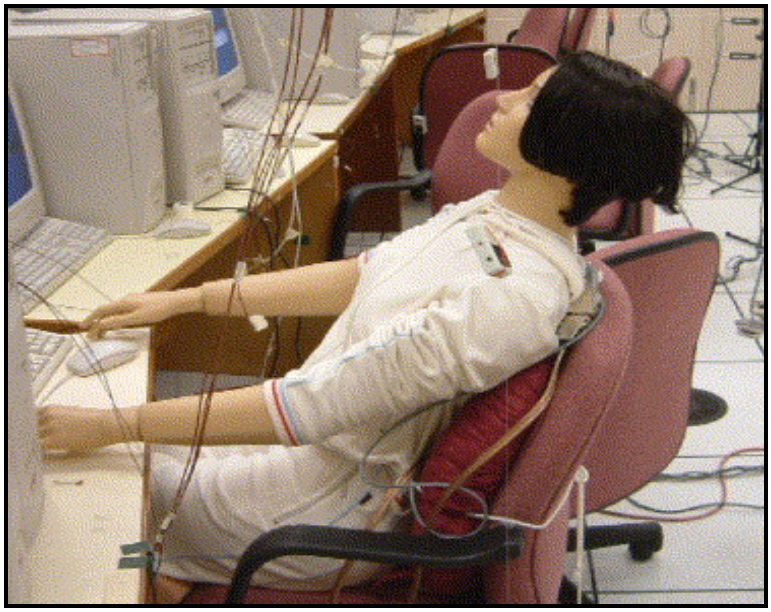


Fig. 5.1.2. The thermal manikin.

It is divided into 26 thermal segments that can be independently controlled and measured. Table 5.1.1 shows the 26 body segments and their respective surface areas. The manikin is controlled by software that has four control modes, namely,

(1) Only measuring—no heat,

(2) Heating to a fixed set point,

(3) Heating with fixed heat loss,

(4) Following a comfort equation.

Table 5.1.1. Body segments and respective areas of the manikin

No	Name of body segments	Area (m^2)
1	L. foot	0.043
2	R. Foot	0.043
3	L. low leg	0.09
4	R. low leg	0.09
5	L. front thigh	0.085
6	R. front thigh	0.088
7	L. back thigh	0.075
8	R. back thigh	0.078
9	Pelvis	0.055
10	Back side	0.11
11	Skull	0.05
12	L. face	0.0258
13	R. face	0.0258
14	Back of neck	0.0248
15	L. hand	0.038
16	R. hand	0.037
17	L. forearm	0.05
18	R. forearm	0.05
19	L. upper arm out	0.0419
20	R. upper arm out	0.0436
21	L. upper arm in	0.0319
22	R. upper arm in	0.0336
23	L. chest	0.07
24	R. chest	0.07
25	L. back	0.065

No	Name of body segments	Area (m^2)
26	R. back	0.065
Total	R. forearm	1.479

In the last mode, comfort mode, surface temperatures of various body segments of the thermal manikin follow a comfort equation and are allowed to change to adapt to the environment. So the manikin is always in thermally neutral state. The comfort equation, t_s=36.4-CQ_t which controls the surface temperature and heat output corresponding to a person in thermal comfort, is derived based on Fanger's comfort criteria. Where t_s is the skin surface temperature (°C), Q_t is measured specific sensible heat loss (W/m^2), C equals to 0.054 (m^2 °C/W).

Based on this equation, skin temperature of various segments will adapt to the environmental conditions to maintain thermal neutrality. The manikin does so by compensating heat loss from the skin surface that will cause a drop in the skin temperature with heat supplied to the heating elements at different segments. Hence, under steady-state conditions, the heat supplied to different segments will be equivalent to the heat loss from the skin surface. Subsequently, the heat loss per unit skin surface can be derived from the surface area and electricity consumption of each segment. Each body segment has its own unique micro controller system that calculates the temperature of the entire surface by measuring the resistance of the nickel wire. In addition, it controls a power switch for heating and calculates the power consumption. The controllers at the various body segments are connected to a computer where measurements such as heat loss per unit skin surface and skin surface temperature are recorded approximately at every half minute interval. In addition, the mean surface temperature and the mean unit heat loss are also recoded.

5.1.3. Experimental design

The experiment was divided into two Stages. In Stage 1, the seated manikin was controlled at thermal comfort mode and exposed to 3 room air temperatures at 0.6 m height (nominally 20, 23 and 26 °C) with 3 vertical air temperature gradients between 0.1 and 1.1 m heights (nominally 1, 3 and 5 K/m). At the thermal comfort mode, the skin temperatures of the manikin's body segments follow the comfort equation and are allowed to change to adapt to the environment, so that the manikin is always in thermal neutrality. In order to correlate subjective responses with the measurements from the manikin at the same overall thermal state, in this experimental stage, 30 subjects were allowed to adjust their

clothing to achieve thermal neutrality during the first 2 h. Jackets were available for subjects who wanted to put on more clothing. In Stage 2, the manikin was controlled at fixed sensible heat mode and exposed to room air temperatures of 20 and 26 °C at 0.6 m height with 3 vertical air temperature gradients of 1, 3 and 5 K/m between 0.1 and 1.1 m heights. Other 30 subjects were exposed to the same room condition together with the manikin.

5.1.4. Objective measurements

Measurement of room air temperature was carried out using type T thermocouple wire with accuracy of ±0.2 °C at 0.1, 0.6, 0.8, 1.1, 1.7 and 2.5 m heights. RH was measured using portable sensor with accuracy of ± 5% at 0.1, 0.6, 0.8, 1.1, 1.7 and 2.5 m heights. Air velocity was measured at 0.1, 0.6, 0.8 and 1.1 m heights near the manikin and the subjects using omni-directional hot wire type of anemometer probes with accuracy of 0.01 m/s.

5.1.5. Subjective assessment

The ASHRAE scale, (−3) cold, (−2) cool, (−1) slightly cool, (0) neutral, (+1) slightly warm, (+2) warm and (+3) hot, was used for the assessment of subjects' thermal sensation. Thermal comfort was assessed by using the Bedford's scale. For draft sensation, a Yes/No category scale was used for the air movement sensation, whereby subjects answering "Yes" for this question, would be required to answer the following three questions: which body parts felt air movement; the acceptability level of the air movement. Draft is unwanted local cooling of the body caused by air movement ASHRAE, [53].

5.1.6. Subjects

A total of 60 subjects, 30 for Stage 1 and 30 for Stage 2, were selected for the experiments. Only students who are acclimatized to the tropical climate were chosen. Table. 5.1.2. shows the data of subjects for this study.

Table 5.1.2. Data of subjects

Sensation	Feeling Neutral			Feeling Cold and warm		
Gender	Females	Males	Total	Females	Males	Total

Sensation	Feeling Neutral			Feeling Cold and warm		
Gender	Females	Males	Total	Females	Males	Total
Numbers	15	15	30	15	15	30
Age (years)	21.7 ± 1.0	23.2 ± 2.8	22.5 ± 2.2	21.8 ± 0.9	24 ± 2.3	22.9 ± 2.1
Height (cm)	160.9 ± 5.6	169.1± 7.0	165 ± 7.5	161.9 ± 5.4	172.1 ± 5.0	167 ± 7.3
Weight (kg)	50.6 ± 6.9	63.9 ± 10.8	57.3 ± 11.2	52.1± 7.4	66.7 ± 10.4	59.4 ± 11.6

5.1.7. Results and Experimental conditions

Table (5.1.3) shows the summary of the nominal and actual experimental conditions. For nominal room air temperatures (Tr) of 20, 23 and 26 °C at 0.6 m height, respective actual values were in the ranges of 19.9–20.3, 22.9–23 and 25.9–26.2 °C.

Table 5.1.3. Experimental conditions

Control mode	Case	Nominal value		Actual value					
		Tr (°C)	Δ*t* (K/m)	Tr (°C)		RH (%)		*V* (m/s)	
Stage 1				Mean	S.D	Mean	S.D	Mean	S.D
Comfort	1	20	1	20.3	0.3	51.5	1.4	0.10	0.03
	2	20	3	20.0	0.2	50.4	0.7	0.07	0.01
	3	20	5	19.9	0.2	49.7	1.3	0.09	0.02
	4	23	1	22.9	0.1	53.9	1.2	0.10	0.04
	5	23	3	24.0	0.3	50.9	1.0	0.07	0.01
	6	23	5	24.0	0.3	50.9	0.6	0.08	0.01

Control mode	Case	Nominal value		Actual value					
		Tr (°C)	Δ*t* (K/m)	Tr (°C)		RH (%)		*V* (m/s)	
Stage 1				Mean	S.D	Mean	S.D	Mean	S.D
	7	26	1	26.1	0.2	53.4	1.4	0.08	0.02
	8	26	3	26.2	0.2	51.1	1.2	0.07	0.01
Stage 2	9	26	5	25.9	0.2	52.7	1.4	0.08	0.02
Heat loss 65 W/m^2	10	20	1	20.1	0.2	50.7	1.3	0.12	0.05
	11	20	3	20.2	0.2	51.2	0.6	0.09	0.01
	12	20	5	20.0	0.3	51.5	0.6	0.08	0.01
Heat loss 35 W/m^2	13	26	1	26.1	0.2	51.9	1.5	0.08	0.03
	14	26	3	26.2	0.1	52.7	1.1	0.07	0.02
	15	26	5	25.9	0.2	52.1	1.1	0.07	0.01

The set points of room air temperature (at 0.6 m height), temperature gradient (between 0.1 and 1.1 m heights) and RH (at 0.6 m height) were achieved by adjusting supply airflow rate and temperature with constant heat load for all the cases. The recorded data of trial tests were used as reference for the experiment. The supply airflow rates (Ls) and air supply temperatures (Ts) for all the cases are shown in Table 5.1.4.

Table. 5.1.4. Supply airflow rates and temperatures

Stage	1	2	
Control mode	Comfort	Heat loss of 65 W/m^2	Heat loss of 35 W/m^2

Case	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Ls (m^3/h)	2345	1300	955	2200	1135	650	1800	790	510	2400	1325	970	1780	760	500
Ts (°C)	16.9	15.2	12.3	20.2	18.4	15.9	24.3	21.4	17.5	16.7	15.1	12.1	24.2	21.5	17.6

5.1.8. Profiles of spatial temperature and velocity

Typical spatial temperature and velocity profiles for Cases 1, 2 and 3 are shown in Fig. (5.1.3) and Fig. (5.1.4) Results show that temperature gradients were steeper below 1.1 m height than beyond 1.1 m height. These results are similar to the findings of, M. XU et al. [54]. It can be seen from Fig. 5.1.4 that the velocity at 0.1 m height was the highest and reduced with the increase in height. At 0.1 m height, the velocity was around 0.1 m/s.

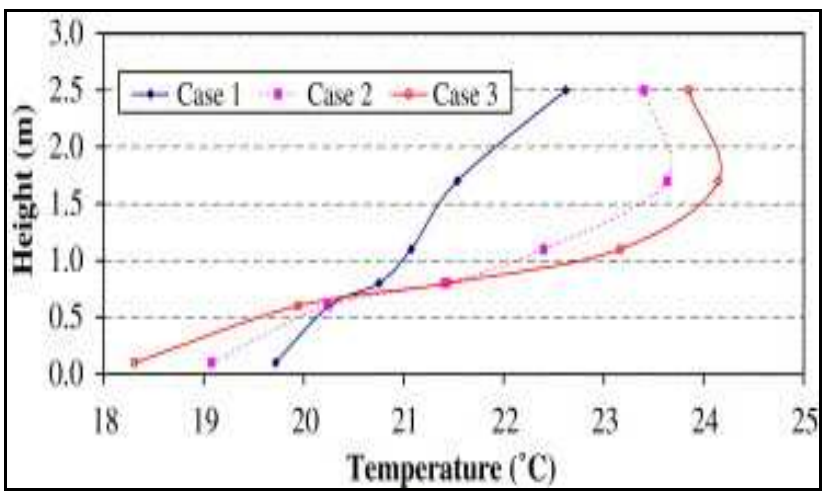


Fig. 5.1.3. Temperature profiles for Cases 1, 2 and 3.

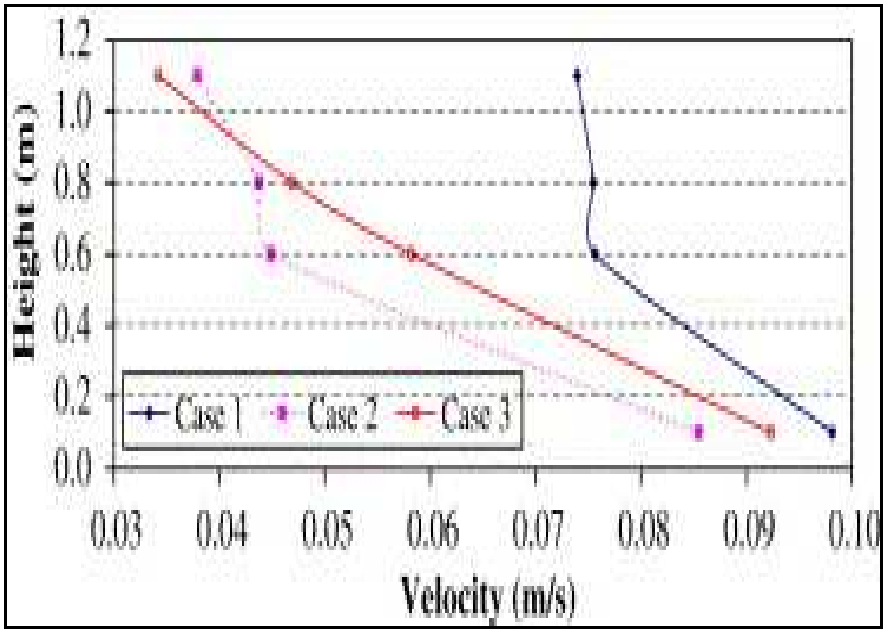


Fig. 5.1.4. Velocity profiles for Cases 1, 2 and 3.

For case 1, 2 and 3 of air velocity changing from the supply air flow in down to up as you see in Fig. 5.1.4 when the height 0.1 m the speed of air velocity high and when 1.0 height the air velocity seeped decrease.

5.1.9. Overall thermal sensation.

Table 5.1.5 shows values of overall thermal clothing value (OTC) and mean sensible body heat loss for all the cases. For Cases 1–9 in Stage 1, subjects were allowed to adjust their clothing to achieve thermal neutrality. The values of OTS were within the range of between −0.6 and 0.10, which was close to the neutral and the values of OTC were within the comfort range of between −0.53 and 0.13. The manikin for Cases 1–9 was in comfort mode. For Cases 1–3 at room temperature of 20 °C with temperature gradients of 1, 3 and 5 K/m, mean body heat losses were 56.0, 53.4 and 52.8 W/m^2, respectively. Average mean body heat loss of the 3 Cases was around 54.1 W/m^2.

For Cases 4–6 at room temperature of 23 °C with temperature gradients of 1, 3 and 5 K/m, mean body heat losses were 46.7, 44 and 45.9 W/m^2, respectively. For Cases 7–9 at room temperature of 26 °C with temperature gradients of 1, 3 and 5 K/m, mean body heat losses were 32.3, 34.4 and 33.2 W/m^2, respectively. Average mean body heat loss of the 3 Cases was around 33.3 W/m^2. The results show that

at comfort mode, mean body heat losses for the three temperature gradients at certain room temperature were about the same.

Table 5.1.5. Values of OTS, OTC, clothing value and mean body heat loss

Stage	1									2					
Cases	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
S.D.	0.63	0.56	0.63	0.81	0.63	0.78	0.7	0.5	0.5	0.70	0.83	0.86	0.4	0.4	0.3
OTC	−0.47	−0.60	−0.50	−0.50	−0.53	−0.37	−0.1	0.13	0	−2.10	−1.97	−1.73	0.8	1.0	0.9
S.D.	0.73	0.62	0.82	0.82	0.63	0.76	0.7	0.5	0.5	0.76	0.85	0.74	0.6	0.6	0.4
Clo.v	1.15	1.13	1.12	0.91	0.89	0.89	0.6	0.6	0.6	0.73	0.73	0.77	0.8	0.8	0.8
S.D.	0.05	0.09	0.10	0.08	0.09	0.07	0.06	0.06	0.06	0.10	0.09	0.09	0.13	0.13	0.11
heat loss	56.0	53.4	52.8	46.7	44	45.9	32.3	34	33	65	65	65	35	35	35

In Stage 2, the manikin operated in fixed heat loss output of 65 and 35 W/m^2 for Cases 10–12 and 13–15, respectively. Cases 10–12 at room temperature of 20 °C corresponded to cold condition for the manikin because the mean heat loss was fixed at 65 W/m^2, which is higher than at room temperature of 20 °C for comfort mode. Cases 13–15 at room temperature of 26 °C corresponded to slightly cool condition for the manikin because the heat loss is fixed at 35 W/m^2, which is slightly higher than at room air temperature of 26 °C for comfort mode. Subjects were not allowed to adjust their clothing in these cases. For Cases 10–12, the values of OTS were close to cold sensation within the range of between −2.30 and −1.87. Average clothing value of subjects was around 0.74 clo. For Cases 13–15, the values of OTS were slightly warm within the range of between 1.1 and 1.2. Average clothing value of subjects was around 0.82 clo.

5.1.10. Effects of temperature gradient and thermal sensation on skin surface.

Fig. 5.1.5 shows overall average and different body segments' skin surface temperatures as the manikin was exposed to room temperature of 20 °C at 3 temperature gradients in comfort mode and fixed heat

loss output of 65 W/m^2. For Cases 1, 2 and 3 with thermal comfort mode, overall average skin surface temperatures were 33.4, 33.5 and 33.5 °C at 3 temperature gradients of 1, 3 and 5 K/m, respectively. For Cases 10, 11 and 12 with the fixed heat loss output, overall average skin surface temperatures of the manikin were 32.6, 32.9 and 32.7 °C at temperature gradients of 1, 3 and 5 K/m, respectively. It appears that at comfort or fixed heat loss modes. The results show that hands, instead of feet, had the lowest surface temperature out of 26 body segments. It is due to that the manikin wore socks but with hands exposed to the air directly during the experiment. The results also show that the back and skull had the highest surface temperature, implying that those were the warmest body parts.

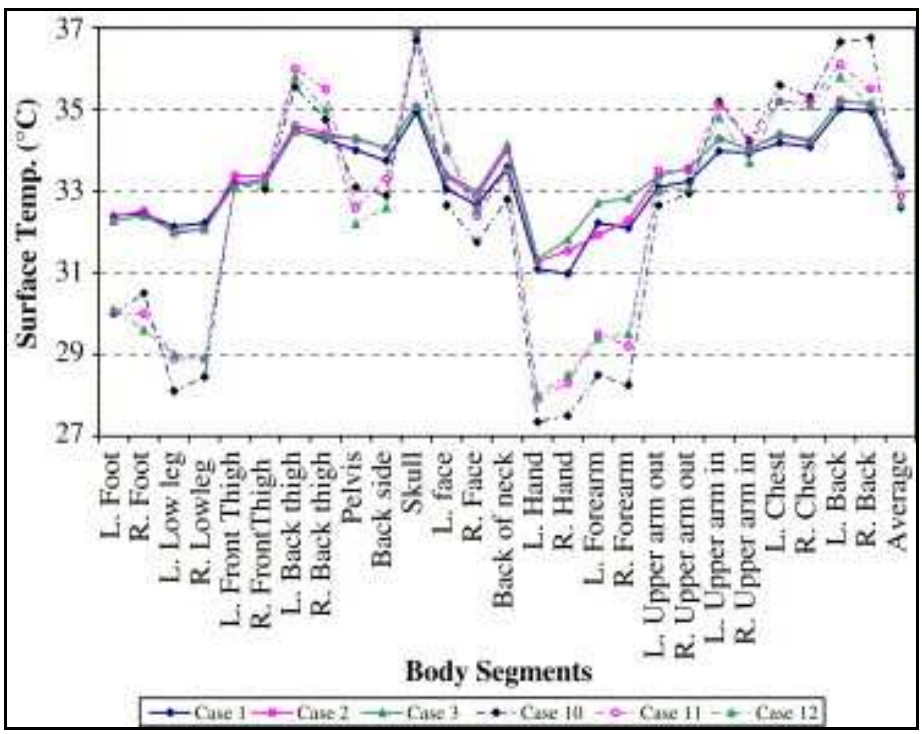


Fig. 5.1.5. Profiles of skin surface temperature of the manikin at temperature of 20 °C.

In the comfort mode, at temperature gradient of 1 K/m, the highest surface temperature was 35 °C at left back while the lowest is 31 °C at the right hand. The highest and lowest surface temperatures were 35.2 (left back) and 31.3 °C (left hand), respectively, at temperature gradients of 3 and 5 K/m. The larger temperature variation of 4 °C occurred

at temperature gradient of 1 K/m while the smaller variation of 3.9 °C was experienced at temperature gradients of 3 and 5 K/m. For fixed heat loss output, the highest surface temperature of 36.8 °C was at right back for gradient of 1 K/m. The highest surface temperature of 36.9 °C was at the skull for

gradients of 3 and 5 K/m. The lowest surface temperatures were 27.4, 27.9 and 28 °C at the left hand for gradients of 1, 3 and 5 K/m, respectively. The variations of the highest and lowest surface temperatures were 9.4, 9 and 8.9 °C for gradients of 1, 3 and 5 K/m, respectively. The results show that the variation of skin surface temperature among body segments for fixed heat loss mode was greater than for thermal comfort mode. However, the variations were almost the same for gradients of 1, 3 and 5 K/m at the comfort or fixed heat loss output modes.

For Cases 4, 5 and 6 with thermal comfort mode at room temperature of 23 °C, overall average skin surface temperatures of the manikin were 33.9, 34 and 33.9 °C for gradients at 1, 3 and 5 K/m, respectively as shown in Fig. 5.1.6. The highest skin surface temperatures were 35.3, 35.4 and 35.4 °C at left back for gradients of 1, 3 and 5 K/m, respectively. The lowest skin surface temperatures were 32.2, 32.1 and 32.3 °C at left hand for gradients of 1, 3 and 5 K/m, respectively. The variations between highest and lowest skin surface temperature were 3.1, 3.3 and 3.1 °C for gradients of 1, 3 and 5 K/m, respectively. This is consistent with earlier findings suggesting that overall average surface temperatures and the variations of skin surface temperature among body segments were almost the same at gradients of 1, 3 and 5 K/m for the comfort mode.

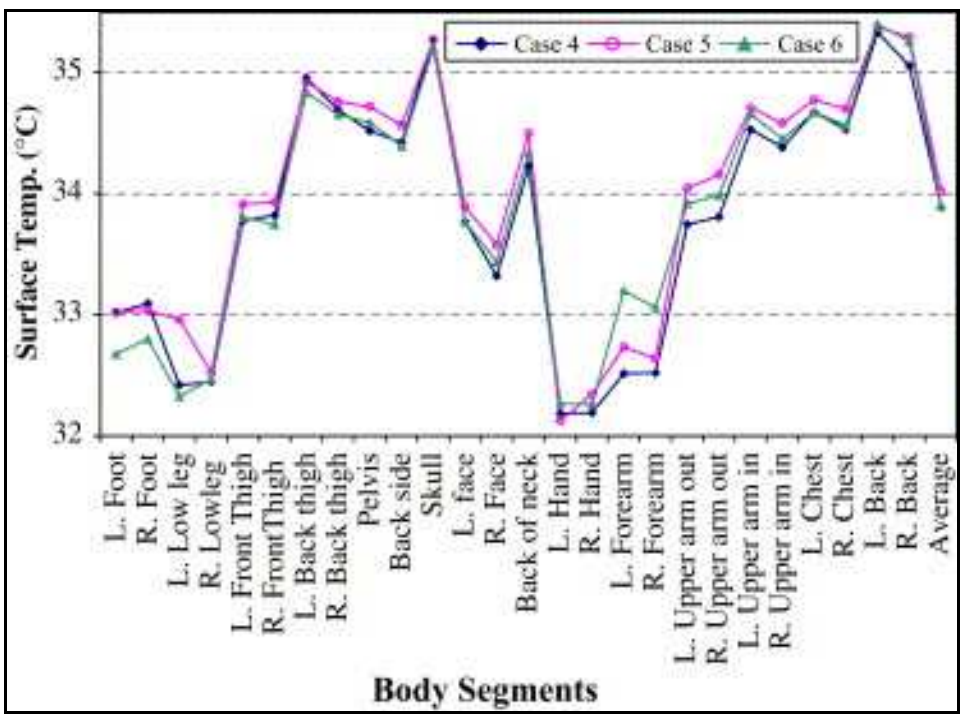


Fig. 5.1.6. Profiles of skin surface temperature of the manikin at 23 °C.

For Cases 7, 8 and 9 with thermal comfort mode at room temperature of 26 °C, overall average skin surface temperatures were 34.7, 34.5 and 34.6 °C for gradients of 1, 3 and 5 K/m, respectively as shown in Fig. 5.1.7. For Cases 13, 14 and 15 with fixed heat loss output of 35 W/m^2, overall average skin surface temperatures of the manikin were 33.7, 33.4 and 33.7 °C for different gradients of 1, 3 and 5 K/m, respectively. For comfort mode, the highest skin surface temperatures were 35.7, 35.6 and 35.7 °C at left back for gradients of 1, 3 and 5 K/m, respectively. The lowest skin surface temperatures were 33.6 (left hand), 33.4 (left hand) and 33.5 °C (left low leg) for gradients of 1, 3 and 5 K/m, respectively. The variations between highest and lowest skin surface temperature were 2.1, 2.2 and 2.2 °C for gradients of 1, 3 and 5 K/m, respectively. For fixed heat loss output of 35 W/m^2, the highest skin surface temperatures were 37 °C at the skull for 3 gradients. The lowest skin surface temperatures were 30.3 (left hand), 30 (right hand) and 30.1 °C (left hand) for gradients of 1, 3 and 5 K/m, respectively. The variations between highest and lowest skin surface temperature were 6.7, 7 and 6.9 °C for gradients of 1, 3 and 5 K/m, respectively. The results confirm that for the comfort or fixed heat loss modes, overall average surface temperatures and the variations of skin surface temperature among body segments for gradients of 1, 3 and 5 K/m were almost the same. However, the variation of skin surface temperature for fixed heat loss mode was greater than for thermal comfort mode.

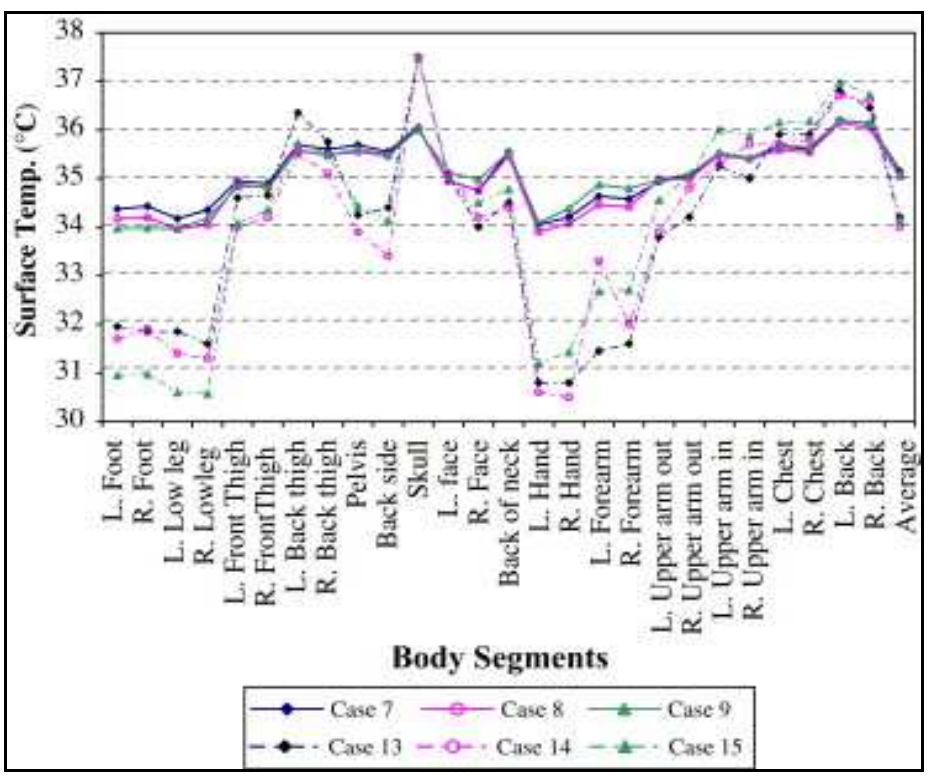


Fig. 5.1.7. Profiles of skin surface temperature of the manikin at 26 °C.

5.1.11. Effect of room air temperature on skin surface temperature

Fig. 5.1.8 displays the profiles of overall average and different body segments' skin surface temperatures for Cases 1, 4 and 7 at respective room air temperatures of 20, 23 and 26 °C with temperature gradient of 1 K/m at the comfort mode. Among overall average and body segments' skin surface temperatures at the 3 room air temperatures, the lowest were always at 20 °C whereas the highest at 26 °C. The findings show that skin surface temperature increased with the increase of room temperature at the comfort mode. This is due to that the heat loss of overall mean and body segments declined with the increase of room air temperature. Similar profiles were observed for Cases 2, 5 and 8 with temperature gradients of 3 K/m and Cases 3, 6 and 9 with temperature gradient of 5 K/m as shown in Fig. (5.1.9) and Fig.(5.1.10).

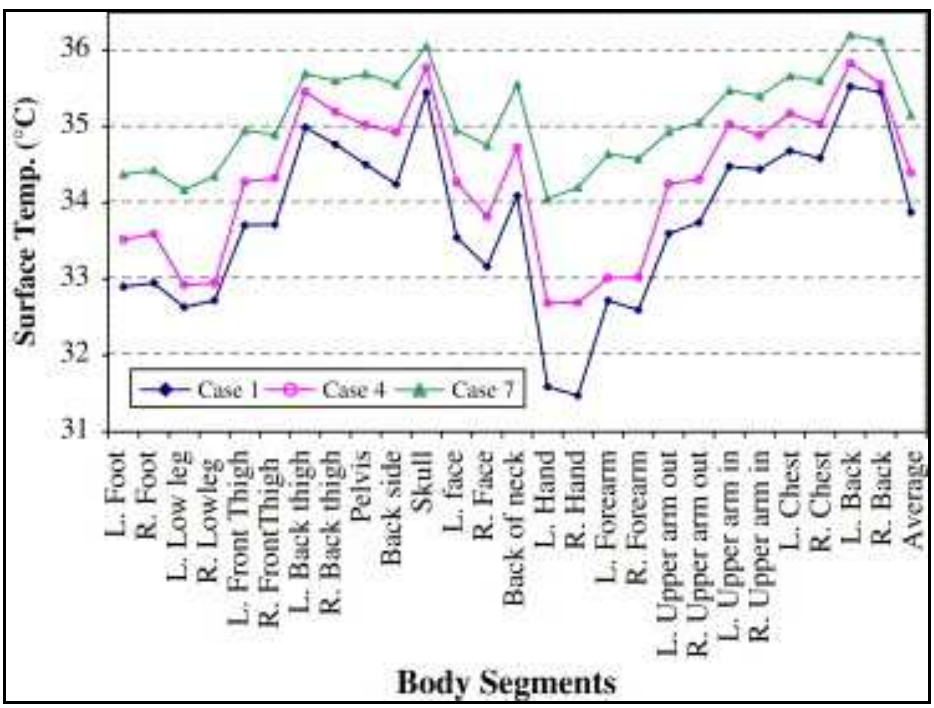


Fig. 5.1.8. Profiles of skin surface temperature at temperature gradient of 1 K/m.

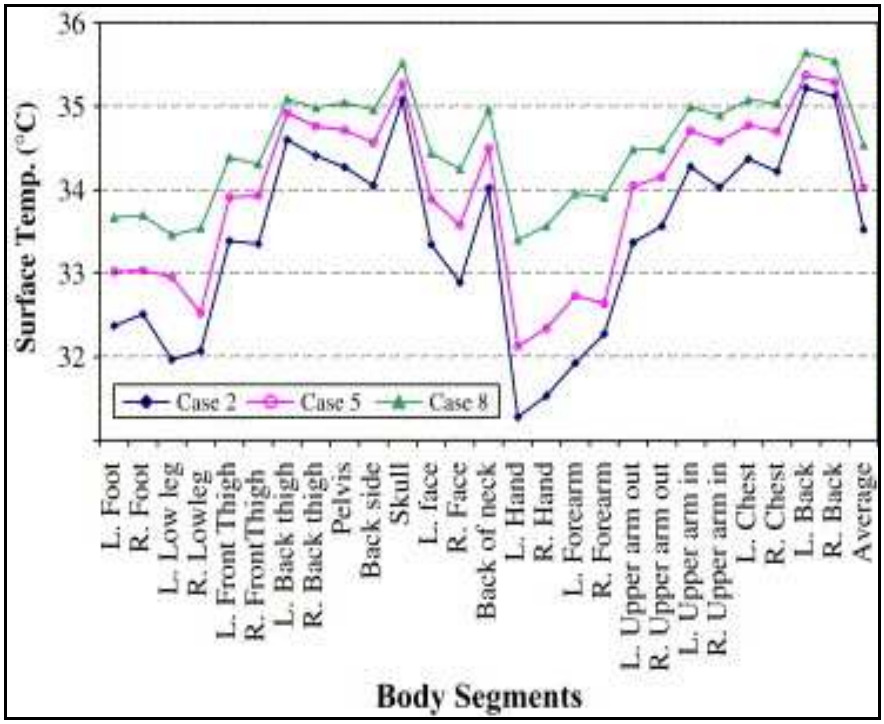


Fig. 5.1.9. Profiles of skin surface temperature at temperature gradient of 3 K/m.

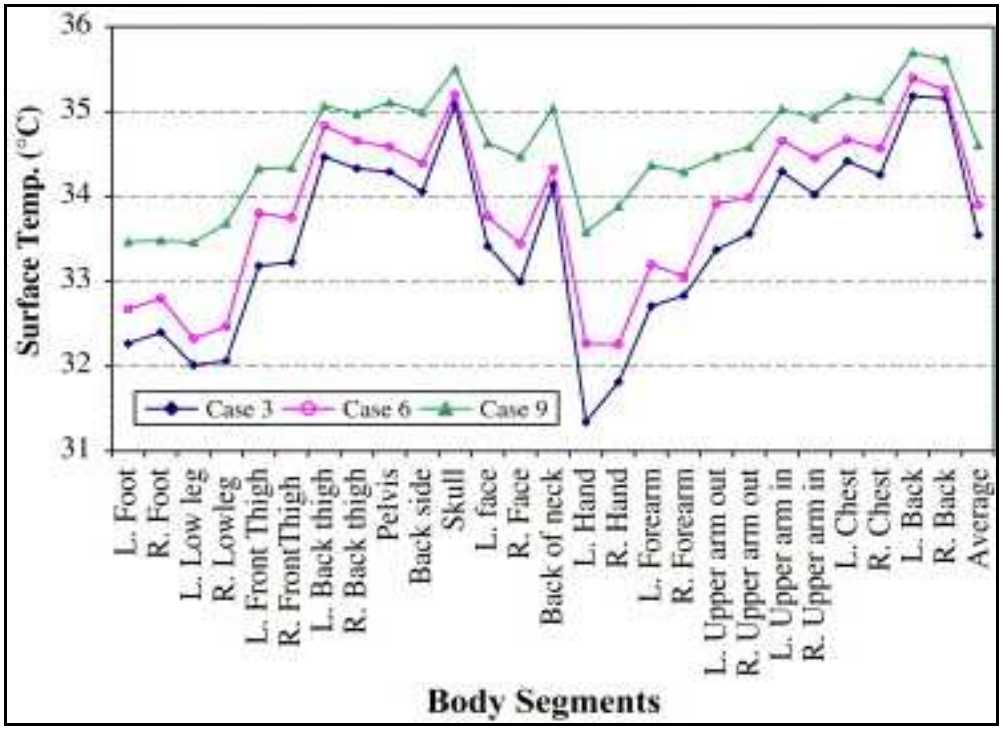


Fig. 5.1.10. Profiles of skin surface temperature at temperature gradient of 5 K/m.

It is observed that at comfort mode for Cases 1, 4 and 7, variations between the highest and lowest skin surface temperature among different body segments were 4, 3.1 and 2.1 °C at room temperatures of 20, 23 and 26 °C, respectively with temperature gradient of 1 K/m. For Cases 2, 4 and 8, the variations were 3.9, 3.3 and 2.1 °C for room temperatures of 20, 23 and 26 °C, respectively with temperature gradient of 3 K/m. For Cases 3, 6 and 9, the variations were 3.9, 3.1 and 2.2 °C for room temperatures of 20, 23 and 26 °C, respectively with temperature gradient of 5 K/m. A linear correlation, ΔT=-0.29t_{room}+9.74, R^2=0.982, existed between the variation (ΔT) and room air temperature (t_{room}). The results indicate that the variation followed a decreasing trend with the increase of room temperature. This is in accordance with the findings of Houdas et al. [55] that skin temperature variation amongst the different body segments was large in cold environment due to vasoconstriction, but much more uniform in warm environments. Local thermal sensation and comfort are closely related with local skin surface temperature. This finding implies that local discomfort decreases with the increase of room air temperature at overall thermally neutral state.

5.1.12. Effect of average skin surface temperature on OTS

Fig. 5.1.11 shows overall average skin surface temperatures and actual values of OTS for Cases 1–12. For comfort mode at room temperature of 20 °C, actual values of OTS of subjects were −0.50, −0.6 and −0.43 and average skin surface temperatures of the manikin were 33.4, 33.5 and 33.5 °C for temperature gradients of 1, 3 and 5 K/m, respectively. At room temperature of 23 °C, actual values of OTS of subjects were −0.38, −0.47 and −0.27 and average skin surface temperatures of the manikin were 33.9, 34 and 33.9 °C for temperature gradients of 1, 3 and 5 K/m, respectively. At room temperature of 26 °C, actual values of OTS of subjects were −0.03, 0.07 and 0.10 and average skin surface temperatures of the manikin were 34.7, 34.5 and 34.6 °C for temperature gradients of 1, 3 and 5 K/m, respectively. For cold sensation, at room temperature of 20 °C, actual values of OTS of subjects were −2.30, −2.07 and −1.87 and average skin surface temperatures of the manikin were 32.6, 32.9 and 32.7 °C for temperature gradients of 1, 3 and 5 K/m, respectively. The results show that when subjects were in neutral sensation at room temperature of 20 and 23 °C, the corresponding average skin temperature of manikin were between 33 and 34 °C. This is in accord with Fanger's finding [56] that skin temperatures associated with comfort at sedentary activities are 33 to 34 °C. However, the average skin temperatures were slightly higher than 34 °C at room temperature of 26 °C. It is likely that clothing value for the manikin was 0.75, which is higher than average clothing value of 0.64 that subjects had at room temperature of 26 °C. Nevertheless, the results demonstrate that actual values of

OTS increased with the increase of average skin surface temperature. A linear correlation (R^2=0.850) between actual value of OTS and average skin surface temperature is shown in Fig. 5.1.11.

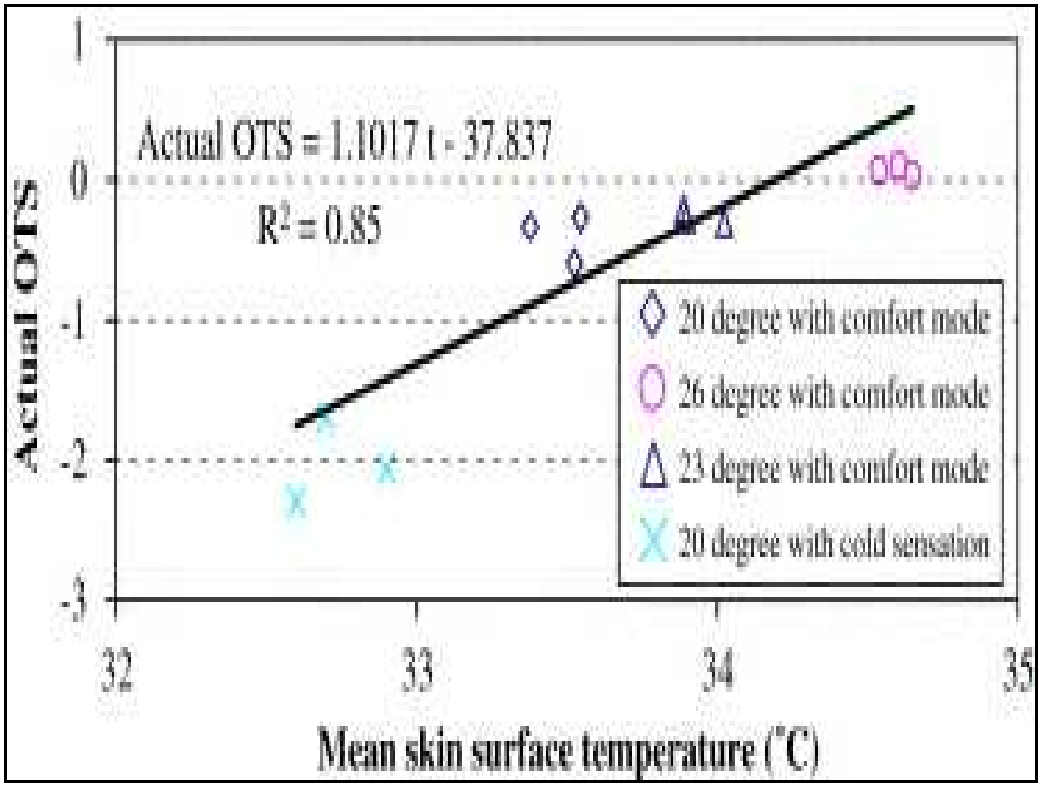


Fig. 5.1.11. Actual values of OTS versus average skin surface temperatures.

5.1.13. Effect of variation of skin surface temperatures on DR

Values of DR and variations of skin surface temperatures among body segments for Cases 1–12 are shown in Fig. 5.1.12. For the comfort mode, at room temperature of 26 °C, values of DR were within the range of 3.3–6.7%, whilst the variations of skin surface temperatures among body segments of the manikin were the lowest within the range of 2.15–2.23 °C. At room temperature of 20 °C, values of DR were within the range of 3.3–10% whilst the variations of skin surface temperatures among body segments of the manikin were within the range of 3.84–4.05 °C. However, at the same room temperature of 20 °C for cold sensation, values of DR increased to the range of 13.3 and 20% whilst the variations of skin surface temperatures among body segments of the manikin increased to the range of 8.9–9.4 °C. The values of DR and the variations of skin surface temperatures among body segments

at overall cold sensation were much higher than at overall neutral sensation. It is possible to speculate that higher variation of skin surface temperatures among body segments will lead to higher values of DR. The possible reason is that at overall neutral and cold sensations, higher variation of surface temperature is caused by lower surface temperature at some body segments. So the body segments will experience colder local thermal sensations and they are vulnerable to draft risk. A linear correlation between value of DR and variation of skin surface temperatures among body segments is obtained (R^2=0.760).

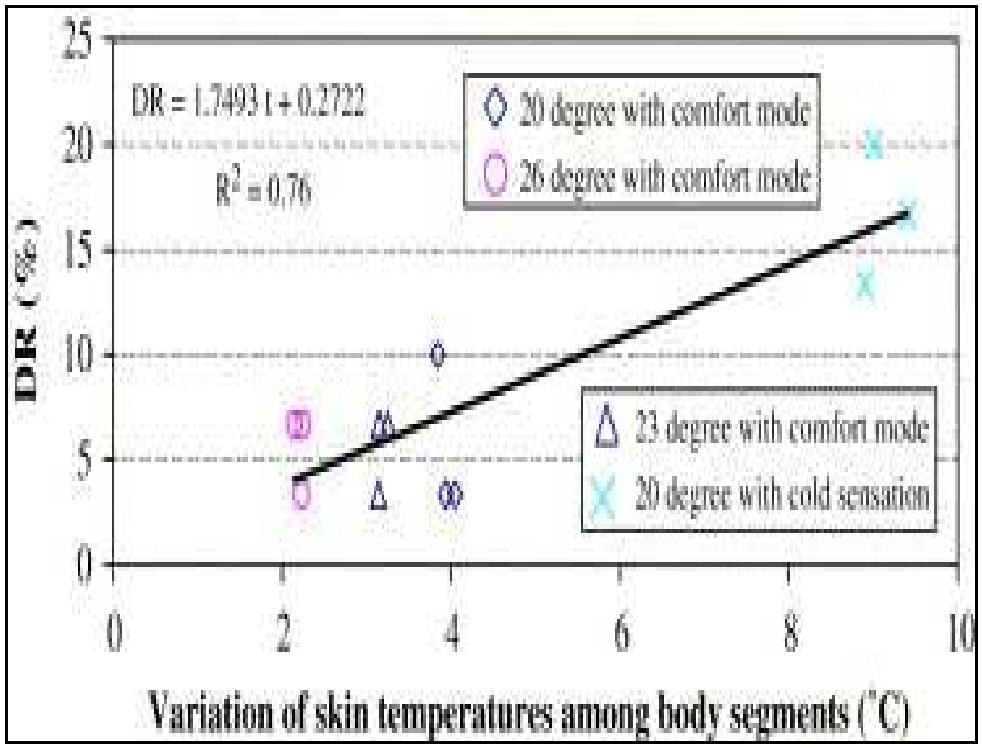


Fig. 5.1.12. Values of DR versus variations of skin temperatures

5.1.14. Effect of variation of skin surface temperatures on (APD)

Values of actual percent dissatisfied (APD) of any body segment and variations of skin surface temperatures among body segments for Cases 1–12 are shown in Fig. 5.1.13. For the comfort mode, at room temperature of 26 °C with temperature gradients of 1, 3 and 5 K/m, values of APD were within the range of 3–6%. Variations of skin surface temperatures among body segments were the lowest in the range of 2.15–2.23 °C. At room temperature of 20 °C with temperature gradients of 1, 3 and 5 K/m,

values of APD were within the range of 50–67%. Variations of skin surface temperatures among body segments were higher in the

range of 3.84–4.05 °C. At the same room temperature of 20 °C with temperature gradients of 1, 3 and 5 K/m for cold sensation, values of APD were much higher at the

range of 87–93%. Variations of skin surface temperatures among body segments were the highest in the range of 8.9–9.4 °C. The results indicate that higher variation of skin surface temperatures among body segments may cause higher percentage of body segments feeling uncomfortable. A linear correlation between value of APD and the variation of skin surface temperatures among body segments is obtained (R^2=0.84). The likely reason is that at overall neutral and cold sensations, higher variation of surface temperature may caused by lower surface temperature at some body segments. Hence, higher percentage of subjects will experience uncomfortable.

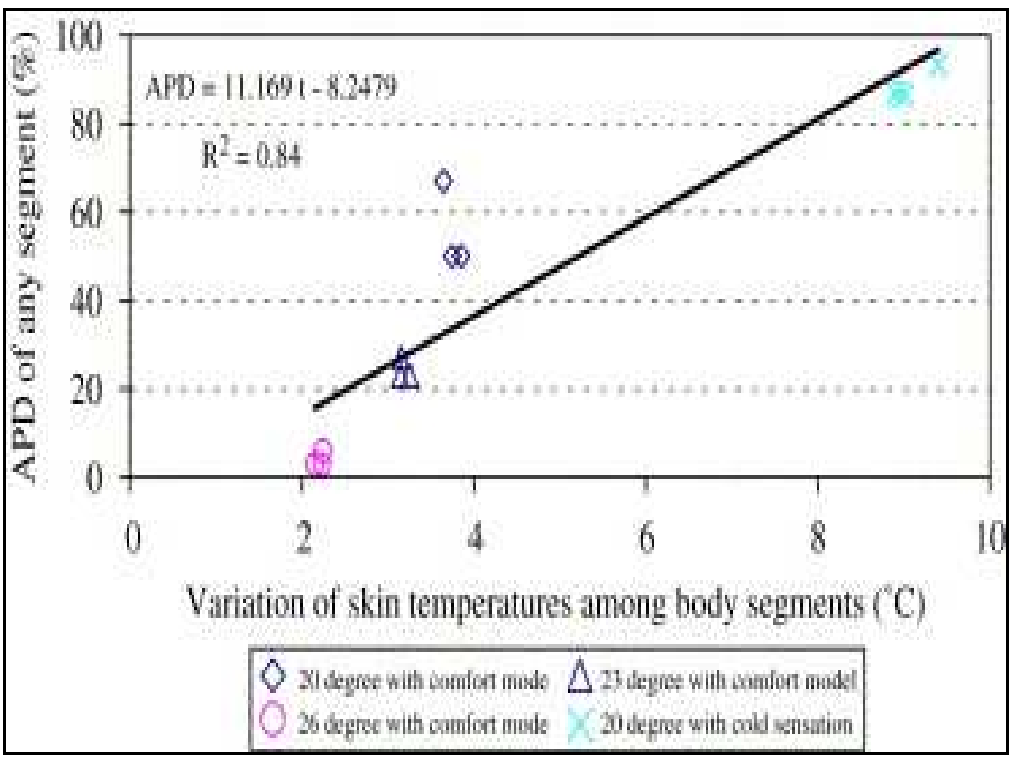


Fig. 5.1.13. Values of APD versus variations of skin temperatures

5.2. Manikin investigation with ventilation

This investigation made by my self in (TUB) Laboratory, 2007, B'NHIDI. L [57], [58], [59].[60], [61], [62], [63], and [64]. The thermal comfort investigation using a thermal manikin in a field environment chamber served by a Displacement Ventilation system.

The manikin has a male body with 18 individually heated and controlled body segments. The manikin together with subjects was exposed to 3 levels of vertical air temperature gradients, 0.50, 1 and 1.5 m height at one room air temperatures of 28, 30 and 33 °C at Relative humidity were maintained were 50% and the air velocity were measured three different levels. In a space served by Displacement Ventilation system (DV) air temperature, air velocity, turbulence level. In a thermally uniform condition, overall thermal neutrality is not always sufficient to provide thermal comfort. The manikin was placed at one locations in a room served by both Mixing Ventilation (MV) and DV systems.

Since skin surface temperature is closely related to the overall and local thermal sensations and comfort, the aims of this study are as follows.

(1) To investigate the distribution of skin surface temperature and sensible heat loss of a manikin in the space served by DV system for different temperature gradients at different room temperatures at 0.5, 1 and 1.5 m height.

(2) To correlate the subjective responses with the measurements from the thermal manikin.

5.2.1. Methods and Facilities

This study was conducted in a Thermal comfort Chamber (TCC) at the Technical University of Budapest between May and September 2007. Fig 5.2.1 showing the layout of field environment chamber and The chamber, 4 m (L) × 3.5 m (W) ×2.20 m (H), has an east-facing wall comprising of large wood which are insulated with blinds internally to reduce heat conduction and solar radiation. The chamber is equipped with an Air-Conditioning and Mechanical Ventilation (ACMV) system that is capable of switching between DV and MV modes.

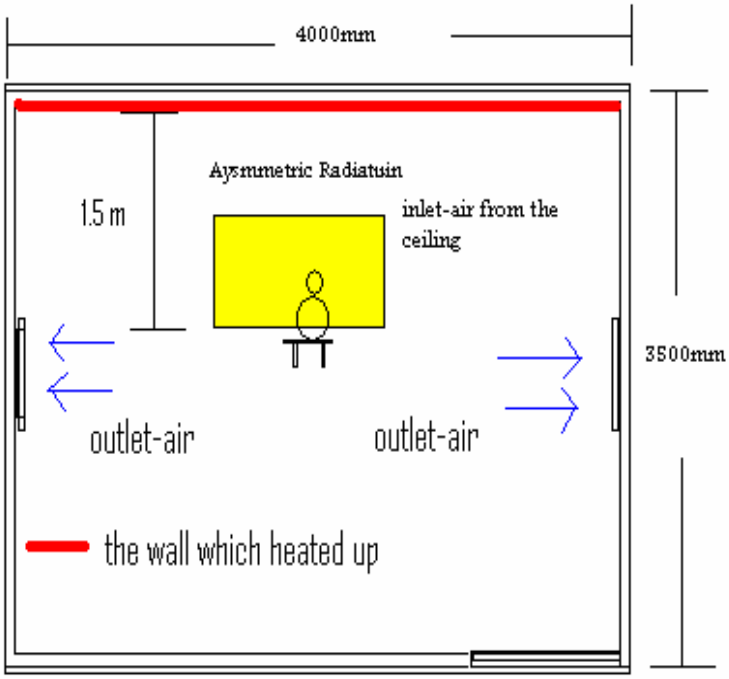


Fig 5.2.1. Layout of field environment chamber

The manikin used in the experiment is an average-sized made with 1.68 m standing height as shown in Fig.5.2.1

It is divided into 18 thermal segments that can be independently controlled and measured.

Table 5.2.1 shows the 18 body segments and their respective surface areas.

The manikin is controlled by software that has four control modes, namely,

(1) Only measuring for summer season,

(2) Heat loss of manikin without ventilation,

(3) Heat loss of manikin with ventilation,

Table 5.2.1 Body segments and respective areas of the manikin

No	Name of body segments	Area (cm^2)
1	Face	657
2	Chest	1830
3	Back	2404
4	LU Arm	877
5	RU Arm	877
6	LL Arm	560
7	RL Arm	560
8	L Hand	514
9	R Hand	514
10	LF Thigh	1185
11	RF Thigh	1185
12	LL Leg	1376
13	RL Leg	1376
14	L Foot	620
15	R Foot	620
16	LB Thigh	1621
17	RB Thigh	1621
18	Scalp	610

5.2.2. Heat loss and body parts temperatures.

Heat loss and body parts temperatures With difference wall surface temperature and difference air room temperature of one warm wall that surface wall front of the manikin with using ventilation

system . There are many cases to measuring and get the results the first case the measuring at 35 ^{o}C Surface temperature with difference air room temperature at 33, 30 and 28 with air ventilation and measuring difference air velocity from inlet air, 1.5 m height and 0.5 m height. the air velocity measuring near to the manikin Body about 10 cm with difference fan speed as shown in the table 5.2.3 and the heat loss and body parts temperature with wall surface temperature 35 ^{o}C and difference air room temperature, 33, 30 and 28 ^{o}C shown in the table 5.2.2

Table 5.2.2 Heat loss and body parts temperature with wall surface temperature 35 C^{o} and air room temperature 33, 30, and 28

body parts name	Heat loss (W/m^2) tr =33 C^o	Heat loss (W/m^2) tr = 30 C^o	Heat loss (W/m^2) t =28 C^o
Face	2.36	22.56	41.39
Chest	0.37	12.03	21.80
Back	3.21	10.43	15.37
LU Arm	6.45	18.10	28.68
RU Arm	5.56	16.72	27.47
LL Arm	6.32	31.37	50.72
RL Arm	5.85	32.08	51.15
L Hand	1.25	27.14	54.72
R Hand	0.21	23.97	49.34
LF Thigh	0.03	7.906	16.02
RF Thigh	0.03	7.845	16.18
LL Leg	11.5	27.20	37.97
RL Leg	11.0	27.41	38.74
L Foot	8.50	22.12	31.34
R Foot	10.8	26.91	36.35
LB Thigh	16.9	23.32	27.67
RB Thigh	12.1	19.69	24.44
Scalp	5.15	29.14	50.11
Total	107.	386.0	619.5

Table 5.2.3 Air velocity with difference air speed and height levels for 35 C^o wall surface temperatures 33, 30 and 28 C^o air room temperature

Height level measuring	max air speed velocity	Med air speed velocity	min air speed velocity
	Air room temperature = 33 ^oC		
Inlet velocity	0.7	0.5	0.2
1.5 m velocity	0.2	0.03	0.1
0.5 m velocity	0.1	0.06	0.04
	Air room temperature = 30 ^oC		
Inlet velocity	0.5	0.4	0.3
1.5 m velocity	0.3	0.2	0.1
0.5 m velocity	0.07	0.06	0.04

In the fig 5.2.3 shown the heat loss of manikin body at 35 ^oC with difference air room temperature 33, 30 and 28 ^oC and the the results are different when the air room temperature near to the Surface temperature the heat loss is decrees that mean the manikin body not comfortable and the PMV about (+3).

This situation could be found in Hot environment as Libya in summer season because the outside temperature in summer season about 38 - 40 ^oC and the monthly maximum temperature in Libya shown in chapter 2 in the fig 2.1 and in the chapter 4 in the fig. 4.2.1 and 4.2.2. Shown The Building construction in Libya without insulation and when the outside temperature more than 40 ^oC the inside surface temperature about 38 ^oC and this investigation for that situation in Libya to solve many problems and to get best comfortable room.

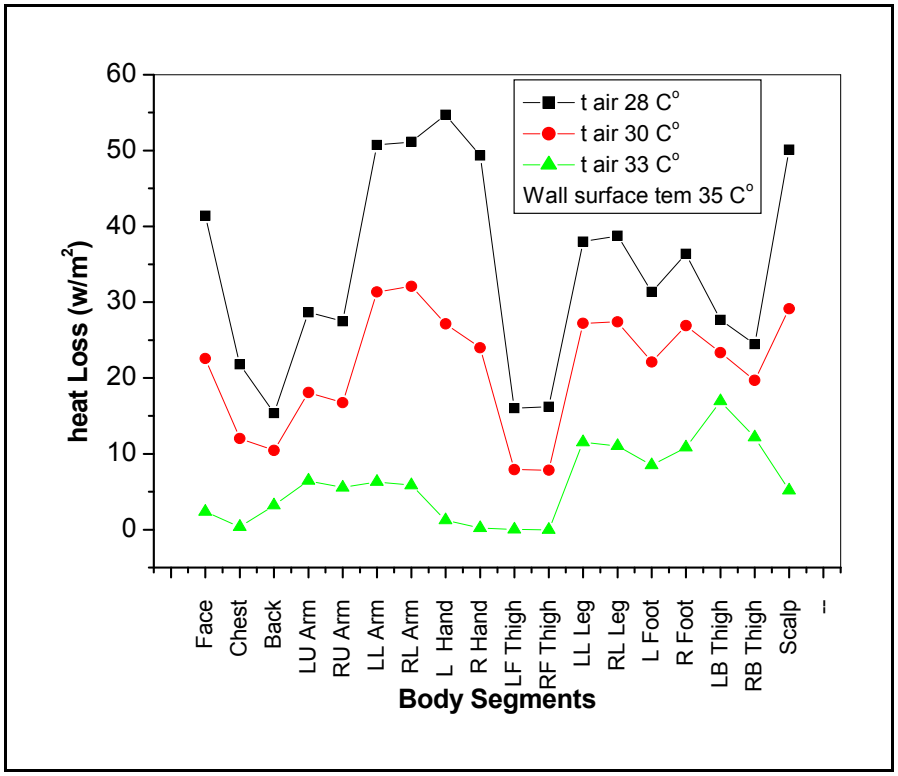


Fig 5.2.3 Heat loss with wall surface temperature 35 C° and difference air room temperature 33, 30 and 28 C°

In the fig 5.2.4 shown body parts temperature with difference surface temperature at 33, 30 and 28 °C in this case as you the fig the body parts temperature changing depends the surface temperature at the highest body parts temperature at Surface 33 °C and the PMV in this situation will be (+3) and this is uncomfortable room. This situation as Libya in the summer season.

Table 5.2.4. Hat loss and body parts temperature with surface temperature 33 °C and difference air room temperature, 30, 28 and 26 °C

body parts name	Heat loss (W/m^2) tr =30 C^o	Heat loss (W/m^2) tr = 28 C^o	Heat loss (W/m^2) t =26 C^o
Face	33.011028	47.377386	57.975236
Chest	34.002923	30.780206	30.375567
Back	34.00856	20.924082	23.1928
LU Arm	33.009529	35.642326	37.662655
RU Arm	33.007879	30.1332	37.887111
LL Arm	32.011572	46.785384	66.438047
RL Arm	32.020183	46.450156	67.828985
L Hand	31.010888	49.414064	67.078573
R Hand	31.009996	43.068173	59.912784
LF Thigh	33.012733	33.216777	24.525267
RF Thigh	33.008091	32.931159	24.434251
LL Leg	32.022454	31.818417	50.275904
RL Leg	32.024439	34.689544	50.424102
L Foot	31.017049	37.325045	46.345276
R Foot	31.020962	35.668938	46.450413
LB Thigh	33.023253	30.057849	35.133807
RB Thigh	33.023881	21.012691	32.645902
Scalp	33.017425	50.486324	69.695078
Total	584.2628	657.7817	828.2818

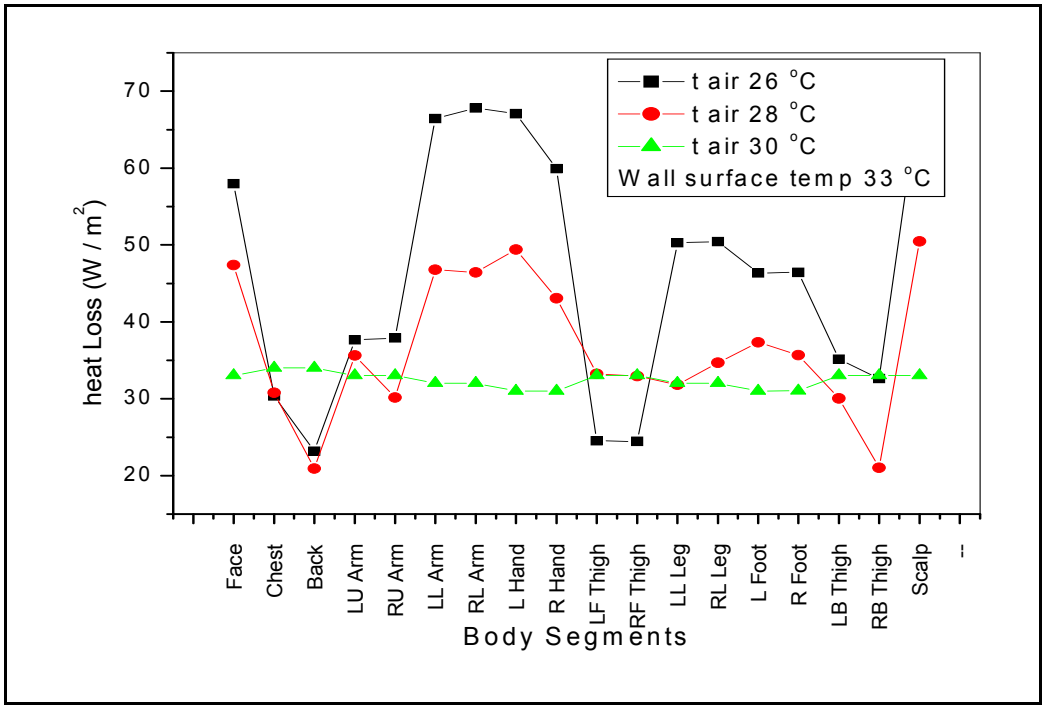


Fig 5.2.4 Heat loss with difference air room temperature

Table 5.2.5. Heat loss and body parts temperature with wall surface temperature 30 ^{o}C and difference air room temperature, 28 and 26 ^{o}C

body parts name	Heat loss (W/m^2) tr =28 C^o	body parts temperature tr = 26 C^o	Heat loss (W/m^2) tr = 26 C^o
Face	40.210912	32.981558	48.245127
Chest	22.494064	33.95085	25.139
Back	19.653833	33.969131	16.902563
LU Arm	27.964609	32.981781	32.278399
RU Arm	25.193854	32.977582	30.485
LL Arm	47.122933	31.978349	55.111501
RL Arm	45.554803	31.942615	55.284535
L Hand	48.987582	30.984981	57.679882
R Hand	41.326663	30.980989	52.46877
LF Thigh	17.894069	33.0026	13.029048
RF Thigh	15.831352	32.987494	17.880247
LL Leg	36.969893	31.998563	41.995202
RL Leg	37.141663	32.010592	42.840866
L Foot	32.616071	30.99061	34.714047
R Foot	34.294127	30.995409	40.949783
LB Thigh	28.754973	33.075561	44.436397
RB Thigh	25.539164	32.991358	24.660868
Scalp	48.747968	32.990978	52.244084
Total	596.29853		686.34532

Table 5.2.6. Air velocity with difference air speed and height levels for 30 ^oC surface temperatures and 28 and 26 air room temperature

Height level measuring	max air speed velocity	Med air speed velocity	min air speed velocity
	Air room temperature = 28 ^oC		
Inlet velocity	0.4	0.8	0.04
1.5 m velocity	0.08	0.08	0.1
0.5 m velocity	0.02	0.05	0.07
	Air room temperature = 26 ^oC		
Inlet velocity	0.6	0.6	0.3
1.5 m velocity	0.1	0.1	0.1
0.5 m velocity	0.04	0.05	0.02

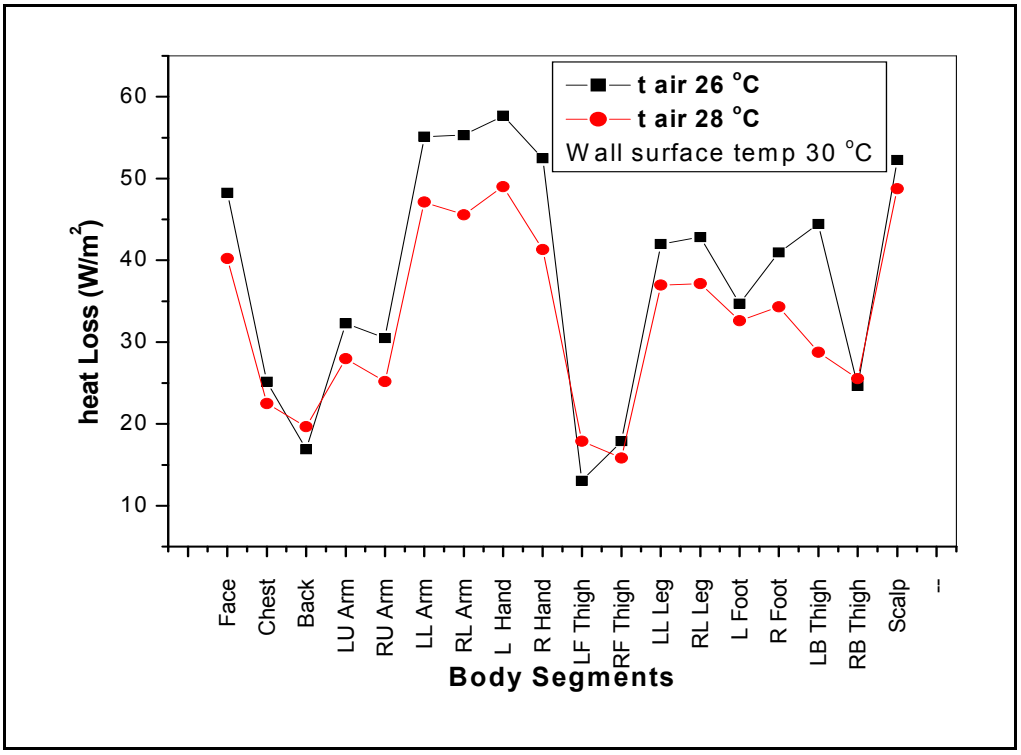


Fig 5.2.5 Heat loss with difference air room temperature

Table 5.2.2 shown the results of heat loss and body parts temperature of Manikin without using air movement in the chamber to find changing of manikin heat loss each points in the manikin.
heated up one wall was front of the manikin set on the table, the wall surface temperature was constant at 35 C^o and the air room temperature was 33, 30 and 28 C^o
the measuring was for difference air room temperature for example when the wall surface temperature was 35 C^o the air room temperature was 33 C^o, when the wall surface temperature was 35 C^o the air room temperature was 30 and when the wall surface temperature was 35 C^o the air room temperature was 28 C^o and I got the heat loss results each measurement and are showing in the table 5.2.2 and the figure 5.2.3.

5.2.3. The results were as following:

Face, LL arm, RL arm and L hand sensitized for difference air room temperatures,
when wall surface as constant at 35 C^o and the air room temperature was 28 C^o the heat loss was 34 W/m^2 and when the wall surface temperature was 35 C^o the heat loss of the face was 6 W/m^2 and the reason of big sensitive in the face because the face of the manikin exactly front of the warm wall. There are some parts of manikin no big sensitive for example LB and RB Thigh when the wall surface

temperature was 30 the heat loss of LB Thigh was 40 because the thigh was under the table which available in the chamber, arms , chest, hands, and these results near the same of felling people in the field study in Libya .

5.2.4. Summery of results of heat loss measurements with ventilation:

t_a – Air room temperature (C^o)
t_s – Wall surface temperature (C^o)

A. heat loss: when $t_a = 22\ C^o$ without effects
B. heat loss: when $t_a = 23\ C^o$ without effects
C. heat loss: when $t_a = 26\ C^o$ and $t_s = 30\ C^o$
D. heat loss: when $t_a = 26\ C^o$ and $t_s = 33\ C^o$
E. heat loss: when $t_a = 26\ C^o$ and $t_s = 35\ C^o$
F. heat loss: when $t_a = 28\ C^o$ and $t_s = 30\ C^o$
G. heat loss: when $t_a = 28\ C^o$ and $t_s = 33\ C^o$
H. heat loss: when $t_a = 28\ C^o$ and $t_s = 35\ C^o$
I . heat loss: when $t_a = 30\ C^o$ and $t_s = 33\ C^o$
J . heat loss: when $t_a = 30\ C^o$ and $t_s = 35\ C^o$

body parts	**A**	**B**	**C**	**D**	**E**	**F**	**G**	**H**	**I**	**J**

Table 5.2.7. total results of heat loss of Manikin with ventilation

name										
Face	**95.1**	**71.8**	**32.9**	**43.9**	**45.3**	**47.2**	**45.3**	**41.3**	**33.0**	**22.5**
Chest	30.3	20.9	33.9	39.3	43.7	36.4	30.7	21.8	34.0	12.0
Back	33.2	20.9	33.9	39.1	44.9	32.6	20.9	15.3	34.0	16.4
LU Arm	35.1	25.4	32.9	37.6	43.6	39.9	35.6	28.6	33.0	18.1
RU Arm	**31.5**	**22.2**	**32.9**	**37.8**	**30.1**	**33.1**	**30.1**	**27.4**	**33.0**	**16.7**
LL Arm	**39.3**	**28.1**	**31.5**	**38.4**	**43.7**	**47.1**	**43.7**	**39.7**	**32.0**	**31.3**
RL Arm	37.0	26.9	31.9	38.8	42.4	45.5	42.4	38.1	32.0	30.0
L Hand	**88.0**	**63.1**	**30.9**	**34.0**	**40.4**	**48.9**	**43.4**	**39.7**	**31.0**	**27.1**
R Hand	**79.1**	**55.1**	**36.9**	**38.9**	**44.0**	**48.3**	**44.0**	**39.3**	**31.0**	**23.9**
LF Thigh	44.4	30.3	33.0	37.5	42.2	29.8	21.2	16.0	33.0	22.9
RF Thigh	42.3	29.2	32.9	36.4	40.9	27.8	21.9	16.1	33.0	21.8
LL Leg	**77.5**	**59.1**	**31.9**	**37.2**	**42.8**	**47.9**	**42.8**	**37.9**	**32.0**	**27.2**
RL Leg	**72.8**	**55.2**	**32.0**	**37.4**	**44.6**	**48.1**	**44.6**	**38.7**	**32.0**	**27.4**
L Foot	68.3	50.9	30.2	34.3	43.3	39.6	35.3	31.3	31.0	22.1
R Foot	67.0	50.3	30.9	34.4	40.6	44.2	40.6	36.3	31.0	26.9
LB Thigh	**46.3**	**35.9**	**33.0**	**36.1**	**42.0**	**37.7**	**31.0**	**27.6**	**33.0**	**23.3**
RB Thigh	**41.7**	**31.7**	**32.9**	**36.6**	**42.0**	**37.5**	**31.0**	**24.4**	**33.0**	**19.6**
Scalp	95.1	66.5	32.9	39.6	44.4	48.7	44.4	38.1	33.0	29.1

5.3. Manikin investigation without ventilation.

The main findings heat loss measuring with difference surface temperatures 35, 33 and 30 and difference air room temperature 32, 30, and 28 . The local discomfort was affected by overall thermal sensation and was lower at overall thermally neutral state. The table 5.3.1 shown heat loss and body parts temperature with using measuring without any ventilation at surface temperature 35, 33 and 30.

In fig 5.3.1. Shown the heat loss measuring results and air room temperature at t surface 30 ^{o}C , air room temperature 28 ^{o}C, at t surface 33, air room temperature 30 ^{o}C and at t surface 35 ^{o}C, air room temperature 32. The heat loss increase at surface temperature 30 ^{o}C Because there is difference heat between the air room temperature and Body manikin, in this case the heat loss is high and the fig 5.2.2. Shown changing of Body temperature in the Surface temperature 35, 33 and 30 ^{o}C

Table 5.3.1. Heat loss and body parts temperature of Manikin with difference surface temperatures and air room temperature.

body parts name	body parts temperature ts =30 C^o tr =26 C^o	Heat loss (W/m^2) ts =30 C^o tr =26 C^o	Heat loss (W/m^2) ts = 33 C^o tr = 28 C^o	Heat loss (W/m^2) ts =35 C^o tr =30 C^o
Face	33.00	33.83	18.63	6.48
Chest	34.00	17.42	8.861	2.12
Back	34.00	16.75	11.01	7.01
LU Arm	33.00	16.95	11.52	6.12
RU Arm	33.00	16.87	10.57	4.50
LL Arm	32.00	19.69	11.77	4.83
RL Arm	32.01	22.36	13.51	5.96
L Hand	31.00	22.66	9.88	0.18
R Hand	42.86	2.086	1.77	0.10
LF Thigh	33.00	13.12	5.85	0.99
RF Thigh	33.008	12.07	5.62	0.78
LL Leg	32.01	35.83	27.51	21.5
RL Leg	32.01	34.03	26.07	20.0
L Foot	31.01	37.31	28.75	21.7
R Foot	31.01	36.75	28.74	22.3
LB Thigh	33.01	41.55	36.75	33.4
RB Thigh	33.01	25.61	20.80	17.3
Scalp	33.00	29.57	17.98	10.4
Total		434.5	295.6	186.0463

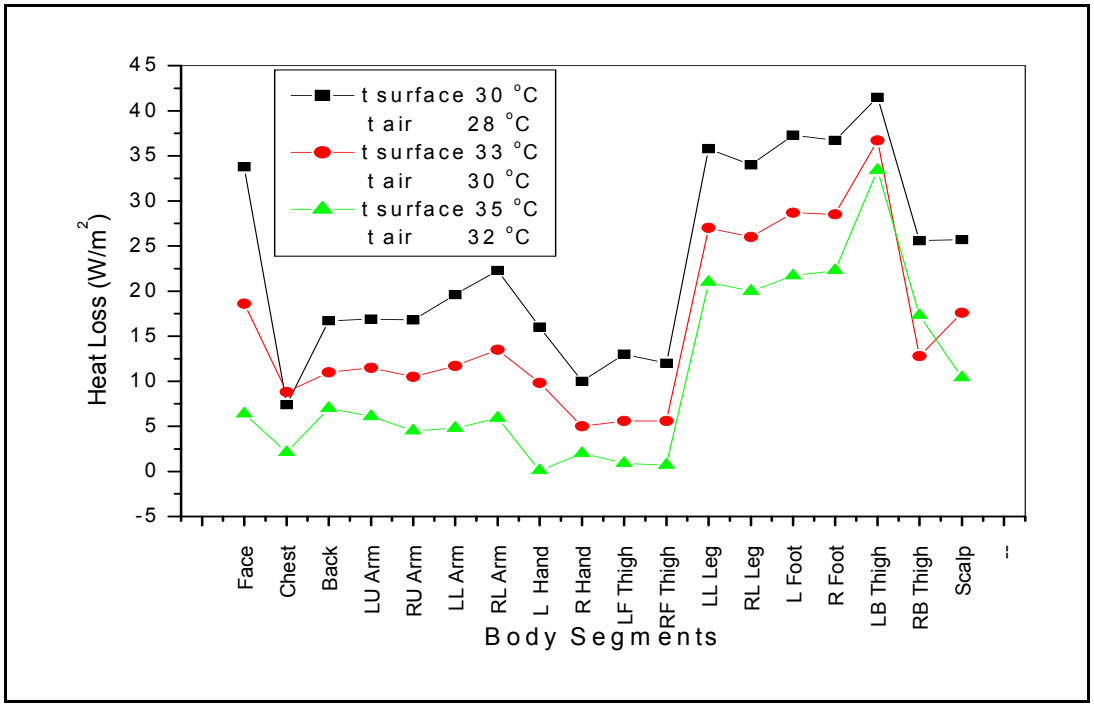


Fig 5.3.1 Heat loss with difference surface temperature and air room temperature without ventilation

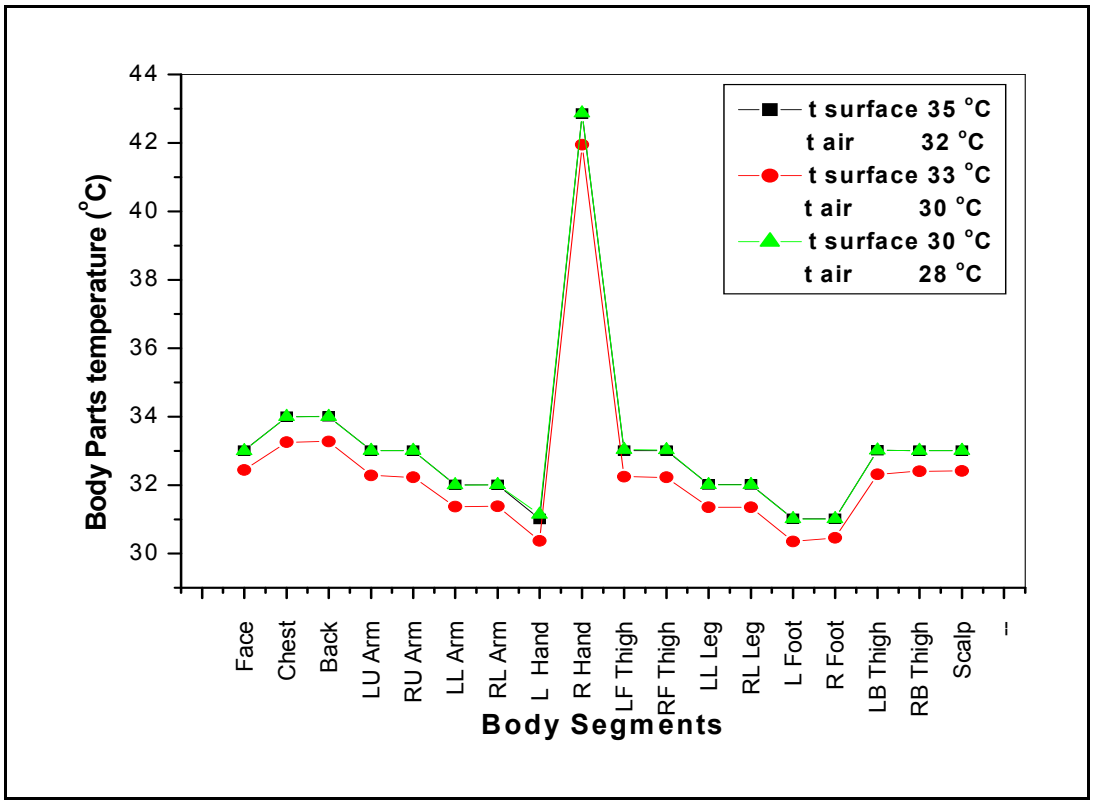


Fig 5.3.2 Body parts temperature with difference surface temperature and air room temperature

Table 5.3.1 shown the results of heat loss and body parts temperature of Manikin without using any air movement in the chamber to find how the change of manikin heat loss each points in the manikin because there are 80 % of buildings in Libya without ventilation until now it the same situation in my measurements in the chamber. the measurements was in the Laboratory of technical University of Budapest at 9:00 pm. The chamber of measurements shown in fig.5.2.1

I first step I heated up on wall the wall was front of the manikin set on the table, the heating was including difference wall surface temperature 30, 33 and 35 C^o**.**
each one wall surface temperature I measured difference air room temperature for example when the wall surface temperature was 30 C^o the air room temperature was 26 C^o, when the wall surface temperature was 33 C^o the air room temperature was 28 and when the wall surface temperature was 35 C^o the air room temperature was 30 C^o and I got the heat loss results each measurement and are showing in the table 5.3.1 and the figure 5.3.1.

5.3.1. The results were as following:

The face was sensitive for difference wall surface temperature when the temperature of wall was 30 C^o the heat loss was 34 W/m^2 and when the wall surface temperature was 35 C^o the heat loss of the face was 6 W/m^2 and the reason of big sensitive in the face because the face of the manikin exactly front of the warm wall.
There are some parts of manikin no big sensitive for example LB and RB Thigh when the wall surface temperature was 30 the heat loss of LB Thigh was 40 because the thigh was under the table which available in the chamber , arms , chest, hands, and these results near the same of felling people in the field study in Libya .

5.3.2. Summery of results of heat loss measurements without ventilation:

t_a – Air room temperature (C^o)
t_s – Wall surface temperature (C^o)

A. heat loss: when t_a = 22 C^o without any effects
B. heat loss: when t_a = 23 C^o without any effects
C. heat loss: when t_a = 26 C^o and t_s = 30 C^o
D. heat loss: when t_a = 26 C^o and t_s = 33 C^o
E. heat loss: when t_a = 26 C^o and t_s = 35 C^o
F. heat loss: when t_a = 28 C^o and t_s = 30 C^o
G. heat loss: when t_a = 28 C^o and t_s = 33 C^o
H. heat loss: when t_a = 28 C^o and t_s = 35 C^o
I . heat loss: when t_a = 30 C^o and t_s = 33 C^o
J . heat loss: when t_a = 30 C^o and t_s = 35 C^o

Table 5.3.2. Total results of heat loss of Manikin without ventilation.

body parts name	A	B	C	D	E	F	G	H	I	J
Face	**95.1**	**71.8**	**33.8**	**18.6**	**12.2**	**38.1**	**32.4**	**28.8**	**10.6**	**6.4**
Chest	30.3	20.9	17.4	8.8	5.1	36.3	33.2	29.4	9.8	2.1
Back	33.2	20.9	16.7	11.0	7.3	35.2	33.2	29.7	11.0	7.0
LU Arm	35.1	25.4	16.9	11.5	6.5	34.1	32.2	28.9	11.5	6.1
RU Arm	**31.5**	**22.2**	**16.8**	**10.5**	**5.4**	**35.5**	**32.2**	**27.8**	**10.5**	**4.5**
LL Arm	**39.3**	**28.1**	**19.6**	**11.7**	**6.1**	**35.3**	**31.3**	**28.6**	**11.7**	**4.8**
RL Arm	37.0	26.9	22.3	13.5	10	35.0	31.3	28.3	13.5	5.9
L Hand	**88.0**	**63.1**	**22.6**	**15.8**	**10.2**	**33.0**	**30.3**	**27.6**	**2.8**	**0.1**
R Hand	**79.1**	**55.1**	**22.0**	**14.7**	**9.5**	**33.1**	**30.9**	**32.0**	**2.7**	**0.1**
LF Thigh	44.4	30.3	13.1	9.8	5.3	35.4	32.2	27.1	7.8	0.9
RF Thigh	42.3	29.2	12.0	8.6	4.6	36.3	32.2	28.0	8.6	0.7
LL Leg	**77.5**	**59.1**	**35.8**	**27.5**	**21**	**34.5**	**31.3**	**27.8**	**27.5**	**21.5**
RL Leg	**72.8**	**55.2**	**34.0**	**26.0**	**19**	**34.8**	**31.3**	**27.0**	**26.0**	**20.0**
L Foot	68.3	50.9	37.3	27.7	21	33.3	30.3	26.3	27.7	21.7
R Foot	67.0	50.3	36.7	28.7	22	33.0	30.4	26.7	28.7	22.3
LB Thigh	**46.3**	**35.9**	**41.5**	**36.7**	**26**	**32.3**	**32.3**	**28.5**	**36.7**	**33.4**
RB Thigh	**41.7**	**31.7**	**25.6**	**20.8**	**14**	**34.7**	**32.4**	**28.6**	**20.8**	**17.3**
Scalp	95.1	66.5	29.5	23.9	16	34.1	32.4	29.3	19.9	10.4

5.4. Comparison of important body parts for all heat loss measurements

Table 5.3.3. Total results of heat loss of Manikin without ventilation and with ventilation

temperatures C^o	Ventilation situation	Heat loss W/m^2						
		Face	RU Arm	LL Arm	L Hand	R Hand	LL Leg	RL Leg
$t_r = 22\ C^o$	Without any effects	95.1	31.5	39.3	88.0	79.1	77.5	72.8
$t_r = 23\ C^o$	Without any effects	71.8	22.2	28.1	63.1	55.1	59.1	55.2
$t_r = 26\ C^o$ $t_s = 30\ C^o$	Without ventilation	32.8	16.8	19.6	22.6	22.0	35.8	32.0
$t_r = 26\ C^o$ $t_s = 30\ C^o$	ventilation	36.9	32.9	31.5	30.9	36.9	31.9	34.0
$t_r = 26\ C^o$ $t_s = 33\ C^o$	Without ventilation	18.6	10.5	11.7	15.8	14.7	27.5	26.0
$t_r = 26\ C^o$ $t_s = 33\ C^o$	ventilation	43.9	37.8	38.4	34.0	38.9	37.2	37.4
$t_r = 26\ C^o$ $t_s = 35\ C^o$	Without ventilation	12.2	11.4	11.1	10.2	10.5	21.0	19.0
$t_r = 26\ C^o$ $t_s = 35\ C^o$	ventilation	45.3	30.1	43.7	40.0	44.0	42.8	44.6
$t_r = 28\ C^o$ $t_s = 30\ C^o$	Without ventilation	34.1	29.5	35.3	33.0	33.1	34.5	34.8
$t_r = 28\ C^o$ $t_s = 30\ C^o$	ventilation	47.2	33.1	47.1	48.9	48.3	47.9	48.1
$t_r = 28\ C^o$ $t_s = 33\ C^o$	Without ventilation	32.4	27.2	31.3	30.3	30.9	31.3	31.2
$t_r = 28\ C^o$ $t_s = 33\ C^o$	ventilation	45.3	32.1	34.7	43.4	44.0	42.8	44.6
$t_r = 28\ C^o$ $t_s = 35\ C^o$	Without ventilation	28.8	22.8	28.6	27.6	32.0	27.8	27.0
$t_r = 28\ C^o$ $t_s = 35\ C^o$	ventilation	41.3	27.4	39.4	39.7	39.3	37.9	38.7
$t_r = 30\ C^o$ $t_s = 33\ C^o$	Without ventilation	10.6	10.5	11.7	09.81	11.71	27.5	26.0
$t_r = 30\ C^o$ $t_s = 33\ C^o$	ventilation	33.0	33.0	32.0	31.0	31.0	32.0	32.0
$t_r = 30\ C^o$ $t_s = 35\ C^o$	Without ventilation	8.40	6.50	6.80	4.11	5.11	21.5	20.0
$t_r = 30\ C^o$ $t_s = 35\ C^o$	ventilation	22.5	16.7	31.3	27.1	23.9	27.2	27.4

CONCLUSION

1- Building standards have been based on fixed comfort temperatures found from tests held in climatic chambers. Those standards assume that the indoor temperature is fixed to a set value and controlled by heating and air conditioning systems.

2- In Libya and all hot environment countries, the heating and air conditioning systems, in case they exist, are not used continuously. Thus, the indoor temperature is fluctuating. The thermal sensation of the building occupants is the only controller of the ventilation, the heating or the cooling of the building.

3- Unlike the conventional thermal regulations, which are based on energy consumption, the special feature of the future Libya thermal regulation is related to the fact that it must ensure a minimum level of thermal comfort when the building is free running without any heating or cooling system.

4- The comfort temperature can be correlated to the monthly mean outdoor temperature. Such concept can be used to design comfortable buildings.

5- The comparison between the comfort temperature and the maximum and minimum outdoors temperatures can help designer to judge whether passive-heating and cooling techniques are appropriate for the climate under consideration.

6- The relationship between the indoor comfort temperature and the range of outdoors temperatures shows whether for example night ventilation can be or not a way to keep the building cool in summer and help the designer to select the appropriate thermal capacity for the building.

7- Fundamental difference between the groups is the activity level, the increase of which pushes the ranges nearer to the origin.

8- Temperature boundaries of the ranges drawn up for summer season look being appropriate related to the practical experiences in spite of the fact that in the time of the development of Fanger's PMV equation building service engineering was limited mainly to the heating and cooling had a smaller importance.

Field study in Libya

1- I carried out in-situ measurements in 51 buildings and 237 subjects. Using their results I determined the internal physical parameters of traditional Libyan buildings and the occupants' subjective thermal sensation. These data will provide an important starting point for:

- the increasingly extensive modernization of traditional buildings (e.g. local cooling, energy use) in terms of the relevant designing parameters,

- the impact of physical parameters influencing the anticipated subjective thermal sensation of users and occupants.

2- Through the measurements I conducted in Libya I proved that the PMV method did not give us the appropriate values – instead AMV should be defined .(see fig below)

3- I compared the data recorded in the previous section with the data of people using Libyan apartments and houses built with a modern technology and equipped with local cooling. This is the first time such quantitative and subjective comparison has been made in Libya, considerably facilitating the work of planners.
All these promote the definition of minimal thermal comfort parameters in Libya.

4- I prepared a new type of questionnaire for the in-situ measurements confirming the above thesis, allowing for the comparison of the two evaluation method.

5- The overall feeling of the occupants in Libya in the summer seasons, reported that they are more satisfied and thermally neutral in old naturally ventilated buildings than in new air-conditioned buildings. In the old buildings, about 54% of the occupants are feeling neutral and 8% are feeling hot, compared to only 15% of the occupants feeling neutral and 33% feeling hot in the new air-conditioned buildings.

In Libya and all hot environment countries, the heating and air conditioning systems, in case they exist, are not used continuously. Thus, the indoor temperature is fluctuating. The thermal sensation of the building occupants is the only controller of the ventilation, the heating or the cooling of the building.

Unlike the conventional thermal regulations, which are based on energy consumption, the special feature of the future Libya thermal regulation is related to the fact that it must ensure a minimum level of thermal comfort when the building is free running without any heating or cooling system.

6- Full-scale measurements have been carried out involving environmental parameters and human thermal comfort responses from 51 buildings in Gharian, Libya. These buildings are either naturally ventilated with courtyards or mechanically ventilated with air-conditioning systems. This field survey was conducted in the summer of 2004, representing a typical hot–dry climate in hot environment. The following conclusions can be drawn:

7- The PMV model in the form of CR 1552 and ISO 7730 can not be used, without modifications, for predicting the overall thermal comfort of the occupants in old naturally ventilated buildings. However, the ISO 7730 standard can be used to measure human thermal comfort in new air-conditioned buildings without modifications.

8- The overall feeling of the occupants in Libya in the summer seasons, reported that they are more satisfied and thermally neutral in old naturally ventilated buildings than in new air-conditioned buildings. In the old buildings, about 54% of the occupants are feeling neutral and 8% are feeling hot, compared to only 15% of the occupants feeling neutral and 33% feeling hot in the new air-conditioned buildings.

9- These modifications would have to address issues related to adaptive effects.

10- In Libya and all hot environment countries, the heating and air conditioning systems, in case they exist, are not used continuously. Thus, the indoor temperature is fluctuating. The thermal sensation of the building occupants is the only controller of the ventilation, the heating or the cooling of the building.

11- Unlike the conventional thermal regulations, which are based on energy consumption, the special feature of the future Libya thermal regulation is related to the fact that it must ensure a minimum level of thermal comfort when the building is free running without any heating or cooling system.

Manikin and subjects investigation with

1- By laboratory measurements were used to investigate the impact of the most frequent discomfort factor (asymmetrical radiation of warm walls) with a thermal manikin. I determined the temperatures of the various body parts, their heat intake and heat transmission in static air in office spaces where there is a combination of 26, 28 and 30°C air temperatures and a warm wall of 33° and 35C.

2- Using the poor subjective values obtained during the in-situ measurements and the laboratory data I investigated the heat transmission values for the various body parts in a vertical air flow.

3- I compared the experiments conducted at the Singapore University with manikins and live subjects to the findings of measurements I carried out in the measuring room of TUB. I chose this solution because at TUB it was not possible to study subjects adapted to a hot climate.

I found that the air inflow was a better solution in terms of the heat transmission of the human body (in particular the head and the upper part of the body) if there is asymmetrical radiation near a warm wall.

The results of the measurements I conducted for various air temperatures and warm walls can be well compared with the Singaporean findings and are complementary.

4- Thermal manikin and subjects were used in this comfort study at the Field Environment Chamber served by DV system.

5- The distribution of overall average and different body segments' surface temperatures was investigated under different room temperatures and temperature gradients. Subjective responses were correlated with overall average surface temperatures and the variations of skin surface temperature among body segments.

6- The results show that overall average surface temperatures for gradients of 1, 3 and 3 K/m at a certain room air temperature were almost the same and the skin surface temperatures increased with the increase of room air temperature.

7- The variations of skin surface temperature among body segments for gradients of 1, 3 and 1 K/m at a certain room air temperature were almost the same but the variations decreased with the increase of room temperature. Moreover, the results also show that the variations at cold and cool thermal sensations were much greater than at overall thermally neutral sensation. By correlating the subjective

responses with overall average surface temperatures and the variations of skin surface temperatures among body segments, it was found that actual values of OTS increased with the increase of overall average skin surface temperature. Higher variation of skin surface temperature among body segments may lead to higher DR and percentage of subjects feeling uncomfortable.

8- In summary, the findings indicate that room air temperature had greater influence on overall and local thermal sensations and comfort than temperature gradient.

M1. Nomenclature

a	Width of a rectangular surface.	[m]
A_{Du}	DuBois body surface area. The total surface area of a naked person as estimated by the DuBois formula.	[m^2]
A_i	Area of plane surface.	[m^2]
A_r	Effective radiant area of a body. Surface that exchanges radiant energy with the environment through a solid angle of 4^1. This is smaller than the actual surface area of the body because the body is not a convex surface.	[m^2]
b	Length of a rectangular surface.	[m]
c	Distance between the two surfaces.	[m]
C_{res}	Respiratory convective heat exchange.	[W/m^2]
D	Diameter of globe transducer.	[m]
DR	Draught Rate. The percentage of people dissatisfied due to draught.	[%]
E	Evaporative heat exchange at the skin.	[W/m^2]
E_c	Evaporative heat exchange at the skin, when the person experiences a sensation of thermal neutrality.	[W/m^2]
E_{res}	Respiratory evaporative heat exchange.	[W/m^2]
E_{sw}	Evaporative heat loss from evaporation of sweat.	[W/m^2]
ET*	Effective temperature (new effective temperature)	[°C]
f_{cl}	Clothing area factor. The ratio of the surface area of the clothed body to the surface area of the naked body.	
F_{p-i}	Angle factor between the person and surface i . Defined as the fraction of diffuse radiant energy leaving the body surface which falls directly upon surface i	
F_{pl-i}	Angle factor between a small plane and surface i . Defined as the fraction of diffuse radiant energy leaving the small plane surface which falls directly upon surface i	
h_c	Convective heat transfer coefficient.	[$W/m^2/°C$]
$h_{c,eq}$	Convective heat transfer coefficient when air velocity in enclosure is zero.	[$W/m^2/°C$]
h_{cg}	Convective heat transfer coefficient for a globe (ellipsoid).	[$W/m^2/°C$]
h_r	Radiative heat transfer coefficient.	[$W/m^2/°C$]
H	Dry Heat Loss. Heat loss from the body surface through convection,	[W/m^2]

	radiation and conduction.	
I_{cl}	Clothing insulation. It is an average including uncovered parts of the body.	[m^2°C/W]
I_{clu}	Garment insulation. Expressed as the overall increase in insulation attributable to the garment.	[m^2°C/W]
K_{cl}	Conductive heat flow through clothing.	[W/m^2]
M	Metabolic rate. The rate of transformation of chemical energy into heat and mechanical work by aerobic and anaerobic activities within the body.	[W/m^2]
p_a	Humidity. Partial water vapour pressure in the air.	[Pa]
p'_a	Humidity in the imaginary room.	[Pa]
PMV	Predicted Mean Vote. The predicted mean vote of a group of people on the 7-point thermal sensation scale.	
PPD	Predicted Percentage of Dissatisfied. The predicted percentage of a group of people who are feeling too cold or too hot.	[%]
q	Heat exchange between body and surroundings.	[W/m^2]
q'	Heat exchange between body and surroundings in the imaginary room.	[W/m^2]
R	Radiative heat exchange.	[W/m^2]
R'	Radiative heat exchange in the imaginary room.	[W/m^2]
RH	Relative Humidity	[%]
SD	Standard Deviation of air velocity	[m/s]
t_a	Air Temperature	[°C]
t'_a	Air Temperature in imaginary room	[°C]
t_{co}	Comfort Temperature. The Equivalent Temperature at which a person experiences a sensation of thermal neutrality.	[°C]
t_{cl}	Clothing surface temperature.	[°C]
t_{eq}	Equivalent Temperature.	[°C]
t_g	Globe Temperature.	[°C]
t_i	Temperature of surface no. i.	[°C]
t_o	Operative Temperature.	[°C]
t_r	Mean Radiant Temperature	[°C]
t'_r	Mean Radiant Temperature in the imaginary room	[°C]
t_{pr}	Plane Radiant Temperature.	[°C]
Δt_{pr}	Radiant Temperature Asymmetry	[°C]
$\bar{t}_{sk}$	Mean skin temperature	[°C]
Tu	Turbulence Intensity.	[%]
v_a	Local Mean Air Velocity	[m/s]
v'_a	Local Mean Air Velocity in the imaginary room	[m/s]
v_{ar}	Relative Mean Air Velocity. The air velocity relative to the occupant, including body movements.	[m/s]
W	Effective mechanical power.	[W/m^2]
ε	Emission coefficient of the body surface expressed as a ratio of the black body emissivity.	
σ	Stefan-Boltzmann constant ($5.67 * 10^{-8}$)	[W/m^2/°C^4]
L	Dry respiration heat loss	[kcal/hr]
v_r	Relative air velocity	[m.s]

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APPENDIX

The published papers during the work on the Dissertation

Journal publication:

1. Adel Akair, Prof. L. Banhidi.: **2006**. Thermal Comfort problems in Hot Environment, Libya, Rehva Journal-Federation of European Heating and Air-Conditioning Associations.

2. A. Akair, Prof.L. Banhidi.:**2006**. Improving thermal comfort in hot environment such as Libya, WTA-International Journal for Technology and Application in Building Maintenance - ISSN 1612-0159. p.185-196.

3. Adel Akair, Prof. L. Banhidi.:**2007.** Investigation Study Results of Thermal Comfort, Sent to: journal of periodica Polytechnica .

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1. S-R.A. Adel – E.Stevensné Szádday – R. Goda. Instalatii pentru.:**2005.** Possibility to into account the joint impact of draught and asymmetrical radiation, In dimensioning thermal comfort in a hot environment, Constructii confortul ambiental, proceeding p. 204-211.

2. A. Akair, L. Banhidi.:**2006.** Improving thermal comfort problems in hot Environment 3rd Mediterranean Congress of HVAC Engineering, ISBN. 2-86834-123-3. Climamed, Lyon, France, p. 525-531.

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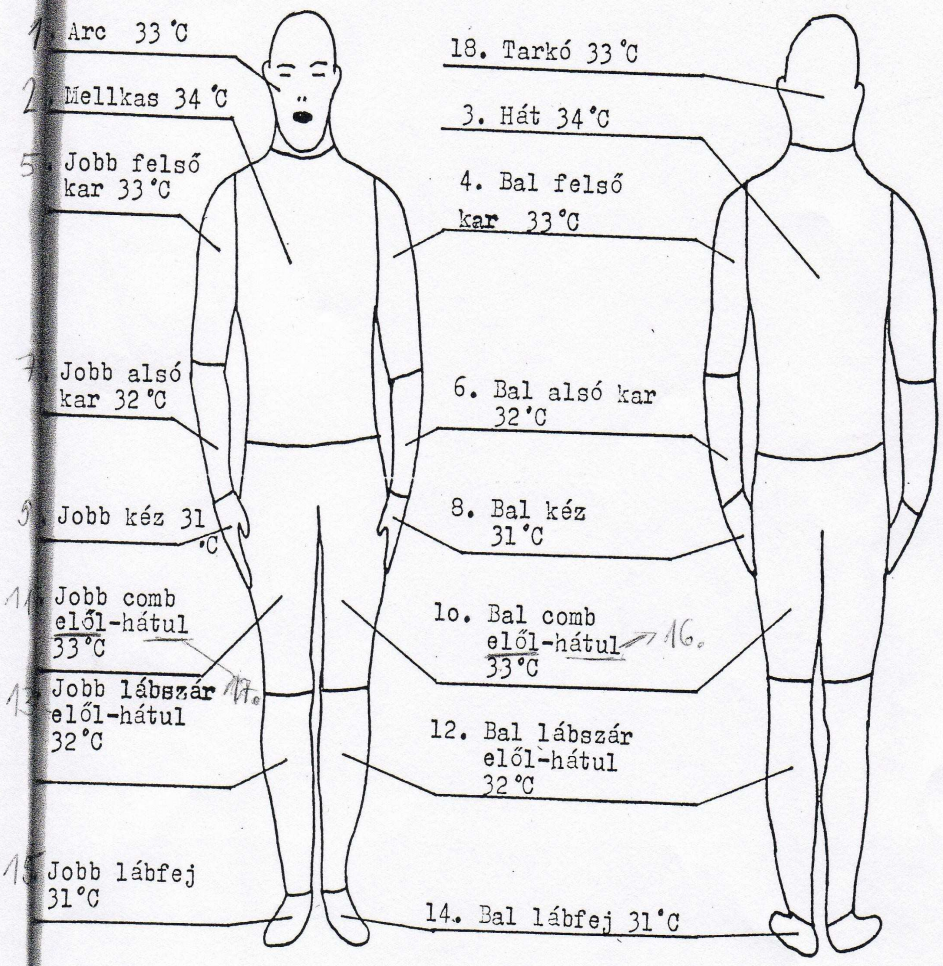
5. A. Akair, ILona Frohner.:**2006**. Comfort range as a mean of dimensioning in hot Environment 3rd Mediterranean Congress of HVAC Engineering, ISBN. 2-86834-123-3. Climamed, Lyon, France, p. 473-478.

6. Adel Akair.:**2007.** Solving the problems of thermal comfort in summer time Roomvent . International conference on Air Distribution in Rooms. Helsinki , Conference, Abstract, p. 86, ISBN 978-952-99898-0-5

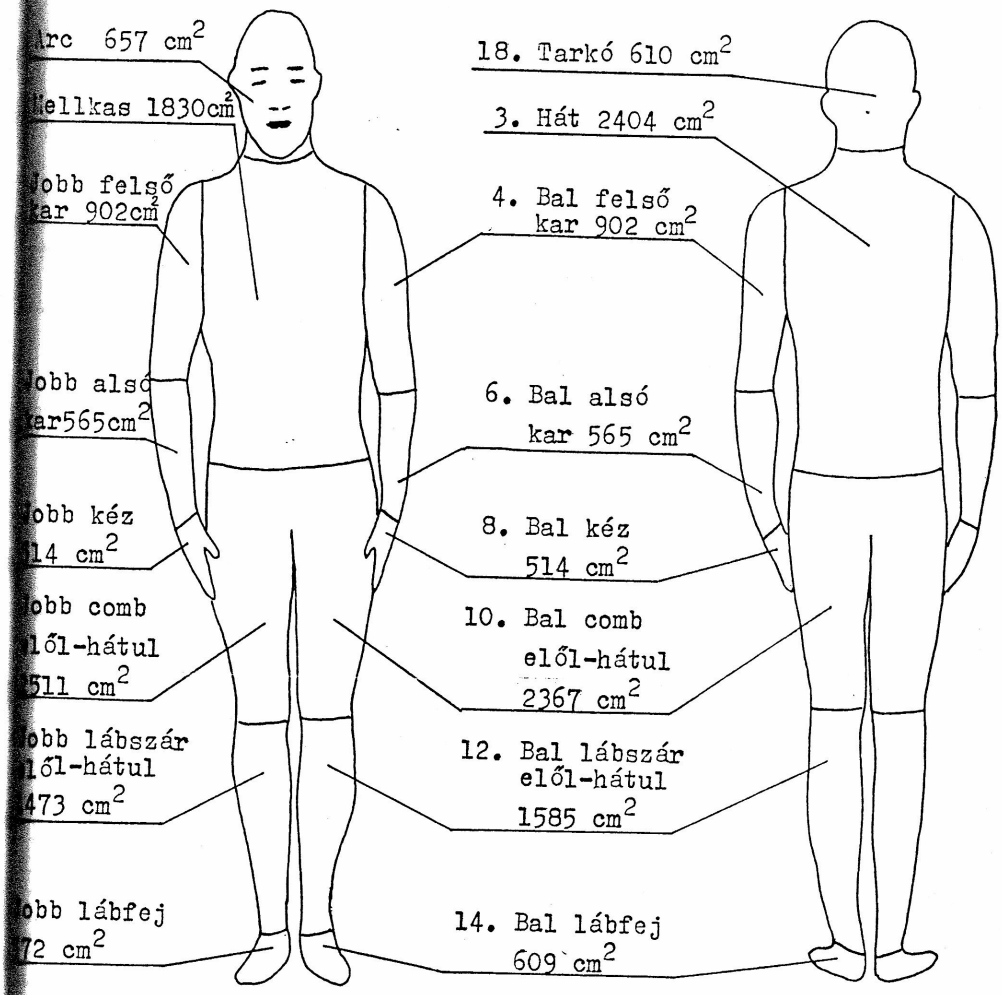
7. Adel Akair.:**2007.** Thermal comfort model of Libya context: field study results Climmed .Enegy, Climate and indoor comfort in Mediterranean countries Genova, Italy .ISBN 9788895620022 Copy by AICARR. p. 125-141.

A müember testrészeinek jele, megnevezése, hőmérséklete



28. ábra

Müember testrészeinek jele,megnevezése,felülete



ÁLLÓ MÜEMBER ÖSSZES TESTFELÜLETE : 18580 cm2

müembernél a testrészek jele, megnevezése, felülete 1 -9 és 18-as számig megegyezik az álló müemberével.

Térés:

10.	Bal comb elől	1422 cm2	12.	Bal lábszár elől-hátul	1466 cm2
16.	Bal comb hátul	1173 cm2			
11.	Jobb comb elöl	1422 cm2	13.	Jobb lábszár elől-hátu	1466 cm2
17.	Jobb comb hátul	1173 cm2			
15.	Jobb lábfej	628 cm2	14.	Bal lábfej	628 cm2

ÜLŐ MÜEMBER ÖSSZES TESTFELÜLETE : 18841 cm2

27. ábra

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Gödöllő, 2009. Adel Akair