UNIVERSITY OF NOVA GORICA
GRADUATE SCHOOL

INDIVIDUALIZATION OF PERSONAL SPACE IN HOSPITAL ENVIRONMENT

DISSERTATION

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SUMMARY

Hospital presents a complex indoor environment of various users, health hazards and specific activities. In hospital environment (HE) various users are present (i.e. patients, staff and visitors) with different demands and needs for thermal comfort that cannot be satisfied with current heating, cooling, ventilation and air–conditioning systems. Users are exposed to numerous health hazards that can be classified into physical, chemical, biological, biomechanical and psychosocial hazards. Specific activity is the reason why HE is under strict sanitary–technical and hygienic demands. Poorly designed building envelope, ignored principles of bioclimatic design and different demands and needs for health and comfort conditions lead to the problem of high energy use for heating and cooling (H/C). The problem of high energy use is today usually solved only non–holistically. Architects and civil engineers are usually concerned about building envelope, while mechanical engineers mainly participate in the design and analysis of heating, ventilating and air−conditioning systems. The problem has to be solved holistically. New H/C systems have to be designed that will satisfy individual demands and needs for heath and comfort conditions, minimize impacts of health hazards and fulfill all regulation demands for HE, with simultaneously attained minimal possible energy use.

The doctoral dissertation deals with the problem of high energy use for H/C of hospital buildings hierarchically, from the analysis of building envelope's characteristics at different locations, to the evaluation of H/C system's efficiency, from the aspect of thermal comfort of their users and building energy use. It introduces exergy concept, where processes inside the human body and processes in a building are jointly treated. With this respect, two models are jointly used: human body exergy balance model and model of building energy use.

The main contribution of the doctoral dissertation includes the development of a new method that enables holistic manipulation of problems in HE and gives us a platform to search for alternative solutions for them. The development of the method was based on the methodology of engineering design by Morris Asimow; it was upgraded considering the specifics of HE. The main emphasis was on the first three steps of

feasibility study: step 1: definition of needs, demands and conditions; step 2: design problem, and step 3: synthesis of the possible solutions. The holistic manipulation of the problems is presented on three groups of health hazards (impact factors): physical, chemical and biological impact factors. The main emphasis is on physical factors, thermal comfort and high energy use for H/C of hospitals. The advantages of the developed method are that all relevant parameters for the design are included; all their interactions are considered as well the searching for effective solutions is enabled. The main goal is to design a H/C system that enables to create healthy and comfort conditions for individual user and at the same time attained minimal energy use for heating and cooling. The objectives of the doctoral dissertation are: (1) development of a method for holistic manipulation of problems and finding alternative solutions to them; (2) solving the problem of high energy use for heating and cooling holistically, using the structure of building envelope and effective H/C system at different locations; (3) solving the problem of high energy use for H/C by keeping at the same time the attained optimal conditions for individual user (required and comfort conditions) and designed (minimal possible) exergy consumption rate valid for thermoregulation; two H/C systems were compared from the aspect of thermal comfort and energy use, i.e. conventional (electric radiators) and LowEx system (ceiling panels); (4) selection the H/C system that enables attainment of optimal conditions (required and comfort conditions) for individual user, designed (minimal possible) human body exergy consumption rate and minimal energy use for H/C at the same time; (5) active regulation and control of required and thermal comfort conditions for individual user.

Experimental part was carried out in real test room at the Faculty of Civil and Geodetic Engineering, University of Ljubljana (UL FGG), and in simulated model room for burn patients. Both rooms had the same geometry (163.4 m^3) and were equipped with conventional and LowEx H/C system. In model room, four analyses were carried out: (1) analysis of exergy consumption patterns for space heating; (2) exergy analysis of thermal comfort for average test subject; (3, 4) exergy analysis of thermal comfort for individual test subjects in normal and extreme experimental conditions. In the framework of the analysis of exergy consumption patterns for space heating we calculated the average heating energy demand for space heating, where we considered building envelope characteristics on different locations; average heating energy demand was compared with calculated exergy consumption rate for the whole heating system and the most effective intervention for decreasing energy use for H/C of the test room was defined. For exergy analysis of thermal comfort, we selected average test subject (a 30 year old male, weighing 70 kg, and 1.75 m tall) and three individual users (burn patients, heath workers and visitor). They were exposed to different experimental conditions, i.e. normal conditions (room conditions) and extreme conditions (hot/cold, dry/wet). Conventional and LowEx H/C system were compared on the basis of calculated human body exergy balance, human body exergy consumption rate (hbExCr), predicted mean vote index (*PMV* index) and measured energy use for H/C of test room.

Results of the analysis of building envelope characteristics on different locations show that energy use depends on location characteristics as well as on the thickness of thermal insulation. Results of exergy analysis are the same as those of energy analysis, if the comparison is made only for the energy supply and exergy supply. However, exergy analysis enables us to analyse the whole heating system and select the most effective intervention for our test room: execution of appropriate thermal insulated building envelope and installation of effective H/C system.

Results of exergy analysis of thermal comfort of average test subjects show that the relative humidity in combination with air temperature has an important effect on separate parts of human body exergy balance and on hbExCr. Results of exergy analysis of thermal comfort on individual test subjects in normal conditions show that the hbExCr and *PMV* index vary between individuals for both systems, even if they are exposed to the same environmental conditions. In extreme hot and dry conditions (35 °C, 30 %) the hbExCr is the largest while at hot and humid conditions (35 °C, 96 %) it is the minimal. LowEx system creates thermally more comfortable conditions than conventional H/C system.

Creating the required conditions in burn patient room (operative temperature T_o 32 °C, 80 % RH_{in}) with LowEx system, we attain conditions that support health care and treatment. These required conditions created with LowEx system cause the lowest

possible hbExCr, lower metabolic thermal exergy rate and also lower exergy rates of exhalation and evaporation of sweat, radiation and convection. These values are lower than in conditions created with conventional system.

Healing oriented conditions for patient do not present also comfortable conditions for healthcare worker and visitor. Therefore, active regulation of thermal comfort zone is developed in the doctoral dissertation. Designed LowEx system enables active regulation of thermal comfort zone with setting up optimal relation between mean radiant temperature and air temperature for every individual separately, without deterioration of thermal comfort of other users. In such way individualized climate is created; consequently, quicker recovery and shorter hospitalization are expected, as well as increased productivity of workers. Conventional system does not enable to do that. The measured energy use for space heating was by 11–27 % lower when using LowEx system than for conventional system. In case of cooling the measured energy use was by 30–73 % lower for LowEx system. At the end the overall harmonization of individual demands and needs for users is attained, required regulated demands for hospital environment are fulfilled and at the same time building energy use for H/C is minimized.

Key words: hospital environment; health hazards; physical factors; chemical factors; biological factors; holistic approach; method for mastering problems; building heating; building cooling; building energy use; thermal comfort; conventional system; low exergy system; exergy consumption rate for space heating; human body exergy balance; human body exergy consumption rate; *PMV* index; burn patient; individualized climate

POVZETEK

Bolnišnice spadajo med kompleksna notranja okolja različnih uporabnikov, dejavnikov tveganj in specifičnih aktivnosti. V bolnišničnem okolju (BO) so prisotni številni uporabniki (bolniki, zaposleni in obiskovalci) z različnimi potrebami in zahtevami po toplotnem udobju, ki jih z obstoječi sistemi gretja, hlajenja in prezračevanja ni moč zadovoljiti. Uporabniki so izpostavljeni številnim dejavnikom tveganj, ki jih razdelimo na fizikalne, kemične, biološke, biomehanske in psihosocialne dejavnike. Specifična aktivnost povzroči, da je BO podvrženo strogim sanitarno−tehničnim in higienskim zahtevam. Slaba zasnova stavbnega ovoja, neupoštevani principi bioklimatskega načrtovanja in različne zahteve in potrebe po zdravih in udobnih razmerah privedejo do problema velike rabe energije za gretje in hlajenje (G/H). Problem velike rabe energije za G/H gretje bolnišnic se danes ne rešuje celostno. Arhitekti in gradbeniki rešujejo problem le na nivoju stavbnega ovoja, strojniki pa so osredotočeni le na delovanje mehanskih sistemov. Problema se je potrebno lotiti celostno. Načrtovati je potrebno nove sisteme G/H, ki bodo zadovoljili individualne potrebe in zahteve po zdravih in udobnih razmerah, zmanjšali vplive dejavnikov tveganj, izpolnili vse zakonske zahteve v BO ob sočasno doseženi minimalni rabi energije.

Doktorska disertacija obravnava problem velike rabe energije G/H bolnišnic hierarhično, od analize stavbnega ovoja na različnih lokacijah, do ocene učinkovitosti sistema G/H z vidika toplotnega udobja uporabnika in rabe energije v stavbi. Vpeljan je eksergijski koncept, ki istočasno obravnava procese v človeškem telesu s procesi v stavbi. S tem namenom sta uporabljena dva modela: model eksergijske bilance v človeškem telesu in model rabe energije v stavbi.

Glavni prispevek doktorske disertacije je razvoj nove metode, ki omogoča celostno obvladovanje problemov, ki jih najdemo v BO. Predstavlja platformo za iskanje alternativnih rešitev. Metoda je razvita po metodologiji inženirskega načrtovanja po Morris Asimowu in nadgrajena za uporabo v BO. Glavni poudarek je na prvih treh korakih študije možnosti: korak 1: analiza dejanskega stanja; korak 2: načrtovanje problema; korak 3: sinteza možnih rešitev. Celostno obvladovanje problemov je predstavljeno na treh skupinah dejavnikov (vplivnih faktorjev): fizikalnih, kemičnih

in bioloških faktorjih. Glavni poudarek je na fizikalnih faktorjih, na toplotnem udobju in veliki rabi energije za G/H bolnišnic. Prednost razvite metode je v tem, da omogoča vključitev vseh parametrov načrtovanja, analizo njihovih medsebojnih interakcij in iskanje učinkovitih rešitev. Glavni cilj disertacije je načrtovanje sistema G/H, ki omogoča zdrave in udobne razmere za individualnega uporabnika ob istočasno doseženi minimalni rabi energije za G/H. Cilji doktorske disertacije so: (1) razvoj metode, ki omogoča celostno obvladovanje problemov in iskanje alternativnih rešitev; (2) rešitev problema velike rabe energije za G/H celostno, od strukture stavbnega ovoja na različnih lokacijah do ocene učinkovitosti sistemov G/H; (3) rešitev problema velike rabe energije za G/H ob sočasno doseženih optimalnih razmerah za individualnega uporabnika (zahtevane in udobne razmere) in načrtovani (minimalni možni) rabi eksergije v človeškem telesu; z vidika toplotnega udobja in rabe energije smo primerjali dva sistema G/H, konvencionalni (električni radiatorji) in LowEx sistem G/H (stropni paneli); (4) izbor sistema G/H, ki omogoča doseg optimalnih razmer (zahtevane in udobne razmere) za individualnega uporabnika ter istočasno doseženo načrtovano (minimalno možno) rabo eksergije v človeškem telesu in minimalno rabo energije za G/H; (5) aktivna regulacija in kontrola zahtevanih in udobnih razmer za individualnega uporabnika.

Eksperimentalni del je potekal v realni testni sobi na Fakulteti za gradbeništvo in geodezijo, Univerze v Ljubljani (UL FGG) in v simuliranem modelu sobe za opeklinskega bolnika. Prostora sta enakih geometrij (163.4 m^3) in opremljena s konvencionalnim in LowEx sistemom G/H. V modelu sobe za opeklinskega bolnika smo izvedli štiri analize: (1) analiza vzorca porabe eksergije za gretje; (2) eksergijska analiza toplotnega udobja za povprečne uporabnike; (3, 4) eksergijska analiza toplotnega udobja za individualne uporabnike v normalnih in ekstremnih razmerah. V okviru analize vzorca porabe eksergije za gretje smo izračunali potrebno energijo za gretje ob upoštevanju značilnosti stavbnega ovoja na različnih lokacijah; le−to smo primerjali z izračunano stopnjo porabe eksergije za celoten sistema gretja ter definirali najbolj učinkovit ukrep za zmanjšanje rabe energije za G/H testnega prostora. Za eksergijsko analizo toplotnega udobja smo izbrali povprečnega uporabnika (30 let star moški, 70 kg telesne teže in 1.75 m višine) ter tri individualne uporabnike (opeklinski bolnik, zdravstveni delavec in obiskovalec). Izpostavljeni so bili različnim eksperimentalnim razmeram, normalne razmere (sobne razmere) in ekstremne razmere (vroče/hladno, suho/vlažno). Konvencionalen in LowEx sistem G/H smo tako primerjali na osnovi izračunane eksergijske bilance v človeškem telesu, stopnje porabe eksergije v človeškem telesu (hbExCr), indeksa napovedanega povprečnega odziva toplotne zaznave (*PMV* index) in izmerjene porabe energije za G/H testnega prostora.

Rezultati analize značilnosti stavbnega ovoja na različnih lokacijah so pokazali, da je raba energije močno odvisna od lokacije ter od izoliranosti stavbnega ovoja. Rezultat energijskih analiz je enak rezultatu eksergijskih analiz, če primerjamo le količino dobavljene energije in dobavljene eksergije. Vendar nam eksergijska analiza omogoča, da z njo preučimo celoten sistem G/H in izberemo najučinkovitejši ukrep za naš testni prostor: izvedbo primerno toplotno izoliranega stavbnega ovoja ter vgradnjo učinkovitega sistema G/H.

Rezultati eksergijske analize toplotnega udobja na povprečnih uporabnikih pokažejo, da ima relativna vlaga v kombinaciji s temperaturo velik vpliv na posamezno dele eksergijske bilance v človeškem telesu in na hbExCr. Rezultati eksergijske analize toplotnega udobja na individualnih uporabnikih v normalnih razmerah pokažejo, da se vrednost *PMV* indeksa in hbExCr spreminja med uporabniki, pri obeh sistemih G/H, čeprav so izpostavljeni enakim eksperimentalnim razmeram. V ekstremno vročih in suhih razmerah (35 °C, 30 %) je hbExCr maksimalna, v vročih in vlažnih (35 °C, 96 %) pa je minimalna. LowEx sistem ustvari bolj udobne razmere od konvencionalnega.

Ko zagotovimo zahtevane razmere v sobi za opeklinskega bolnika (operativna temperatura T_o 32 °C, 80 % RH_{in}) z LowEx sistemom, dosežemo razmere, ki podpirajo nego in zdravljenje bolnika. V ustvarjenih razmerah z LowEx sistemom so vrednosti hbExCr, stopnje eksergije z metabolizmom, stopnje eksergije z dihanjem in evaporacijo potu, stopnje eksergije z radiacijo in konvekcijo nižje kot v razmerah ustvarjenih s konvencionalnim sistemom.

Zdravi pogoji za pacienta ne predstavljajo tudi udobnih razmer za zdravstvenega delavca in obiskovalca. V doktorski disertaciji je razvita aktivna regulacija cone toplotnega udobja. Načrtovan LowEx sistem namreč omogoča aktivno regulacijo cone toplotnega udobja tako, da spremenimo nastavljeno vrednost srednje sevalne temperature in temperature zraka za vsakega uporabnika posebej, ne da bi poslabšali udobje ostalih uporabnikov. Tako ustvarimo individualno klimo, za katero predpostavljamo, da vpliva na hitrejše okrevanje, krajšo hospitalizacijo in dvig produktivnosti zaposlenih. Konvencionalen sistem nam tega ne omogoča. Izmerjena raba energije za gretje in hlajenje je bila za 11–27 % nižja pri LowEx sistemu kot pri konvencionalnem sistemu. V primeru hlajenja pa je bila nižja za 30–73 %. Dosežena je celostna harmonizacija individualnih zahtev in potreb ter zahtevanih razmer za BO, ob istočasno zmanjšani rabi energije za G/H.

Ključne besede: bolnišnično okolje; dejavniki tveganj; fizikalni faktorji; kemični faktorji; biološka faktorji; celosten pristop; metoda obvladovanja problemov; gretje stavb; hlajenje stavb; raba energije v stavbah; toplotno udobje; konvencionalen sistem; nizko eksergijski sistem; stopnja porabe eksergije za gretje prostora; eksergijska bilanca človeka; stopnja porabe eksergije v človeškem telesu; *PMV* index; opeklinski bolniki; individualizirana klima

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ABBREVIATIONS

AB = antibiotic

ACH = air changes per hour

ASS = active solar systems (Passive solar …, 2009)

BREATH = break–through technologies for individual passenger climate and minimization of passenger to passenger transmission (Break–through technologies …, 2005)

C exergy = cool exergy

CDC = Centers for Disease Control and Prevention

CH = hydrocarbons

Ch = cooling hours

CPD = Constructional Products Directive (Directive 89/106/EEC)

DA = dry air

db = dry bulb (temperature)

EADS = European Aeronautic Defence and Space Company

EPBD = Energy Performance of Buildings Directive (Directive 2002/91/EC; recast Directive 2010/31/EU)

ETICS = external thermal insulation composite system

EU = European Union

EU–25 = 25 European Member States

G+ bacteria = Gram−positive bacteria

G– bacteria = Gram−negative bacteria

H/C = heating/cooling (Slovenian abstract: G/H = gretje/hlajenje)

hbEx = human body exergy

hbExCr = human body exergy consumption rate

HE = hospital environment (Slovenian abstract: BO = bolnišnično okolje)

Hh = heating hours

HR = humidity ratio

HVAC = heating, ventilation, air−conditioning systems (ANSI/ASHRAE Standard 62.I.; ISO/TC 205/WG 002)

IAQ = indoor air quality

ICSIE = an integrated control system of internal environment on the basis of fuzzy logic

IEA ECBCS = International Energy Agency, Energy Conservation in Buildings and Community systems

ISO = International Organization for Standardization

LNG = liquidified natural gas

LowEx systems = low–exergy systems, low–temperature–heating and high temperature–cooling systems (heating–cooling panels) (Guidebook to IEA ECBCS Annex 37 …, 2006)

LW = long−wavelength

MRSA = methicillin−resistant *Staphylococcus Aureus*

NO_x = mono−nitrogen oxides

OR = operating room

PAH = polycyclic aromatic hydrocarbon

PM = particulate matter

PSA = passive solar architecture (Passive solar ..., 2009)

P_t = sensor (ultrasonic compact heat and cooling meter)

PURES 2010 = Rules on efficient use of energy in buildings: PURES 2010 (O.J. RS no. 52–2856/10)

RS–MOP = Ministry of the Environment and Spatial Planning, Republic of Slovenia

SD = standard deviation

SO_x = sulphur oxides

TBSA = total body size area (Herndon, 1996: 36)

TLV−TWA limit = threshold limit value−time−weighted average

TVOC = total volatile organic compound

UL FGG, KSKE = University of Ljubljana, Faculty of Civil and Geodetic Engineering, Chair for Buildings and Constructional Complexes

US = United States

VOC = volatile organic compound

VRE = vancomycin−resistant enterococci

W exergy = warm exergy

WHO = World Health Organization

ZRMK = Building Civil and Engineering Institute of Slovenia

SYMBOLS

(Fanger, 1970; ASHRAE Handbook & product …, 1977; Kladnik, 1989; Shukuya, 1994; ISO 7726; Shukuya and Hammache, 2002; ISO 8996; ISO 7730; Çengel and Boles, 2007; ISO 13790; Shukuya et al., 2010)

A_{du} = body surface area (DuBois area) [m^2]

$A_r A_d$ = ratio of the area of the body exposed to radiation versus total surface area (=0.70 for seated postures; 0.73 for standing postures)

A_{window} = window area [m^2]

c_{air} = specific heat of air [=1005 J/(kgK)]

c_{pa} = specific heat capacity of dry air [=28.97 g/mol]

c_{pv} = specific heat capacity of water vapour [=1846 J/(kgK)]

c_{pw} = specific heat capacity of liquid water [=4186 J/(kgK)]

$\dot{C}$ = rate of heat transfer via convection from the surface of a clothed body [W/m^2]

C_{body} = specific heat capacity of human body [=3490 J/(kgK)]

C_{CO2} = concentration of CO_2 [ppm]

C_{cool} = energy use for cooling [MJ, kWh]

C_{heat} = energy use for heating [MJ, kWh]

d = material thickness [m]

dE_{hb}/dt = rate of change in energies of the human body [W/m^2 of body surface]

dE_{sys}/dt = rate of change in internal, kinetic, potential, etc., energies of the system [kW]

dQ = change in heat transfer [kJ]

dS = change in entropy [kJ/K]

dS_{sys} = change in entropy of the system [kJ/K]

dS_{sys}/dt = rate of change in entropy of the system [kW/K]

dt = infinitesimal increment of time [s]

dT = temperature change of interior air layer [K]

dT_{cr} = infinitesimal increment of body–core temperature [K]

dT_{sk} = infinitesimal increment of skin temperature [K]

dX_{sys}/dt = rate of change in exergy of the system [kW]

$\dot{E}_d$ = rate of evaporative heat transfer via skin diffusion due to water vapour diffusion through the skin surface [W/m^2]

E_{in} = energy transfer by heat, work and mass into the system [kJ]

$\dot{E}_{in}$ = rate of energy transfer by heat, work and mass into the system [kW]

$E_{in,hb}$ = energy transfer into the human body [kJ]

$\dot{E}_{in,hb}$ = rate of energy transfer into the human body [W/m^2 of body surface]

$\dot{E}_{\max}$ = rate of the maximum evaporative potential from the skin surface to the surrounding room space [W/m^2]

E_{out} = energy transfer by heat, work and mass out of the system [kJ]

$\dot{E}_{out}$ = rate of energy transfer by heat, work and mass out of the system [kW]

$E_{out,hb}$ = energy transfer out of the human body [kJ]

$\dot{E}_{out,hb}$ = rate of energy out of the human body [W/m^2 of body surface]

$\dot{E}_{re}$ = rate of latent heat loss due to respiration through the skin surface [W/m^2]

$\dot{E}_{sw}$ = rate of heat transfer by sweat evaporation from the skin surface [W/m^2]

f_{cl} = ratio of the human body area with clothing to the naked human body area (=1.05– 1.5)

f_{eff} = ratio of effective area of human body for radiant–heat exchange to the surface area of the human body with clothing (=0.696 – 0.725)

h_c = heat transfer coefficient by convection over clothed body surface [W/(m^{2}K)]

h_{ccl} = average convective heat–transfer coefficient over clothed body surface [W/(m^{2}K)]

h_r = heat transfer coefficient by radiation over clothed body surface [W/(m^{2}K)]

h_{rb} = radiative heat transfer coefficient of black–body surface [W/(m^{2}K)]

$h_{rb,hb}$ = radiative heat transfer coefficient over clothed body surface, during the infinitesimal period of time [(Ons/s)/m^2)]

$\dot{H}$ = rate of internal heat production on the body surface area [W/m^2]

$\text{int}\ rev$ = internally reversible process

LNG_{ch} = ratio of the chemical exergy to the higher heating value of LNG [dimensionless]

LNG_{el} = ratio of produced electricity to the higher heating value of LNG

[dimensionless]

$\dot{L}$ = rate of thermal load on the body surface area [W/m^2]

Il_{in1} & Il_{in2} = internal work plane illumination (workplace 1 and 2) [lx]

Il_{out} = external illumination [lx]

I_{rdo} = reflected solar radiation [W/m^2]

I_{rgo} = direct solar radiation [W/m^2]

m_a = molar mass of dry air [=28.97 g/mol]

$\dot{m}_{ar}$ = mass flow rate of air [kg/s]

m_{body} = body mass [kg]

m_{body}/A_{du} = ratio of body mass to body surface area [kg/m^2]

m_{ra} = mass of room air [kg]

m_w = molar mass of water molecules [=18.05 g/mol]

$\dot{M}$ = metabolic energy generation rate on the body surface area, metabolic rate [W/m^2] (=1 met = 58 W/m^2)

n = number of air changes per hour [h^{-1}]

p_{ao} = pressure of the outdoor air [Pa]

p_{ar} = pressure in the room space [Pa]

p_{vcl} = non–saturated water–vapour pressure [Pa]

p_{vo} = water–vapour pressure of the outdoor air [Pa]

p_{vr} = water–vapour pressure in the room space [Pa]

$p_{vs}(T_{ai})$ = saturated water–vapour pressure at room air temperature [Pa]

$p_{vs}(T_{ao})$ = saturated water–vapour pressure at outdoor air temperature [Pa]

$p_{vs}(T_{cr})$ = saturated water–vapour pressure at body–core temperature [Pa]

$p_{vs}(T_{sk})$ = saturated water–vapour pressure at skin temperature [Pa]

P = atmospheric air pressure in the room space [Pa]

P_e = precipitation detection [dimensionless]

PMV index = predicted mean vote index [dimensionless]

q = space heating exergy demand [W]

q_i = minimal flow (ultrasonic compact heat and cooling meter) [m^3/h]

$\dot{q}_{in}$ = heat flow rate into the wall across the boundary surface [W/m^2]

$\dot{q}_{out}$ = heat flow rate out of the wall [W/m^2]

q_p = nominative flow (ultrasonic compact heat and cooling meter) [m^3/h]

q_s = maximal flow (ultrasonic compact heat and cooling meter) [m^3/h]

Q = heat transfer [kJ]

Q_{core} = heat capacity of body core [$J/(m^2K)$]

Q_g = heat gains from external environment to the heated space by interior heat gains and solar heat gains [MJ]

Q_h = delivered energy to heating system to satisfy the heat use [MJ]

Q_H = amount of heat supplied to heat engine from a high–temperature source of imaginary heat engine (high–temperature reservoir) [kJ]

Q_i = interior heat gains [MJ]

Q_l = heat losses from heated space to the external environment by transmission and by ventilation [MJ]

Q_L = amount of heat rejected from heat engine to a low–temperature sink of imaginary heat engine (low–temperature reservoir) [kJ]

Q/T_{sys} = entropy transfer by heat transfer, from the outer surface of the wall into the surrounding air [kJ/K]

Q_s = solar heat gains [MJ]

Q_{shell} = heat capacity of body shell [$J/(m^2K)$]

R = gas constant [J/(molK)]

R_e = thermal resistance of the exterior air film [m^2K/W]

R_i = thermal resistance of the interior air film [m^2K/W]

R_l = thermal resistance of the layer [m^2K/W]

$\dot{R}_{rad}$ = rate of heat transfer via radiation from the surface of a clothed body [W/m^2]

R_{wall} = thermal resistance of the wall, *R–value* [m^2K/W]

RH = relative humidity [%]

RH_{in} = relative humidity of indoor air [%]

RH_{out} = relative humidity of outdoor air [%]

S_{gen} = entropy generation [kJ/K]

$\dot{S}_{gen}$ = rate of entropy generation [kW/K]

$\dot{S}_{gen,bw}$ = entropy generation rate for the building wall [$W/(m^2K)$]

$S_{gen,hb}$ = entropy generation of the human body [kJ/K]

S_{in} = entropy transfer by heat and mass into the system [kJ/K]

$\dot{S}_{in}$ = rate of entropy transfer by heat and mass into the system [kW/K]

$S_{in,hb}$ = entropy transfer into the human body [kJ/K]

S_{out} = entropy transfer by heat and mass out of the system [kJ/K]

$\dot{S}_{out}$ = rate of entropy transfer by heat and mass out of the system [kW/K]

$S_{out,hb}$ = entropy transfer out of the human body [kJ/K]

t = time [s] and dt is its infinitesimal increment [s]

T = temperature [K, °C]

T_{ai} = room air temperature [K, °C]

$T_{ai,avg}$ = average room air temperature [°C]

T_{ao} = outdoor air temperature as environmental temperature for exergy calculation [K, °C]

$T_{ao,avg}$ = average outdoor temperature [°C]

T_{avg} = average temperature [°C]

$T_{black\ globe}$ = black globe temperature [°C]

$T_{black\ globe,avg}$ = average black globe temperature [°C]

T_{cl} = clothing surface temperature [K, °C]

T_{cr} = body core temperature [K, °C]

$T_{exterior\ west\ wall,avg}$ = average surface temperature of exterior west wall [°C]

T_h = outlet air temperature of the heat exchanger [K]

T_H = temperature of the high–temperature source of imaginary heat engine (high–temperature reservoir) [K]

T_L = temperature of the low–temperature sink (low–temperature reservoir) of imaginary heat engine [K]

T_{mb} = mean surface temperature of the body [°C]

T_{mr} = mean radiative temperature of indoor circumferential room surfaces [K, °C]

$T_{mr,avg}$ = average mean radiative temperature of indoor circumferential room surfaces [°C]

T_o = operative temperature [K, °C]

$T_{o,avg}$ = average operative temperature [°C]

T_r = inlet air temperature of the heat exchanger [K]

T_{range} = temperature range [°C]

T_{sk} = skin temperature [K, °C]

T_{surf} = surface temperature [K]

T_{surr} = temperature of surroundings [K]

T_{sys} = temperature of the system [K]

T_{water} = water temperature [°C]

$T_{working}$ = set up temperature [°C]

T_0 = temperature of the cold source (environmental temperature) [K]

U = thermal transmittance, *U–value* [W/(m^{2}K)]

U_{add} = thermal transmittance of external wall, additional thermal insulation (Case 3) [= 0.15 W/(m^{2}K)]

$U_{add+eff}$ = thermal transmittance of external wall, additional thermal insulation, boiler efficiency 0.95 (Case 3/1) [= 0.15 W/(m^{2}K)]

$U_{ceiling, floor}$ = thermal transmittance of ceiling and floor [W/(m^{2}K)]

U_{cu} = thermal transmittance of external wall, no thermal insulation (Case 1) [= 1.29 W/(m^{2}K)]

$U_{exterior\ wall}$ = thermal transmittance of exterior wall [W/(m^{2}K)]

$U_{interior\ wall}$ = thermal transmittance of interior wall [W/(m^{2}K)]

U_{reg} = thermal transmittance of external wall, required thermal insulation (Case 2) [= 0.28 W/(m^{2}K)]

$U_{reg+eff}$ = thermal transmittance of external wall, required thermal insulation, boiler efficiency 0.95 (Case 2/1) [= 0.28 W/(m^{2}K)]

U_{wall} = thermal transmittance of wall [W/(m^{2}K)]

U_{window} = thermal transmittance of window [W/(m^{2}K)]

v_a = air velocity [m/s]

V = volume of heated space [m^3]

$\dot{V}_{in}$ = volumetric rate of inhaled air [(m^3/s)/m^2]

$\dot{V}_{out}$ = volumetric rate of exhaled air [(m^3/s)/m^2]

$\dot{V}_{w-core}$ = volumetric rate of liquid water generated in the body core, which turns into water vapour and is exhaled through the nose and the mouth [(m^3/s)/m^2]

$\dot{V}_{w-shell}$ = volumetric rate of liquid water generated in the body shell as sweat [(m^3/s)/m^2]

$\dot{X}_{consumed,hb}$ = rate of consumed exergy on the body surface area [W/m^2]

$X_{destroyed}$ = exergy destroyed during a process; equal to exergy consumed [kJ]

$\dot{X}_{destroyed}$ = rate of destroyed exergy during a process; equal to exergy consumed [kW]

X_{heat} = exergy transfer by heat [kJ]

X_{in}= exergy transfer by heat, work and mass into the system [kJ]

$\dot{X}_{in}$= rate of exergy transfer by heat, work and mass into the system [kW]

$\dot{X}_{in,hb}$ = rate of exergy transfer into the human body [W/m^2 of body surface]

$\dot{X}_{LNB}$= rate of chemical exergy supplied to the boiler [W]

X_{out}= exergy transfer by heat, work and mass out of the system [kJ]

$\dot{X}_{out}$= rate of exergy transfer by heat, work and mass out of the system [kW]

$\dot{X}_{out,hb}$= rate of exergy transfer out of the human body [W/m^2 of body surface]

$\dot{X}_{r}$= thermal radiant exergy flow rate from a unit area of building wall surfaces [W/m^2]

X_{ra}= thermal exergy contained by a volume of room air [kJ]

$\dot{X}_{ra}$= rate of net "warm" exergy delivered by the air circulated through the water–to–air heat exchanger [W]

X_{rm} = space heating exergy load for the room space [W]

$\dot{X}_{stored,hb}$ = rate of exergy stored on the body surface area [W/m^2]

w = skin wettedness [dimensionless]

W = amount of work extracted by the heat engine [kJ]

W_d = wind direction [°]

W_p = wind speed [m/s]

α_{in} = coefficient of the heat transmission on the air [W/(m^{2}K)]

α_q = ratio of chemical exergy to the higher heating value of LNG (=0.94) [dimensionless]

α_{pj} = absorption coefficient between the human body surface and a surrounding surface denoted by j [dimensionless] (it can be assumed to be equal to configuration factor, the ratio of incoming diffuse radiation to the human body to the diffuse radiation emitted from surface j in most cases)

α_{sk}= fractional skin mass depending on the blood flow rate to the body shell/skin [dimensionless]

ΔE_{sys} = change in internal, kinetic, potential, etc., energies of the system [kJ]

ΔS_{sys} = change in entropy of the system [kJ/K]

ΔT = temperature change [K]

ΔX_{sys} = change in exergy of the system [kJ]

δS_{gen} = amount of entropy generation during the infinitesimal period of time [(Ons/s)/m^2]

ε = overall emittance of the surface [dimensionless]

ε_{cl} = emittance of the clothing surface [dimensionless] (= 5.7 – 6.3)

ε_{sk} = emissivity of the skin surface [dimensionless] (=0.98)

η = utilization factor for the heat gain [dimensionless]

η_b = thermal efficiency of the boiler [dimensionless]

η_c = Carnot efficiency [dimensionless]

λ = coefficient of thermal conductivity [W/(mK)]

λ_{marble} = coefficient of thermal conductivity of marble [W/(mK)]

λ_{gypsum} = coefficient of thermal conductivity of gypsum [W/(mK)]

$\tau_{ao\ v}$ = glass visible transmittance [dimensionless]

ρ_w = density of liquid water [=1000 kg/m^3]

σ = Stefan–Boltzmann constant [=5.67 10^{-8} W/(m^{2}K^4)]

$\tau_{ao\ v}$ = glass visible transmittance [dimensionless]

TERMS

Active zone: User zone inside active space; a space or a group of spaces within a building with any combination of heating, cooling or lighting requirements sufficiently similar so that desired conditions can be maintained by a single controlling device (ISO/TC 205/WG 002).

Actively regulated required and thermal comfort conditions: Actively regulated required conditions for patients together with thermal comfort conditions for healthcare workers/staff and visitors.

Active solar systems: Systems composed of different mechanical and electrical components; for their activity, they need mainly renewable energy sources (i.e. photovoltaic, solar collectors). They are employed to convert solar energy into usable light, heat, cause air–movement for ventilation or cooling, or store heat for future use (Passive solar ..., 2009).

Active space: Intended 3–D space for specific activity circumscribed by constructional complexes.

Active systems: Systems composed of different mechanical and electrical components; they use non–renewable energy sources for their performance (ISO/TC 205/WG 002).

Anergy: The product of entropy and environmental temperature. It implies dispersed energy; it is expressed in the unit J (W for the rate) (Strnad, 1983; Shukuya et al., 2010).

Annual energy use for heating: Required energy for building heating per heating season; it is expressed in MJ/a; the difference between heating losses from the building (transmission losses, ventilation losses) and heat gains to the building (solar heat gains, internal heat gains) (ISO 13790).

Average heating energy demand: The whole amount of energy use divided by the period of time of the heating season; it is expressed in W (Dovjak et al., 2010a).

Conventional H/C systems: Active systems, they need high value energy sources for their working, usually non–renewable energy sources (electric radiator, split system with indoor unit).

Cool exergy: The amount of exergy contained in a substance relative to its environment. The substance has cool exergy as a quantity of state if its temperature is lower than environment, and is the ability of the substance in which there is lack of thermal energy compared to the environment, to let the thermal energy in the environment flow into it (Shukuya et al., 2010).

Designed exergy consumption rate of the human body: Minimal possible human body exergy consumption rate valid for thermoregulation in optimal conditions.

Dry exergy: Ability of volume of air with water vapour to disperse to the surrounding environment that has higher humidity (Shukuya et al., 2010).

Entropy: Measure for quantity of energy that cannot be transformed into work, measure for system's disorder (ISO/TC 205/WG 002).

Environmental temperature: Reference temperature; the temperature of the environmental space (immediate surroundings).

Exergy: Maximum work obtainable from energy or produced by a system. Maximal work from the process that enables the system to keep balance with the surrounding environment (Rant, 1955; Shukuya et al., 2010).

Exergy consumption inside human body: Energy–entropy processes inside the human body; relation between stored and released thermal energy and entropy. It is calculated from energy balance equation, entropy balance equation and microclimatic parameters inside the room (Shukuya et al., 2010). The calculated human body exergy consumption rate is the rate which is used only for thermoregulation; it is expressed in W/m^2 (body surface).

Exergy consumption rate for the whole process of space heating: The total rate of consumed exergy of the whole chain of supply and demand for space heating. It is calculated in the following respective five sub–systems: 1) building envelope; 2) room air with fan; 3) heat exchanger; 4) water circulation with pump and 5) boiler; it is expressed in W.

Exergy–entropy processes: All natural or human processes, such as bio–chemical processes inside the human body or any technological process, present exergy–entropy processes. Their main characteristics are supply of exergy, consumption of exergy, entropy generation and entropy disposal.

Human body exergy balance: Thermal exergy balance of human body combines the water balance equation, the energy balance equation, the entropy balance equation and the environmental temperature for exergy calculation. It shows how individual human body releases the heat depending on experimental conditions; it consists of exergy inputs, exergy outputs, stored exergy and consumed exergy (Shukuya et al., 2010).

Impact factors: Hazard environmental factors, classified into biological, chemical, physical, biomechanical and psychological factors. Influences depend on doses, exposure time and individual characteristics.

Individualization of personal space: Individual generation and control of all parameters of human comfort (for instance generation and control of air temperature, surface temperature, relative humidity, air velocity, illuminance levels for individual user of the space).

Individualized climate: Individual generation and control of parameters of human comfort for attainment of optimal conditions for individual user.

LowEx systems: Low exergy active systems that need low value energy sources for their performance (renewable energy sources, other sustainable energy sources). Low–temperature–heating and high–temperature–cooling systems which are operated at temperatures close to room temperatures (large surface heating–cooling ceiling panels) (Guidebook to IEA ECBCS Annex 37 ..., 2006).

Near zero energy building: A building using 0 kWh/(m^2a) primary energy. Numerical indicator of primary energy use is expressed in kWh/(m^2a) (Directive 2010/31/EU). Primary energy is energy from renewable or non−renewable sources which has not undergone any conversion or transformation process.

Optimal conditions for individual user: Required conditions (regulated demands and recommendations) and thermal comfort conditions, dealing with individual user of test room.

Passive solar architecture systems: Concepts and methods, included into building for using daylight and heat exchange with the environment on the particular location (Passive solar ..., 2009).

Predicted mean vote (*PMV*): Is an index that predicts the mean value of the votes of a large group of persons on the 7–point thermal sensation scale (+3 hot, +2 warm, +1 slightly warm, 0 neutral, –1 slightly cool, –2 cool, –3 cold), based on the heat balance of the human body (ISO 7730).

Recommended T_o: Recommended (appropriate or required) value for conditions inside of burn patient room equipped with LowEx system; calculated operative temperature for burn patient.

Thermal comfort: Thermal comfort is that condition of a mind, which expresses satisfaction with the thermal environment (ISO 7730).

Users of hospital environment: Patient, healthcare workers/staff, visitor.

Warm exergy: The amount of exergy contained in a substance relative to its environment. The substance has warm exergy as a quantity of state if its temperature is higher than environment and is the ability of thermal energy contained by the substance to disperse into the environment (Shukuya et al., 2010).

Wet exergy: The ability of the volume of air with water vapour to disperse to the surrounding environment that has lower humidity (Shukuya et al., 2010).

1 INTRODUCTION

"Individuality is the state or quality of being an individual; a person separate from others and possessing his or her own needs, goals, and desires; single person with individual characteristics, who is considered separately (Crowie et al., 1994)."

Hospital presents a complex indoor environment (Skoog et al., 2005; Balaras et al., 2007). It should be treated as a crossection of 3–D systems of user, environmental factors and specific activity. Beside sick population (patients), we should not forget healthy individuals (healthcare workers/staff and visitors). Studies in hospital environment (Burch, 1956; Silverman and Blanc, 1957; Silverman et al., 1958, 1966; Hey and Katz, 1970; Smith and Rae, 1977; Arturson et al., 1978; Collins et al., 1981; Martin et al., 1992; Wallace et al., 1994; Mallick, 1996; Parsons, 2002; Nicol, 2004; Melhado et al., 2005; Skoog et al., 2005; Oldenburg Neto and Amorim, 2006; Balaras et al., 2007; Hwang et al., 2007; Hashiguchi et al., 2008; Yau et al., 2011; Dovjak and Shukuya, 2011d) have proved a significant effect of gender, age, acclimatization and health status on individual changes in the perception of human comfort. The statement that patients, staff and visitors could be treated as a uniform group with the same needs, demands and conditions, is completely wrong (Skoog et al., 2005). It is very difficult to comply with regulated demands for patient, which are not necessarily the same as comfort requirements for staff and visitor.

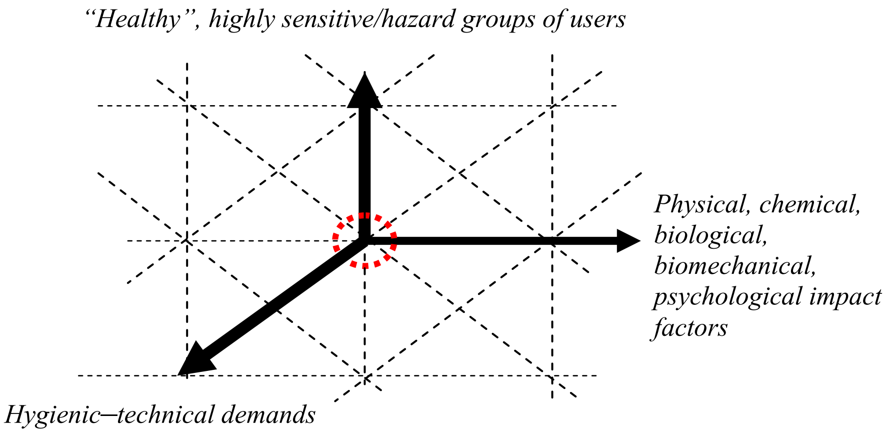

Fig 1: *3–D system of aspects in HE*

Users in hospital environments are exposed to numerous health hazards (hereinafter referred to as impact factors) that can be classified into physical (microclimate, illuminance, noise), chemical (organic and inorganic compounds), biological (bacteria, viruses, other pathogenic microorganisms), biomechanical (compulsory body pose, repeated movement, accidents) and psychosocial (stress). Exposure to such environment could cause acute or chronic effects on human health. Influence depends on dose, exposure time and individual characteristic (Yassi et al., 2001; Dovjak and Shukuya, 2011d).

Specific activity is the reason why hospital environment is under strict sanitary–technical and hygienic demands, especially on the level of prevention and control of nosocomial infections. The main guidance for the design of such environment is overall harmonization of individual demands and needs for users, minimization and elimination of health hazards and fulfillment of all regulated demands for hospital environment (Figure 1).

During the preparation of the doctoral dissertation, an analysis of the present conditions was done. It includes 721 relevant sources of literature, from 1934 up to now. Analysis revealed many problematic areas that have to be confronted and eliminated. Among them, the most important are problems related to old studies, poor regulations, financial problems, user's health and high energy use for heating and cooling of hospitals (Fanger, 1970; Martin et al., 1992; Wallace et al., 1994; ASHRAE HVAC design …, 2003; Sehulster and Chinn, 2003; ANSI/ASHRAE Standard 55; ANSI/ASHRAE/ASHE Standard 170P; Balaras et al., 2007; ASHRAE Handbook HVAC applications …, 2007; Energy efficiency ..., 2007; Beggs et al., 2008; Dascalakia et al., 2008; Qian et al., 2008; ANSI/ASHRAE/ASHE 170).

Residential and tertiary sectors, the major part of which are buildings, accounting for more than 40% of total energy use in the Community, are expanding; a trend which is bound to increase its energy use and hence also its carbon dioxide emissions (Directive 2002/91/EC; Directive 2010/31/EU). Data from General Directorate for Energy and Transport of European Commission (Energy efficiency …, 2007) show that annual energy use in hospitals of 25 European Member States (hereinafter EU–25) is 1.152–1.656 GJ/(m^2a) (320–460 kWh/(m^2a)), and the maximal energy use is

2.376 GJ/(m^2a) (660 kWh/(m^2a)). Annual energy use for heating is 360–486 MJ/(m^2a) (100–135 kWh/(m^2a)) (oil, gas), ventilation 14.4–198 MJ/(m^2a) (4–55 kWh/(m^2a)), climatic systems use 18–50.4 MJ/(m^2a) (5–14 kWh/(m^2a)) (electrician), HVAC systems (heating, ventilation and air–conditioning systems) use 540–734.4 MJ/(m^2a) (150–204 kWh/(m^2a)), lighting 122.4–140.4 MJ/(m^2a) (34–39 kWh/(m^2a)), and sanitary water 216–324 MJ/(m^2a) (60–90 kWh/(m^2a)). Other uses (lifts, cooking, incinerator) represent 266.4–442.8 MJ/(m^2a) (74–123 kWh/(m^2a)). Overall energy use in hospitals in EU–25 is 385.2–540 TJ/a (107–150 GWh/a). It is worth mentioning that the estimated energy saving potential is between 20 % and 45 %, which comes up to 75.6–244.8 TJ/a (21–68 GWh/a). Energy potential with HVAC technologies could be 36–108 TJ/a (10–30 GWh/a), and with building envelope 25.2–79.2 TJ/a (7–22 GWh/a) (Energy efficiency ..., 2007).

Existing statistical data related to building sector indicate just how much energy is supplied, but they do not indicate where exergy is consumed and at which rate. Therefore, it is important to analyze how much exergy is "consumed" by respective parts of a whole system in question in addition to revealing how much energy is supplied for the purpose of heating, cooling, lighting and other demands. When using expressions such as "energy consumption", "energy saving", and even "energy conservation", they implicitly refer to "energy" as energy available from fossil fuel and condensed uranium. The principal characteristic of the majority of well–established scientific terms is that energy "must be conserved". It is therefore smart to use exergy[1] concept, which exactly quantifies "consumption", and to have another look at what is called energy issues in addition to ordinary energy evaluation (Takahashi, 1979; Oshida, 1986; Shukuya and Hammache, 2002).

The problem of high exergy consumption is today usually solved only non–holistically. For example, architects and civil engineers are usually concerned about building envelope, while mechanical engineers mainly participate in the design and analysis of heating, ventilating and air−conditioning systems. No doubt, that improvement of boiler or improvement of building envelope is important.

[1] Exergy is the maximal work obtainable from energy. It is part of energy with a value (Rant, 1955).

But there exists a great need for holistic approach (Dovjak et al., 2010a, 2010b, 2011e, 2011g). Exergy analysis enables us to make connections among processes inside the human body and processes in a building (Shukuya and Hammache, 2002; Shukuya, 2007). It helps to connect optimal thermal comfort conditions with a rational energy use in building.

Individual climate has already been introduced in personal means of transport, i.e. cars. Local ventilation has spread into working environment, since some managers have already realized the positive impact of thermal comfort conditions on productivity of their workers (Melikov et al., 2002). Overall individualization of personal space that enables individual generation and control of all parameters of human comfort (for instance generation and control of air temperature, surface temperature, relative humidity, air velocity, illuminance levels for individual user of the space) has not been implemented yet, either as idea, or as a prototype in a test or real environment (Dovjak et al., 2006, 2010c, 2011a). Furthermore, the method that enables the holistic manipulation of all parameters of indoor environment has not been developed yet.

The main contribution of the doctoral dissertation includes the development of a new method that enables holistic manipulation of all problems found in such environment and gives us a platform to search for alternative solutions for them. The dissertation deals with the problem of high energy use for heating and cooling of hospital buildings, from user and building perspective. Problem of high energy use is treated hierarchically, from the analysis of building envelope's characteristics at different locations, to the evaluation of indoor conditions and thermal comfort of their users. It introduces exergy concept, where processes inside the human body and processes in a building are jointly treated. With this respect two models are jointly used: human body exergy balance model and model of building energy use.

The main goal of the doctoral dissertation is to find such design, construction and use of heating and cooling (hereinafter H/C) system that enables fulfillment of all regulated demands and thermal comfort conditions (hereinafter called optimal conditions) for individual user, by keeping together at the same time minimal possible energy use for heating and cooling of buildings. From the exergetic point of view, the comparison between the H/C systems is carried out, and from these two the final solution is selected.

The dissertation is structured into 11 chapters and 11 Annexes. The Chapter 1 (8 pages) presents the English abstract, Slovenian abstract, and keywords. The Chapter 2 (6 pages) deals with the introduction of the problem. The Chapter 3 (21 pages) deals with theoretical background and includes general characteristics of exergy concept, setting up the human body exergy model, exergy concept in built environment, buildings H/C systems, LowEx systems with relation to thermal comfort. The purpose, objectives and hypothesis of the dissertation are presented in Chapter 4, Chapter 5 and Chapter 6 (4 pages). The Chapter 7 (25 pages) deals with experimental setup and includes the developed methodology and experimental settings. The Chapter 8 (94 pages) deals with the results and discussion of the results. It includes results of feasibility study, results of analysis of exergy consumption patterns for space heating, results of exergy analysis of thermal comfort for average test subjects and individual test subjects as well as results of measurements of energy use for heating and cooling. Conclusions are presented in Chapter 9. The doctoral dissertation is based on the analysis of 721 sources of relevant literature. The literature includes 178 cited references and 8 other references given in Chapter 10. The dissertation involves 80 figures and 58 tables. There are well over 100 footnotes in the dissertation. Detailed explanation of calculations are given in 6 Annexes A, B, D, F–H (19 pages): calculation of operative temperature (Annex A), calculation of human body exergy consumption rate and *PMV* index (Annex B), calculation of average space heating energy demand (Annex D), calculation of exergy consumption rate for space heating (Annex F), calculation of heat flow rate and exergy flow rate through building wall (Annex G) and calculation of thermal radiant exergy flow rate from a unit area of building wall surfaces (Annex H). The comparison between energy analysis and exergy analysis of thermal comfort is presented in Annex C (5 pages). Annex E (42 pages) includes comprehensive tables with references as a result of the analysis of real–time conditions. Annex I (4 pages) contains a table with defined individual characteristics, experimental conditions with quoted references−data used for exergy analysis of human body exergy consumption rate for individual test subjects. Annex J (7 pages) includes tables with additional results of the analysis of human body exergy consumption rate for average and individual test subjects. Annex K (4 pages) contains the table with defined objectives of doctoral dissertation and candidate's contribution.

2 THEORETICAL BACKGROUND

2.1 General characteristics of exergy concept

The first law of thermodynamics is the expression of the conservation of energy principle. It deals with the quantity of energy and asserts that energy cannot be created or destroyed during a process (Çengel and Boles, 2007: 2). The second law, however, deals with the quality and quantity of energy, and actual processes occur in the direction of decreasing quality of energy. More specifically, it is concerned with the degradation of energy during a process, the entropy generation, and the lost opportunities to do work; and it offers plenty of room for improvement. The second law of thermodynamics has proven to be a very powerful tool in the optimization of complex thermodynamic systems (Çengel and Boles, 2007: 433).

The second law deals with changes of entropy (Rant, 1955; Kladnik, 1989: 213; Çengel and Boles, 2007: 406; Shukuya et al., 2010). However, entropy of a system is the measure for the quantity of energy that could not be converted into work. It is a quantitative measure of microscopic disorder for a system (i.e. molecular disorder, randomness of a system), a measure to quantify in what degree an amount of energy or matter is dispersed or how much the dispersion occurs (Çengel and Boles, 2007: 406; Shukuya et al., 2010).

The entropy is defined as:

$$dS = \left(\frac{dQ}{T}\right)_{\mathrm{int}\ rev} \quad [\mathrm{kJ/K}], \tag{1}$$

where dS is entropy change, dQ is change in heat transfer, T is temperature (Çengel and Boles, 2007: 406; Kladnik, 1989: 213).

The second law of thermodynamics states that entropy can be created but it cannot be destroyed. Therefore, the entropy change of a system during a process is greater than the entropy transfer by an amount equal to the entropy generated during the process within the system, and the increase of entropy principle for any system is expressed as:

$$S_{in} - S_{out} + S_{gen} = \Delta S_{sys} \text{ [kJ/K]}, \quad (2)$$

where S_{in} in entropy transfer into the system, S_{out} is entropy transfer out of the system, S_{gen} is entropy generated during a process and ΔS_{sys} is change in entropy of the system (Çengel and Boles, 2007: 406). An alternative statement of the second law of thermodynamics is called decrease of exergy principle. It is the counterpart of increase of entropy principle and is presented below.

The exergy of a system in thermodynamic concept means the maximum work of the process that enables the system balance with surrounding environment. The origin of the word exergy goes back to 1940s, when in the United States in the M.I.T. School of Engineering, the terms "availability" and "available energy" were introduced (Çengel and Boles, 2007: 406; Strnad, 1983). Today, an equivalent term exergy has found global acceptance. The term exergy was introduced in Europe in the 1950s (Rant, 1955).

2.1.1 Exergy consumption theorem

Let us take an example of one subsystem as a simple imaginary heat engine working under a steady–state condition (Figure 2). However, an imaginary heat engine represents a portion of the built environment systems. It is working with the high–temperature source whose temperature is constant at T_H and with the low–temperature sink whose temperature is constant at T_L. The engine converts a part of the heat to work, W, which is not yet dispersed, through the two dispersing flows of thermal energy, Q_H and Q_L, from the high–temperature source to the low–temperature sink. Energy transfer by heat is necessarily accompanied with entropy transfer and entropy generation, while on the other hand, energy transfer by work itself alone is accompanied with no entropy transfer (Kladnik, 1989: 210; Çengel and Boles, 2007: 286, Shukuya et al., 2010).

In the case of a system such as a heat engine as shown in Figure 2, the low–temperature sink can be regarded as the environmental space for the high–temperature source and for the heat engine. Since the concept of entropy is, as mentioned above, a measure to quantify the degree of dispersion and its unit is J/K

(=Onnes)) (W/K (=Onnes/s) for the rate), the dispersed energy level of the high–temperature source surrounded by the environmental space can be expressed as the product of entropy contained by the high–temperature source and its environmental temperature in Kelvin scale, the temperature of the low–temperature sink (Shukuya et al., 2010).

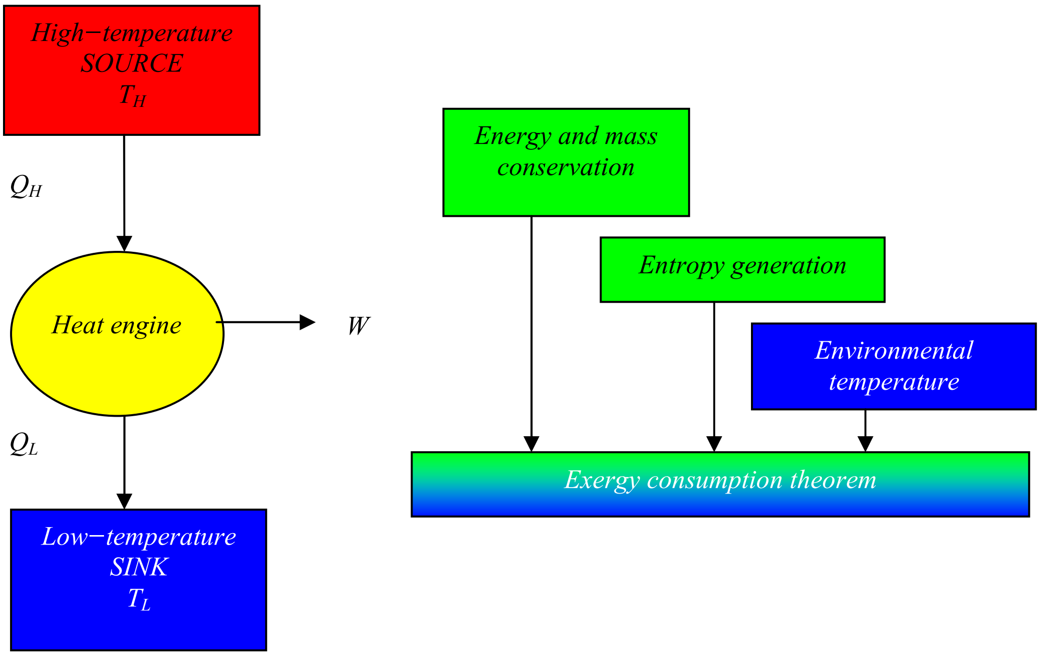

Fig 2: *Imaginary heat engine and exergy consumption theorem (Shukuya et al., 2010: 8)*

The product of entropy and environmental temperature is called “anergy”, which implies dispersed energy; the unit of both energy and anergy is J (W for the rate) (Strnad, 1983; Shukuya et al., 2010). Environmental temperature, T_0, is sometimes called “reference temperature”. Historically speaking, the term “reference temperature” must have originated from the quantification of temperature by measurement with the use of a substance such as water “referring” to particular temperatures, i.e. freezing and boiling temperatures. With this meaning of “reference” in mind, environmental temperature should not be confused with reference temperature. Generally speaking, an amount of energy contained by a certain body consists of two portions of energy: one is not yet dispersed and the other already dispersed; the latter is the energy fully in equilibrium with that in the environmental space whose temperature is exactly the “environmental temperature”. In other words, a portion of energy to be expressed as the difference between total energy and its dispersed portion, anergy, is the amount of energy, which has the ability to bring about dispersion of energy and matter. This is exactly the concept of "exergy". Exergy balance equation is therefore obtained from the two balance equations in terms of energy and entropy together with the concept of "environmental temperature" (Shukuya et al., 2010).

2.1.2 Exergy a measure of work potential

The property exergy is the work potential of a system in a specific environment and represents the maximum amount of useful work that can be obtained as the system is brought to equilibrium with the environment (Çengel and Boles, 2007: 444; Dincer and Rosen, 2007: 10). Unlike energy, the value of exergy depends on the state of environment. The portion of energy that cannot be converted to work is called unavailable energy. Unavailable energy is simply the difference between the total energy of a system at a specified state and the difference between that total energy of a system at a specified state and the exergy of that energy (Çengel and Boles, 2007: 437).

2.1.3 Exergy transfer by heat, work and mass

Exergy, like energy, can be transferred to or from a system in three forms: heat, work and mass flow. Exergy transfer is recognized at the system boundary as exergy crosses it, and it represents the exergy gained or lost by a system during a process. The only two forms of exergy interactions associated with fixed mass or closed system are heat transfer and work (Çengel and Boles, 2007: 452). Therefore, heat transfer is always accompanied by exergy transfer. Heat transfer Q at a location at thermodynamic temperature T is always accompanied by exergy transfer X_{heat} in the amount of:

$$X_{heat} = (1 - \frac{T_0}{T})Q \text{ [kJ]} \tag{3}$$

This relation gives the exergy transfer accompanying heat transfer Q, whether T is greater than or less than T_0. When $T > T_0$, heat transfer to a system increases the exergy of that system and heat transfer from a system decreases it. But the opposite is true when $T < T_0$. In this case, the heat transfer Q is the heat rejected to the cold medium (the waste heat), and it should not be confused with the heat supplied by the environment at T_0. The exergy transferred with heat is zero when $T=T_0$ at the point of transfer. The Carnot efficiency $\eta_c=1-(T_0/T)$ represents the fraction of the energy transferred from a heat source at temperature T that can be converted to work in an environment at temperature T_0 (Çengel and Boles, 2007: 452).

Building wall is a closed system, since no mass crossed the system boundary during the process. Entropy generation and destruction of exergy during a heat transfer process through a finite temperature difference is presented in Figure 3. Note that heat transfer through a finite temperature difference is irreversible, and some entropy is generated as a result. The entropy generation is always accompanied by exergy destruction, as illustrated in Figure 3. Also note that heat transfer Q at a location at temperature T is always accompanied by entropy transfer in the amount of Q/T and exergy transfer in the amount of $(1-T_0/T)Q$ (Çengel and Boles, 2007: 452).

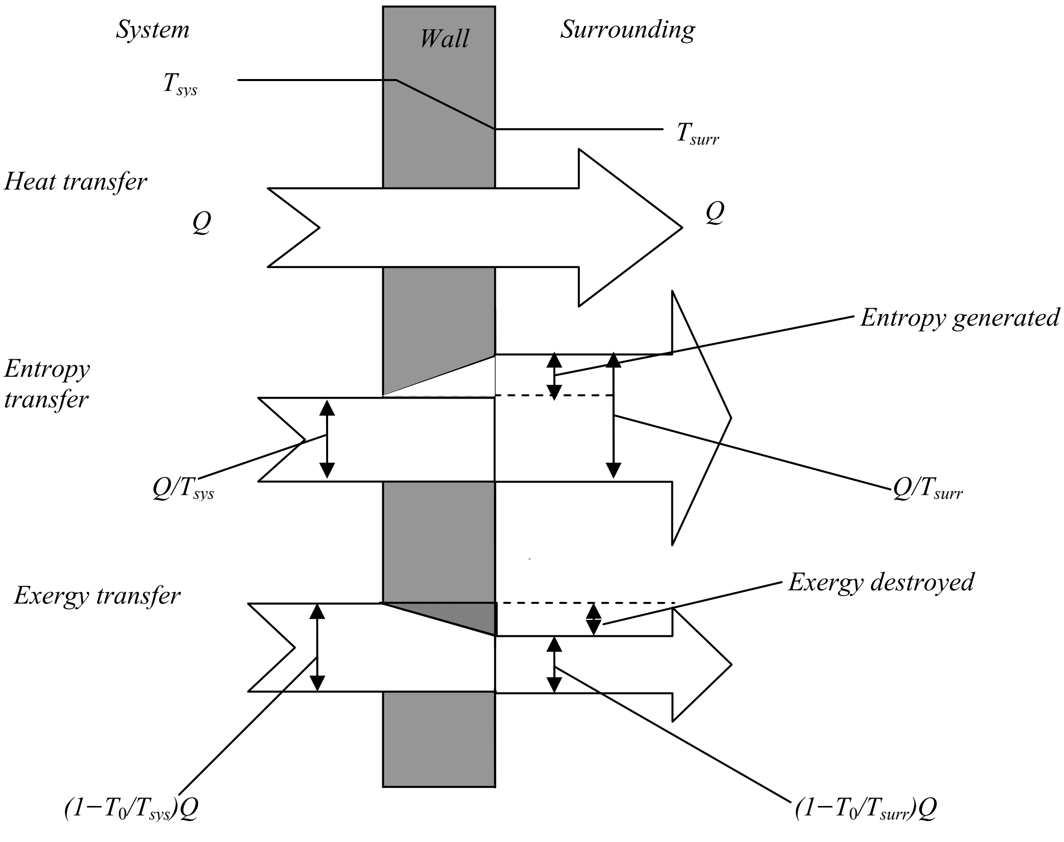

Fig 3: *Graphical representation of entropy generation and exergy destruction during a heat transfer process through a building wall with a finite temperature difference. Q is heat transfer, S_{gen} is entropy generated, Q/T_{sys} is entropy transfer by heat transfer, from the outer surface of the wall into the surrounding air, T_{sys} is temperature of the system, T_{surr} is temperature of surroundings (Çengel and Boles, 2007: 452)*

2.1.4 The decrease of exergy principle and exergy destruction

The conservation of energy principle defines that energy cannot be destroyed during a process. The increase of entropy principle, which can be regarded as one of the statements of the second law indicates that entropy can be created but cannot be destroyed. That is, entropy generation S_{gen} must be positive (actual processes) or zero (reversible processes), but it cannot be negative. An alternative statement of the second law of thermodynamics, called the decrease of exergy principle, is the counterpart of the increase of entropy principle (Çengel and Boles, 2007: 453). Energy, entropy and exergy balances of any system undergoing any process can be expressed as:

Energy balance of a system (Çengel and Boles, 2007: 453):

$$E_{in} - E_{out} = \Delta E_{sys} \quad \text{[kJ]}, \tag{4}$$

where E_{in} is energy transfer by heat, work and mass into the system, E_{out} is energy transfer by heat, work and mass out of the system, ΔE_{sys} is change in internal, kinetic, potential, etc., energies of the system.

Energy balance of a system in the rate form (Çengel and Boles, 2007: 453):

$$\dot{E}_{in} - \dot{E}_{out} = dE_{sys} / dt \quad \text{[kW]}, \tag{5}$$

where $\dot{E}_{in}$ is the rate of energy transfer by heat, work and mass into the system, $\dot{E}_{out}$ is the rate of energy transfer by heat, work and mass out of the system, dE_{sys} / dt is the rate of change in internal, kinetic, potential, etc., energies of the system.

Entropy balance of a system (Çengel and Boles, 2007: 383):

$$S_{in} - S_{out} + S_{gen} = \Delta S_{sys} \quad \text{[kJ/K]}, \tag{6}$$

where S_{in} is entropy transfer by heat and mass into the system, S_{out} is entropy transfer by heat and mass out of the system, S_{gen} is entropy generation, ΔS_{sys} is change in entropy of the system.

Entropy balance of a system in the rate form (Çengel and Boles, 2007: 383):

$$\dot{S}_{in} - \dot{S}_{out} + \dot{S}_{gen} = dS_{sys} / dt \quad \text{[kW/K]}, \tag{7}$$

where $\dot{S}_{in}$ is the rate of entropy transfer by heat and mass into the system, $\dot{S}_{out}$ is the rate of entropy transfer by heat and mass out of the system, $\dot{S}_{gen}$ is the rate of entropy generation, dS_{sys}/dt is the rate of change in entropy of the system.

Exergy balance of a system (Çengel and Boles, 2007: 453):

$$X_{in} - X_{out} - X_{destroyed} = \Delta X_{sys} \quad \text{[kJ]}, \tag{8}$$

where X_{in} is exergy transfer by heat, work and mass into the system, X_{out} is exergy transfer by heat, work and mass out of the system, ΔX_{sys} is change in exergy of the system. $X_{destroyed}$ is destroyed exergy, which is equals exergy consumed.

Exergy balance of a system in the rate form (Çengel and Boles, 2007: 453):

$$\dot{X}_{in} - \dot{X}_{out} - \dot{X}_{destroyed} = dX_{sys}/dt \quad \text{[kW]}, \tag{9}$$

where $\dot{X}_{in}$ is the rate of exergy transfer by heat, work and mass into the system, $\dot{X}_{out}$ the rate of exergy transfer by heat, work and mass out of the system, $\dot{X}_{destroyed}$ is the rate of destroyed exergy (equal to exergy consumed), dX_{sys}/dt is the rate of change in exergy of the system.

The exergy destroyed (exergy consumed) during a process can be determined from an exergy balance, or directly from (Çengel and Boles, 2007: 453):

$$X_{destroyed} = T_0 \, S_{gen} \quad \text{[kJ]}, \tag{10}$$

where T_0 is environmental temperature, S_{gen} is generated entropy. $X_{destroyed}$ must be positive (irreversible processes) or zero (reversible processes), but it cannot be negative.

The exergy destroyed (exergy consumed) during a process in the rate form (Çengel and Boles, 2007: 454):

$$\dot{X}_{destroyed} = T_0 \, \dot{S}_{gen} \text{[kW]}, \tag{11}$$

where $\dot{X}_{destroyed}$ is the rate of destroyed exergy (exergy consumed) during a process, T_0 is environmental temperature, $\dot{S}_{gen}$ is the rate of entropy generation.

2.2 Setting up the human body exergy model

Animals including human being live by feeding on organic matters containing a lot of exergy in chemical forms. They move muscles by consuming it not only to get their food but also not to be caught as food by other animals. All of such activity realized by their body structure and function is made possible by chemical−exergy consumption. The chemical−exergy consumption brings about quite a large amount of "warm" exergy. In fact, this is the exergy consumed effectively by those animals called homeotherms including human being to keep their body–core temperature almost constant, at which various bio–chemical reactions necessary for life proceed smoothly at a controlled rate. This temperature level is generally higher than the environmental temperature (Shukuya et al., 2010).

Either homeotherms or poikilotherms generate a certain amount of entropy in proportion to the exergy consumption inside their bodies in due course of life and they must excrete the generated entropy into their environmental space, by long−wavelength (LW) radiation, convection, conduction, and evaporation. It is vitally important for the homeotherms to be able to get rid of the generated entropy immediately and smoothly to be alive because of their relatively large rate of exergy consumption. We humans are no exception (Shukuya et al., 2010). Doctoral dissertation discusses the exergy balance of human body as a system of homeotherms and then its relation to thermal comfort in the built environment.

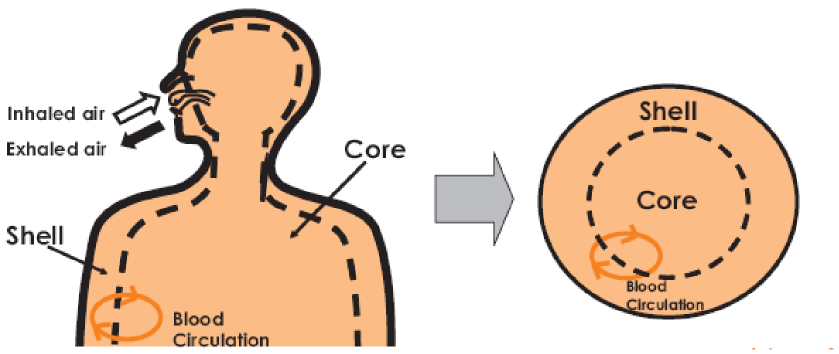

Fig 4: Modelling of a human body consisting of two subsystems: the core and the shell (Shukuya et al., 2010: 10)

Human being is one kind of homeotherms, but the temperature of the peripheral part of the body such as hands and feet in particular varies with the surrounding temperature variations. The human body consists of two subsystems for thermodynamic modelling: the core and the shell, as shown in Figure 4. The core is one subsystem whose temperature is maintained nearly constant at 37 °C almost independently from the variations of surrounding temperature and humidity variations; while on the other hand, the shell is the subsystem whose temperature is rather heavily dependent on their variations. Between these two systems, there is circulation of blood, whose rate is variable dependent on external and internal conditions of the body.

From the thermodynamic viewpoint, the human body is a typical dissipative structure, which self−organizes its form by running the "exergy−entropy" process, the chain of exergy supply, its consumption and the resultant entropy generation, and the entropy disposal. The production of work is never realized without chemical exergy consumption for the body structure and its associated function. If the liquid water contained by foodstuffs is squeezed, then they would burn very well. The same would be true for the human body. There is always water inflow and outflow through the human body. 65 to 70 % of our body weight is always filled with liquid water so that a sudden rise of body temperature is not likely to happen; if it happened, it could cause an irreversible fatal damage of the complex body structure and function. We can say that the structure and the function of our body are formed by a moderate rate of burning foodstuffs in a special manner with abundance of water. In due course, a large amount of thermal energy and entropy is produced necessarily. The thermal energy has to be dumped into the environmental space, because it is accompanied with a lot of entropy generated within the human body for the complex bio–chemical reactions. Otherwise the human body could malfunction as described above. Let us assume that a human body system, as shown in Figure 5, resides in a room space. The temperatures of the human body, room air and outdoor air are assumed to be higher in this order. Thermal energy outgoing across the body surface first enters the room space and then flows out into the outdoor environmental space. The liquid water secreted from the sweat glands forms a thin water film over the skin surface and then it evaporates into the room space, unless the moisture contained by the room air is saturated. A portion of the room air having the water vapour originating from the human body has to be ventilated so that the room air can always allow the moisture discharged from the human body to disperse. Most portions of the outside surface of the body shell are covered by cloth, and the rest is naked; the head, the face, and the hands are usually exposed to the environmental space. The whole shape of human body is complex because of the head, the arms, and the legs hanging on the body centre (Shukuya et al., 2010).

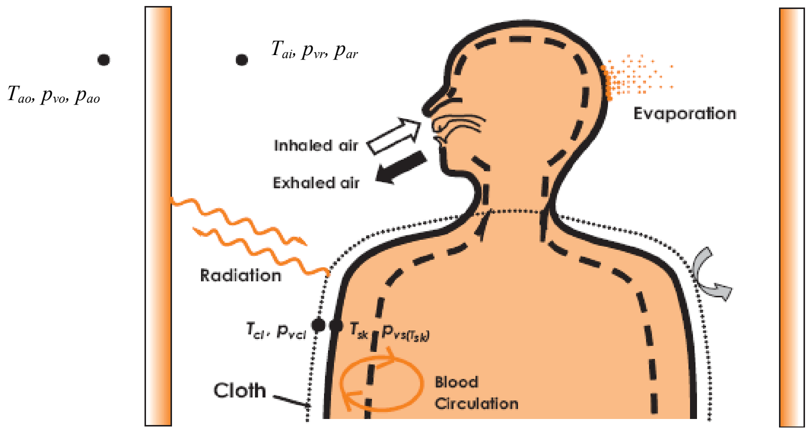

Fig 5: *Human body resides in a room space: experimental variables, heat transfer modes.* T_{ao} *is outdoor air temperature as environmental temperature for exergy calculation,* p_{vo} *is water–vapour pressure of the outdoor air,* p_{ao} *is pressure of the outdoor air,* T_{ai} *is room air temperature,* p_{vr} *is water–vapour pressure in the room space,* p_{ar} *is pressure in the room space,* T_{cl} *is clothing surface temperature,* p_{vcl} *is non–saturated water–vapour pressure,* T_{sk} *is skin temperature,* $p_{vs}(T_{sk})$ *is saturated water–vapour pressure at skin temperature (Shukuya et al., 2010: 12)*

Thermal exergy balance of human body (Shukuya et al., 2010) was derived by combining the water balance equation, energy balance equation[1], the entropy balance equation under steady–state condition. All of them are the resultant equations of the mathematical operations described in Shukuya et al. (2010), together with the

[1] Energy balance equation developed by Shukuya et al. (2010) is based on the modified energy balance two−node model of the human body by Gagge et al. (1971, 1986). Modifications: 1) the net thermal energy transfer due to the humid air transport by breathing and the evaporation of sweat was changed into five explicit forms of the enthalpy values: those of inhaled and exhaled humid air, those of liquid water produced by metabolism in the body core and in the body shell, and that of water vapour discharged from the skin surface by evaporation; 2) the net radiant energy transfer between the human body and its surrounding was made into the explicit forms of radiant energy: one absorbed by the whole of skin and clothing surfaces and the other emitted from the whole of skin and clothing surfaces. The energy balance equation by Shukuya et al. (2010) assumes that the thermal conduction from the foot to the floor or from the back to the chair is simplicity included in the portion of convective energy transfer. It is also assumed that the output of work is neglected and can be allied to the human body at the posture of standing or seating with up to light office work.

environmental temperature for exergy calculation, which is outdoor air temperature. If an overall investigation of the human body exergy balance is made together with space–heating or –cooling system's exergy balance, the environmental temperature to be taken must be the same for both human body and space heating or cooling system (indoor air temperature).

[Warm exergy generated by metabolism]
+ [Warm/cool and wet/dry exergies of the inhaled humid air]
+ [Warm and wet exergies of the liquid water generated in the core by metabolism]
+ [Warm/cool and wet/dry exergies of the sum of liquid water generated in the shell by metabolism and dry air to let the liquid water disperse]
+ [Warm/cool radiant exergy absorbed by the whole of skin and clothing surfaces]
–[Exergy consumption valid only for thermoregulation]
= [Warm exergy stored in the core and the shell]
+ [Warm and wet exergies of the exhaled humid air]
+ [Warm/cool exergy of the water vapour originating from the sweat and wet/dry exergy of the humid air containing the evaporated water from the sweat]
+ [Warm/cool radiant exergy discharged from the whole of skin and clothing surfaces]
+ [Warm/cool exergy transferred by convection from the whole of skin and clothing surfaces into the surrounding air] (12)

The first term of Eq. (12) is the warm exergy produced as the result of chemical exergy consumption for a variety of cellar activities, mainly for the contraction of muscle tissues, the composition of proteins and the sustenance of the relative concentrations of various minerals in the body cells.

The metabolic exergy balance can be expressed as follows:

[Chemical exergy supply]
– [Exergy consumption]

= *[Exergy supply for body function]*

+ *[Warm exergy generated]* (13)

The chemical exergy supplied to the human body by eating food is the exergy trapped by the special compositions of carbon, hydrogen, oxygen, nitrogen and other miscellaneous atoms, which originates from the short−wavelength radiant exergy provided by solar radiation. The hydrogen atoms in the liquid water generated by metabolism originate from the hydrogen atoms contained within the liquid water molecules absorbed by the roots of plants for photosynthesis. All of the warm and wet exergies generated within the human body come from the matters brought by other living creatures. The last term of Eq. (13) is exactly the warm exergy that appeared in the first term of Eq. (12). The exergy−consumption that appeared in the sixth term of Eq. (12) is due to two kinds of dispersion: one is thermal dispersion caused by the temperature difference between the body core, whose temperature is almost constant at 37°C, and the body shell, i.e. the skin, whose temperature range is from 30 °C to 35 °C, and the clothing surface, whose temperature range is from 20 °C to 35 °C; the rest is dispersion of liquid water into water vapour, in other words, free expansion of water molecules into their surrounding space. The chemical exergy consumption shown in Eq. (13) usually amounts to more than 95 % of chemical exergy supply. It implies that the amount of entropy generated in due course is very large, since the amount of entropy generation is exactly proportional to that of exergy consumption. All last five terms of Eq. (12) except the exergy storage, play important roles in disposing of the generated entropy due to chemical exergy consumption within the human body, while at the same time disposing of the generated entropy due to thermal exergy consumption appears in Eq. (12). These processes of outgoing exergy flow together with exergy consumption influence very much human well−being: health and comfort (Shukuya et al., 2010: 10).

2.3 Exergy concept in built environment

Zoran Rant from the Department for Mechanical Engineering at the Technical Faculty in Ljubljana explained the difference between the terms energy and exergy back in 1955. The maximum work obtainable from a given quantity of energy is by all means a very important and remarkable property that deserves its own name. The word “energy” is derived from two Greek words: έν = inside and ἔργον = work, meaning the “work” hidden “inside” a system. The name for the work obtainable “from” (in Greek: έχ or έξ) this system can therefore be composed as “exergy”. The maximum work obtainable from energy shall be referred to as exergy. Any type of energy contains a corresponding amount of exergy. Exergy is the part of energy that has value. Energy without exergy is valueless (Rant, 1955).

The exergy concept can be derived from two fundamental concepts, energy and entropy, and the concept of environmental temperature. It can explicitly indicate how much is consumed in a variety of natural phenomena occurring inside and outside built environment (Takahashi, 1979; Oshida, 1986; Shukuya and Hammache, 2002). A system that is in equilibrium with its surroundings has zero exergy and is said to be in dead state. Exergy is the used work potential of the energy. When we use energy (to heat our homes for example), we are not destroying any energy; we are merely converting it in to a less useful form, a form of less exergy (Çengel and Boles, 2007: 479).

All natural or human processes, such as bio–chemical processes inside the human body or any technological process, present exergy–entropy processes. Their main characteristics are supply of exergy, consumption of exergy, entropy generation and entropy disposal. The use of the exergy concept in the built environment in relation to thermal comfort is relatively new. Exergy analysis jointly treats processes in a building with those inside the human body (Shukuya and Hammache, 2002) and enables us to design a new H/C system for individual user (Dovjak et al., 2010c, 2011a; Dovjak and Shukuya, 2011d).

Final energy use in EU–25 in 2007 was 47.1 PJ (13.1 TWh). In 2007, households in EU–25 presented 24 % of final energy use (11.5 PJ or 3.2 TWh) and tertiary buildings 11 % (5.3 PJ or 1.5 TWh). Average building heating energy use in Europe was more than 627.5 MJ/(m^2a) (174.3 kWh/(m^2a)). In 2005 final energy use in EU–25 was 47.7 PJ (13.3 TWh), and in households and tertiary sector 19.7 PJ (5.5 TWh). In 2005, households in EU–25 presented 25 % of final energy use and tertiary buildings 14 % (Energy yearly …, 2008; Final energy …, 2009). In 2007 final energy use in Slovenian households was 43.9 TJ (12.2 GWh), which corresponds to 22 % of final energy use (204.02 TJ or 56.7 GWh). Final energy use in Slovenian tertiary sector in 2007 was 5.4 TJ (1.5 GWh), which corresponds to 3 % of final energy use. The use of electric energy presented 19.9 TJ (5.5 GWh). In 2002, 45 % of electric energy was used for domestic appliances and lighting, 25 % for sanitary water, 22.5 % for building heating and 7.5% for cooking (Energy yearly …, 2008; Final energy …, 2009). Statistical Office of the Republic of Slovenia shows that average annual heating energy use in residential buildings is 360 MJ/(m^2a) (100 kWh/(m^2a)) (Energy consumption ..., 2008). Measured energy use by Building and Civil Engineering Institute of Slovenia ZRMK d.o.o. is 720 MJ/(m^2a) (200 kWh/(m^2a)) (Internal …, 2001; Building energy use …, 2001). EU has been focusing on several initiatives in the field of energy efficiency. Minimum energy requirements for buildings are defined together with technical demands for mechanical systems, such as boilers and air−conditioning equipment (Directive 89/106/EEC; Directive 92/42/EEC; Directive 2002/91/EC; For an energy–efficient …, 1999; Directive 2010/31/EU; Regulation EU 305/2011). The main directives of the European Commission relating to buildings and construction products are Constructional Products Directive 89/106/EEC (hereinafter CPD) with Interpretative document No.6: Energy economy and heat retention, Regulation EU 305/2011 on harmonized conditions for the marketing of construction products and recast Energy Performance Building Directive 2010/31/EU (hereinafter called EPBD). EPBD emphasizes the minimization of building energy use and the use of renewable energy sources as two important actions for energy independence among EU members (Directive 2010/31/EU). The importance of energy efficiency in buildings for attaining the 20–20–20 targets[1] was stressed by EPBD (Directive 2010/31/EU). EPBD requires nearly zero energy buildings or very low amount of energy required that should be covered to a very significant extent by energy from renewable sources, including energy from renewable sources produced on−site or nearby. Based on the EPBD`s definition, near zero energy building is a building using 0 kWh/(m^2a) primary energy.

[1] 20–20–20 targets of EPBD (Directive 2010/31/EU) include the 20 % reduction of the Union's energy use by 2020, at least 20 % reduction of the overall greenhouse gas emissions by 2020, and the use of energy from renewable sources accounting for 20 % of the total Union's energy use by 2020.

Numerical indicator of primary energy use is expressed in kWh/(m^2a) (Directive 2010/31/EU). Construction Act (2002) presents the umbrella law in the field of energy use in Slovenia. The specific technical demands that have to be fulfilled for efficient energy use in buildings in the field of thermal insulation, heating, ventilation, cooling, sanitary water and lighting are defined in Rules on efficient use of energy in buildings: PURES 2010 (2010) (herein after called PURES 2010). Article 6 defines boundary conditions for the calculation of building energy use and include building's life expectancy, type of use, climate data, construction materials, position and orientation of buildings, parameters of indoor environment, installed systems and devices and use of renewable energy sources. The upper limits for efficient energy use are calculated with equations defined in article 7. The upper limits depend on annual temperature of a specific location and have to be calculated for every building separately. They are expressed as maximal coefficient of specific transmission heat losses through building envelope, or maximal annual energy use for heating and cooling, or maximal thermal transmittance (*U–value*) for constructional complexes defined in Table 1, Sec. 3.1.1 of Technical guidelines for construction ...(2010). Moreover, previous Rules on efficient use of energy in buildings (2008) defined permissible yearly primary energy use (art. 6, 7) and the demands for building envelope (art. 9). For example, the maximal *U–value* for exterior wall was 0.28 W/(m^2K). In Rules on thermal insulation and efficient energy use in buildings from 2002, the maximal *U–value* for exterior wall was 0.6 W/(m^2K) (Rules on thermal insulation …, 2002). In previous rules climate data, local characteristics and indoor climate demands were not considered for the definition of minimum energy performance requirements. They become important with recast EPBD (Directive 2010/31/EU) and were transferred into new PURES 2010 (Rules on efficient …, 2010).

Typical Slovenian everyday practice is that 10 cm of thermal insulation on exterior building walls is enough for effective energy savings for the whole building at any location. Sales analysis for Slovenian market carried out by Fragmat, the biggest manufacturer of thermal insulation materials in Slovenia, shows that the most popular thickness of external thermal insulation composite systems (ETICS) is 12 cm

(Internal …, 2001). It takes a lot of time to implement any new regulation demands and situation on existing buildings could not be changed so easily. If Slovenia wants to fulfill the 20–20–20 targets, all problems for high energy use have to be analyzed and the most effective solutions have to be defined. A holistic approach is needed (Dovjak et al., 2010a, 2010b, 2011e, 2011g). Energy and exergy analyses together help us understand the problem and enable us to search for effective solutions.

2.4 Building H/C systems

First focus is on current H/C systems inside the building. Passive solar architecture (hereinafter called PSA) systems are defined as concepts and methods that could be included into the building for using thermal and daylight fluxes at a particular location. To improve their efficiency, they may use small amounts of either renewable or non–renewable energy sources. Active systems are in general defined as systems composed of different mechanical or electrical components. For their activity, they need mainly non–renewable energy sources, mainly fossil fuels (ISO/TC 205/WG 002). Additionally, active solar systems (hereinafter called ASS) are also composed of different mechanical or electrical components, but for their activity they need mainly renewable energy sources (i.e. photovoltaic, solar collectors). They are employed to convert solar energy into usable light, heat, cause air–movement for ventilation or cooling, or store heat for future use (Passive solar ..., 2009).

New active systems that are fully harmonized with passive ones have to be found. The basic principle of a good bioclimatic design presents a statement that the design of a building envelope is based on location characteristics. It means that the thermal dynamics of the building is regulated with a good design of a building envelope[1] and all available renewable energy sources are used on the particular location. Required energy for building heating and cooling (hereinafter annual building energy use for heating and cooling) presents the difference between heat losses from the building

[1] Good design of a building envelope works in such way that during winter conditions it enables solar heat gains from outside environment into the building and at the same time prevents heat losses from the building to the outside environment. During summer, it prevents heat gains into the building and thus its overheating problems.

and heat gains to the building[1]. It is supplied by active systems that are designed according to specific indoor space requirements and user specifics. In most cases, conventional active systems are used for heating and cooling (for instance room heating with radiator). Nowadays, generally conventional high value energy sources, based on non–renewable energy sources are used. At the market there also exist what we call low exergy active systems (hereinafter LowEx) that use low value energy sources such as renewable and other sustainable sources. They enable building heating and cooling at room temperatures.

Currently there are many different LowEx technologies available that could be classified as surface H/C systems (i.e. floor, ceiling, wall H/C systems), air H/C systems (e.g. recuperators), generation/conversion of cold and heat (e.g. heat pump, sun collectors), thermal storage (e.g. seasonal storage wall), distribution (e.g. district heating). The doctoral dissertation will be focused on surface H/C systems (large surface heating–cooling ceiling panels). They present low–temperature heating and high–temperature–cooling systems. The term LowEx system will relate to heating–cooling ceiling panels. Heating and cooling panels are made from materials with high density and specific heat capacity (e.g. concrete, stone, carbon). The medium inside pipe system of panels could be water or glycol. The temperature of the medium is in cooling panel 10–20 °C, and in heating panel 25–50 °C. In conventional systems the temperature of the medium is higher, i.e. 70/50 °C or 90/70 °C for heating systems and 5–10 °C for cooling systems (Guidebook to IEA ECBCS Annex 37 ..., 2006).

Heating–cooling panels have many advantages in comparison with conventional H/C systems. The most important advantages present improvement of comfort conditions (Zöllner, 1985; Dongen, 1985; Skov and Valbjorn, 1990; Cox et al., 1993; Fort, 1995; Olesen, 1997, 1998; Dijk et al., 1998; Museums energy efficiency …, 1999; Sammaljarvi, 1998; Krainer et al., 2007, 2011; Dovjak et al., 2010c, 2011a, 2011c, 2011f; Košir et al., 2010; Dovjak and Shukuya, 2011d), reduction of energy use for heating and cooling of buildings (Olesen, 1997, 1998; Dijk et al., 1998; Museums

[1] Heat losses from the building include transmission and ventilation losses. Heat gains to the building include solar heat gains and interior heat gains (human metabolism, released heat from appliances) (ISO 13790).

energy efficiency …, 1999; Krainer et al., 2007, 2011; Sakulpipatsin, 2008; Dovjak et al., 2010c, 2011a, 2011c, 2011e, 2011f; Košir et al., 2010; Dovjak and Shukuya, 2011d) and improved indoor air quality due to higher relative humidity of air, higher number of air changes, and lower concentration of mites (Schata et al., 1990; Lengweiler et al., 1997; Sammaljarvi, 1998; Shukuya in Hammache, 2002; Shukuya, 2006a, 2007, 2008a, 2008b; Shukuya et al., 2006b, 2010).

Heating–cooling panels have proved to have positive impact on thermal comfort conditions. Large heated and cooled surface area enables more pleasant thermal conditions in a room than point sources of heating and cooling with conventional systems (i.e. usually room radiator and split system with indoor unit). Moreover, heating–cooling panels cause 50–70 % higher heat transmission with radiation flux in comparison with conventional systems where the transmission is only 20–40 % (Zöllner, 1985). Heating–cooling panels have proved to have lower vertical temperature gradient (Dijk et al., 1998; Olesen, 1997; Cox et al., 1993; Museums energy efficiency …, 1999; Krainer et al., 2007); optimal temperature of floor in the range of 20–28 ºC (Olesen, 1997); slow temperature fluctuation (Olesen, 1997, 1998; Fort, 1995); lower draught perception and air turbulence (Olesen, 1997, 1998; Fort, 1995). Due to heating of surfaces, the air temperature can be 1–2 °C lower with unchanged comfort conditions (Dongen, 1985; Dovjak et al., 2010c). Achievement of lower air temperature decreases irritation of respiratory mucosa (Skov in Valbjorn, 1990) and causes lower reactivity of air particles (Sammaljarvi, 1998).

Heating–cooling panels have positive effect on lower exergy consumption for heating and cooling of buildings. The main principle for diminished building exergy consumption is in theoretical concept that is based on two laws of thermodynamics (law of conservation of energy and the entropy law). The amount of heat energy contained in 20 litres of water at 40 °C is the same as that of 5 litres at 100 °C (1674 KJ = 1674 KJ), but exergy is different; the latter is three–and–a–half times larger than the former (55 KJ < 194 KJ)[1] (Towards sustainable …, 2005). Heating–cooling

[1] Environmental temperature is 20 °C as the reference state for the internal energy.

panels decrease heat losses through interior building elements. Heat loss to interior environment has to be prevented with efficient thermal insulation of building envelope (Olesen, 1997, 1998; Dijk et al., 1998; Dovjak et al., 2010a, 2010b, 2011e, 2011g). Further improvement can be attained by special structure of outside wall with discrete H/C wall panels (Museums energy efficiency …, 1999; Krainer et al., 2007). Because of lower air temperature, the heat losses with ventilation are reduced by 5 % of the building energy use per year (1.6 GJ or 0.4 MWh) (Olesen, 1997, 1998; Dijk et al., 1998). In comparison with conventional H/C systems, the heating–cooling panels cause 67 % lower energy use for heating (Museums energy efficiency …, 1999; Krainer et al., 2007) and 40 % lower annual energy use for cooling (The top ten …, 2007). An important fact is also lower investment costs for HVAC, from € 530,319 to € 434,075, i.e. approx. 20 % reduction in the case of Slovenian Ethnographic Museum (Museums energy efficiency …, 1999; Krainer et al., 2007).

Heating–cooling panels have positive effect on indoor air quality. Because of direct heating of surfaces and not of room air, they enable more air changes and in such way better indoor air quality. In comparison with conventional H/C systems, lower levels of air fluctuation, higher relative humidity of room air, lower concentration of mites and lower concentration of air suspended particles on surfaces have been proven (Schata et al., 1990; Lengweiler et al., 1997; Sammaljarvi, 1998). Beside numerous advantages, H/C systems also enable effective cleaning and maintenance and lower microbiological risk (*Legionella* in split system with indoor unit) (Curtis, 2008). Heating–cooling ceiling panels have been studied mainly in in–situ conditions, where thermal comfort was simulated and energy use was calculated (Vangtook and Chirarattananon, 2006; Myhren and Holmberg, 2008; Nagano and Mochida, 2004). There exist only a few studies where ceiling panels were applied into hospitals (Senuma, 1998).

2.5 Thermal comfort conditions and LowEx systems

Operative temperature (hereinafter called T_o) is defined as the uniform temperature of an enclosure in which an occupant would exchange the same amount of heat by radiation plus convection as in the existing non–uniform environment (ASHRAE Handbook & product …, 1977; ISO 7726; ISO 7730). T_o is a function of air temperature, surface temperatures, air velocity and individual characteristics (skin temperature, effective clothing insulation, metabolic rate). Calculation of T_o and direct measurements are defined in ISO 7726, and the equation for the calculation of T_o for the purpose of doctoral dissertation is defined in Annex A.

The basic purpose of heating–cooling panels is related to large surfaces that are heated or cooled and in such a way contribute to higher or lower mean radiant temperature of circumferential room surfaces (hereinafter called T_{mr})[1]. Thus, air temperature can be less heated or cooled due to unchanged T_o. With conventional H/C systems T_o can be regulated only with air temperature (hereinafter called T_{ai}), while with heating–cooling panels the regulation is possible also with surface temperatures.

Experiences of architects and engineers working on the design of comfort conditions in winter show that higher T_{mr} and lower T_{ai} can result in more acceptable comfort conditions. This coincides with the fact that thermally comfortable conditions equal to thermal neutrality seem to lead to lower human body exergy consumption rate valid for thermoregulation. The relation was first investigated by Isawa et al. (2003), Shukuya et al. (2003) and Shukuya (2006a). Simone et al. (2010, 2011a, 2011b) studied the relation between the human body exergy consumption rate and the human thermal sensation. Results (Simone et al., 2010, 2011a, 2011b) showed that the minimum human body exergy consumption rate was related to the thermal sensation votes close to thermal neutrality, tending to slightly cooler side of thermal sensation. The whole human body exergy balance under typical summer conditions in hot and

[1] T_{mr} is the uniform temperature of an imaginary enclosure in which radiant heat transfer from the human body is equal to the radiant heat transfer in the actual non–uniform enclosure. T_{mr} can be calculated from measured values of the temperature of the surrounding walls and the size of these walls (ISO 7726).

humid regions was analyzed by Iwamatsu and Asada (2009) and Shukuya et al. (2010). Tokunaga and Shukuya (2011) investigated the human–body exergy balance calculation under un–steady–state conditions.

So far only the exergy consumption of an average man in normal conditions has been considered, but not of individual users in different environments (i.e. hot/humid, cold/humid, hot/dry, cold/dry). However, ANSI/ASHRAE Standard 55 recommends that the relative humidity in occupied spaces has to be controlled in the ranges from 30 % up to 60 % and at air temperature between 20 °C and 25 °C. In hot and humid environment, the values could differ from the recommended conditions and affect thermal comfort (Yu et al., 2003; Mallick, 1996). Yu et al. (2003) showed that in tropical climate, the humidity level is a governing factor in the variations of thermal sensations when subjects are exposed to an almost constant room temperature of 23 °C and a room average air velocity of 0.07 m/s. In some cases, much lower or higher levels are defined also for patient therapy where air−conditioning could be an important factor or in some instances also a major treatment (ASHRAE Handbook HVAC applications …, 2007). ASHRAE Handbook HVAC applications …(2007) defines: “a ward for severe burn victims should have temperature controls that permit adjusting the room temperature up to 32 °C and relative humidity up to 95 %”. On the path towards sustainability of our future buildings, the possibility of the design of H/C systems that is based on individual user needs beside minimal energy use for heating and cooling has to be well considered.

3 PURPOSE OF DOCTORAL DISSERTATION

The purpose of doctoral dissertation is to develop a method for holistic manipulation of all problems that appear in the indoor environments and to search for alternative solutions to them. It follows the procedure from abstract (planning the method) to concrete (selection of final solution).

The holistic manipulation of problems is presented on chemical, biological and physical factors in hospital environment. The main focus is on physical factors and high energy use for heating and cooling of hospital buildings. The problem can be solved with existing conventional or with LowEx systems for heating and cooling. From the exergetic point of the view, the comparison of both H/C systems in relation to optimal conditions for individual user[1] and building energy use for heating and cooling will be investigated. Final solution enables mapping of energy flows between different hierarchy levels (Figure 6).

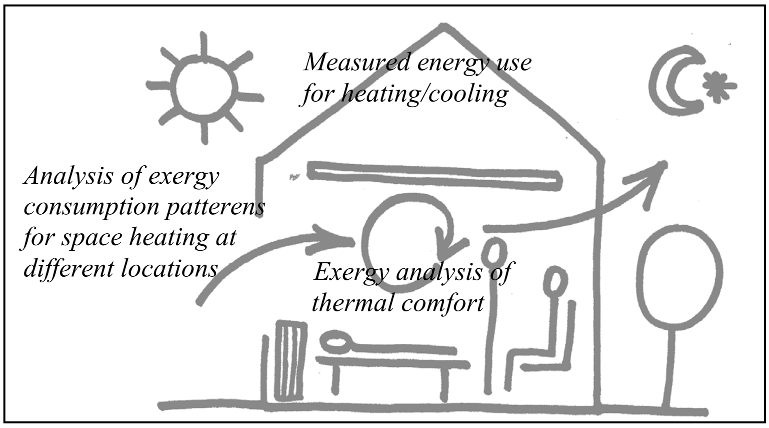

Fig 6: *Mapping of energy flows between different hierarchy levels*

The doctoral dissertation is focused on a patient room equipped with both H/C systems (conventional, LowEx), and three possible users (patient, healthcare worker, visitor). First, in the "*Analysis of exergy consumption patterns for space heating*", the whole system (from boiler to building envelope) is analyzed at different locations,

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[1] Optimal conditions present the required conditions for burn patient room (regulated demands and recommendations) and thermal comfort conditions for healthcare worker and visitor.

exergy analysis is done together with energy analysis, and results are compared and discussed. In the continuation, both systems are compared with "*Measured energy use for heating and cooling*" and "*Exergy analysis of thermal comfort*" for an average man and individual user exposed to normal and extreme room conditions.

The required conditions for burn patient room 32 °C air temperature and 95 % relative air humidity (ASHRAE Handbook HVAC applications …, 2007) do not necessarily present thermal comfort conditions for health workers and visitors at the same time and do not always reflected in minimal exergy consumption rate inside the human body. Regarding that, the doctoral dissertation aims at finding such H/C system that creates optimal conditions for individual user and minimal possible building energy use. Based on the results of experimental work, the final solution is selected and actively regulated thermal comfort zone with respect to required conditions is designed.

4 OBJECTIVES OF DOCTORAL DISSERTATION

The objectives of the doctoral dissertation are (1 – 5):

1) **To develop a method** for holistic manipulation of problems and find alternative solutions to them.
2) **To solve the problem** of high energy use for heating and cooling holistically, using the structure of building envelope and effective H/C system at different locations.
3) **To solve the problem** of high energy use for heating and cooling by keeping at the same time the attained optimal conditions for individual user (required and comfort conditions) and designed (minimal possible) exergy consumption rate inside the human body[1].
   3a) To compare the two H/C systems (conventional system and LowEx system) from exergetic point of the view. It includes exergy analysis of thermal comfort conditions for individual user and measured energy use for heating and cooling of active space. Exergy analysis of thermal comfort conditions will be based on the calculation of human body exergy balance, human body exergy consumption rate, and *PMV* index.
   3b) To find the connection between optimal conditions (required and comfort conditions) and individual human body exergy consumption rate.
   3c) To find out which H/C system enables at the same time attainment of optimal conditions (required and comfort conditions) for individual user, designed (minimal possible) human body exergy consumption rate and minimal energy use for heating and cooling.
4) **To select the solution** of the H/C system that enables attainment of optimal conditions (required and comfort conditions) for individual user, designed (minimal possible) human body exergy consumption rate and minimal energy use for H/C at the same time.
5) **To enable active regulation** and control of required and thermal comfort conditions for individual user.

[1] The term designed exergy consumption rate inside the human body means minimal possible human body exergy consumption rate valid for thermoregulation, regarding the required and thermal comfort conditions.

5 HYPOTHESIS

The developed method enables holistic manipulation of problems, selection of system solution and active regulation of thermal comfort zone, and the required conditions. It is predicted that selected LowEx system will improve energy efficiency of test room as well as attain optimal conditions (required and comfort conditions) for individual user, because (1 – 3):

1) Conventional system does not enable attainment of optimal conditions (required and comfort conditions) for individual user, attainment of designed (minimal possible) exergy consumption rate inside the human body and at the same time minimal possible energy use for heating and cooling.
2) LowEx system enables active regulation of optimal conditions (required and comfort conditions) in active zone for individual user, designed (minimal possible) exergy consumption rate inside the human body, and at the same time low energy use for heating and cooling.
3) LowEx system enables 30–40 % lower energy use for heating and cooling of test room in comparison with conventional system.

The hypothesis is based on the results of international EU project MUSEUMS NNE5–1999–20 (Museums energy efficiency …, 1999). In the framework of the project, heating–cooling panels were installed in Slovenian Ethnographic Museum. When the conventional heating system was replaced by heating–cooling panels, the annual energy use for heating was reduced by 68 %, from 562 MJ/(m^2a) (156 kWh/(m^2a)) to 181.4 MJ/(m^2a) (50.4 kWh/(m^2a)). Annual energy use for cooling was low, 40.3 MJ/(m^2a) (11.2 kWh/(m^2a)) (Museums energy efficiency …, 1999; Krainer et al., 2007, 2011; Košir et al., 2010). Comparison with a similar building (museum with the same building envelope structure and surface area) showed 40 % decrease of annual energy use for cooling with heating–cooling panels (The top ten …, 2007).

All the work presented in this dissertation is contribution of the author Mateja Dovjak. Objectives of doctoral dissertation, time schedule, location, author's contribution, performer, supervisor and field are presented in Table 58 of the Annex K.

6 EXPERIMENTAL

6.1 Methodology

6.1.1 The basis for method development

The method was primarily developed for aircraft environment (Dovjak et al., 2006). The aircraft cabin was found to be appropriate as a model of indoor environment, because of its specific characteristics of individualized space and potential problems evolving from it. The methodology of designing individualized personal space was developed in the framework of the project proposal to FP5, EC TREN (BREATH, Break–through technologies for individual passenger climate and minimization of passenger to passenger transmission, coordinated by EADS Deutschland GmbH, Corporate Research Centre Germany, where UL FGG was one of the partners in the project) (Break–through technologies …, 2005). The main goals of the project were development and testing of new technologies for individualization of indoor climate and minimization of pathogens and volatile organic compounds transmission in the cabin air. The doctoral thesis will be focused on hospital environment. Transformation of methodology from aircraft environment to hospital environment will be presented below.

6.1.2 Transformation into hospital environment

The methodology comprises the systematization of impact factors, evaluation and definition of problems and indication of their solving. The goal is to design an individualized space in order to improve comfort of users and consequently their performance efficiency together with preservation and protection of their health. The expected results are based on the evaluation of a group of physical factors in hospital environment. The final solution is selected H/C system. The method can also be used as a basis for systematic manipulation of other factors in all indoor environments such as residential, educational or commercial buildings. If transformation from aircraft environment to a specific hospital environment is possible, it can be concluded that the method could be applicable in any other working or living environment.

6.1.3 Method: Morphology of engineering design

“All models are wrong; some are useful” (Box and Draper, 1987).

The methodology is based on the Morris Asimov’s morphology of engineering design that comprises seven phases: Phase I–Feasibility Study, Phase II–Preliminary Design, Phase III–Detailed Design, Phase IV–Planning for Production Process, Phase V–Planning for Distribution, Phase VI–Planning for Consumption, Phase VII–Planning for Retirement of the Product.

The methodology was upgraded and applied considering the specifics of hospital environment. The main emphasis will be on the first phase, i.e. Step 1: Definition of needs, demands and conditions, Step 2–Design problem, and Step 3–Synthesis of the possible solutions. Outputs of feasibility study present the basis for Phase II–Preliminary Design (Figure 7) (Asimow, 1962).

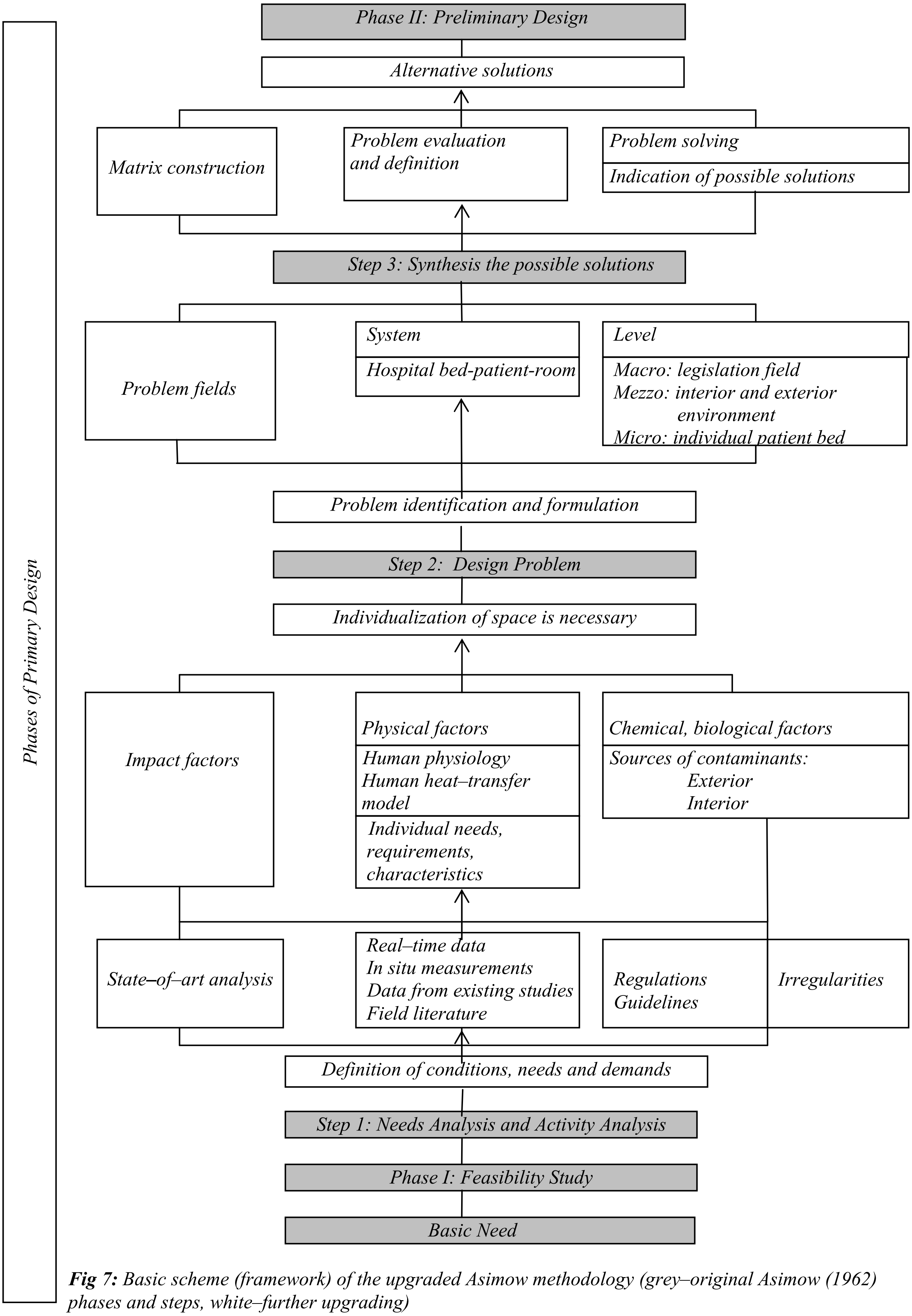

Fig 7: *Basic scheme (framework) of the upgraded Asimow methodology (grey–original Asimow (1962) phases and steps, white–further upgrading)*

The upgraded method comprises of the following steps:

1. Definition of conditions, demands and needs (State–of–art analysis):
- To certify the existence of the basic need for individualization of personal space: Is individualization necessary or could environment be regulated on general level as until now?
- With the research of actual conditions in the healthcare environment within regulated demands and recommendations, and their comparison, it is concluded that current situation is not satisfactory.

2. After the state–of art survey a large number of impact factors can be identified:
- Many of them cannot be precisely defined due to their interaction with other influences.
- They can be classified into three main groups: physical, chemical and biological factors.
- The number of groups can be expanded with noise factor and lighting factor groups, which can be presented in the same manner as other groups.

3. Problem identification and formulation:
- Identification of problem includes searching for their causes and consequences; their activities are focused on the future and past (retrospective).
- Quality of identification depends on the quality of previous steps, especially on state–of–art analysis that indicates the sources of problems and detects possible irregularities.
- For better manipulation of problems, the identified problems are disposed into three problem fields (macro, mezzo and micro level).

4. Synthesis of possible solutions:
- The next step of the developed method is a matrix where all impact factors with interacting influences can be systematically unfolded and identified.
- Designed matrix enables the problem definition and consecutive systematic and unburdened search for solutions.

5. Selection of the final solution and the design of actively regulated thermal comfort zone and required conditions for individual user.

Regarding the results of the proposed methodology, the impact factors are divided into principal groups for influential factor evaluation, problem identification and indication of problem solving directions. The design and application of matrices enable holistic consideration of impact factors jointly with their interactions during all the phases of the engineering design.

6.1.4 Plan of feasibility study

6.1.4.1 Step 1: Definition of needs, demands and conditions

The existence of the basic need for individualization of personal space has to be identified. Does the individualization represent the potential or could environment be regulated on general level as so far? Needs, demands and conditions required for user comfort can be obtained optimally by analyzing real state conditions. Actual conditions can be confronted to regulations and guidelines. At that level, irregularities and health impact can be detected and also possible solutions can be defined. State–of–art analysis is focused on searching for the data from the most relevant literature, regarding time, location (Slovenia, EU, world), type of literature (articles, studies, statistic data, regulations, guidelines, standards, manuals, scientific reports, monographs, etc.) and research filed (individualization, physical, chemical and biological impact factors).

We searched Pub Med and Science Direct for peer–reviewed publications from 1934 to 2012 written in English, with the keywords “hospital”, “hospital ward”, “patient room”, “operation room” together with “hazards”, “thermal comfort”, “air temperature”, ”operative temperature”, ”surface temperature”, “air velocity”, “relative humidity”, ”metabolic rate”, ”clothing insulation”, ”ventilation”, “hospital acquired infections”, “surgical side infections”, ”nosocomial infections”, “airborne infections”, “microorganisms”, “microbiological contamination”, “indoor air quality”, “waste anaesthetic gases”, “personal climate”, “local ventilation”, and "burn patient". Titles, abstracts, or both, of all articles were reviewed to assess relevance. We reviewed guidelines of Centers for Disease Control and Prevention, Communicable Disease Surveillance Centre, American Institute of Architects; ISO standards; manuals and handbooks of American Society for Heating, Refrigerating, and Air–Conditioning Engineers, Occupational Safety and Health Administration, US Department of Labor, WHO; reports of Health Protection Agency, American Academy of Pediatrics, National Institute for Occupational Safety and Health. We also searched the above keywords with the Google search engine.

The data will be organized into tables presenting the created database. In the field of individualization, the most relevant data important for users will be searched and the required parameters of comfort will be defined. For example: evidenced optimal air temperature for burn patients is 32 °C db, relative humidity of indoor air (RH_{in}) 95 % (ASHRAE Handbook HVAC applications …, 2007). After that, the selection of impact factors will follow. On the level of selected data, three matrices will be constructed, i.e. the matrices of physical, chemical and biological factors (Table 1).

Table 1: *Activity procedure in Step 1 of feasibility study*

Plan of feasibility study	
Step 1: Needs and activity analyses: Definition of needs, demands and conditions	
1.0	*Analyses of real state conditions*
1.1	*Investigation of in situ measurements, data from existing studies and filed literature*
1.2	*Comparison of actual conditions with regulations and guidelines*
2.0	*Definition of specific human needs that have to be satisfied on the level of "individual patient bed in the hospital environment"*
2.1	*Determination of the influences of defined parameters on patient, visitors and personnel*
2.2	*Investigation of the model of heat transfer, human physiology, individual needs and characteristics*
3.0	*Definition of physical factors (microclimate), chemical and biological factors in HE, resulting from interactions among man, patient room and bed*
4.0	*Selection of relevant factors for the matrix and their construction*

6.1.4.2 Step 2: Design problem

Design and application of matrix enables holistic consideration of impact factors and their interactions in all phases of engineering design. In matrices, all problems could be detected though interactions among selected parameters. In the basic matrix, they are arranged into columns and rows. The matrix can have more dimensions.

Table 2: *Activity procedure in Step 2 of feasibility study*

Plan of feasibility study	
Step 2: Design problem	
2.0	*Design of the problem arising from relation patient–patient bed–hospital environment*
2.1	*Detection of the problem on macro, mezzo and micro level on the basis of real–time conditions*
2.2	*Classification of the problem on macro, mezzo and micro level*

6.1.4.3 Step 3: Synthesis of the possible solutions

The doctoral dissertation is focused on searching for alternative solutions for detected problem in the matrix of physical factors. A detected problem is high building energy use for heating and cooling. The goal of the doctoral dissertation is to solve the problem holistically, i.e. from the analysis of building envelope at different locations to effective H/C system. It tries to find such a H/C system that enables attaining optimal conditions for individual user (required conditions for patient and thermal comfort conditions for healthcare worker and visitor), designed (minimal possible) human body exergy consumption rate valid for thermoregulation and simultaneously minimal energy use for heating and cooling.

Optimal conditions present the required room conditions for patient (required and recommended air temperature and relative humidity for patient room) and thermal comfort conditions for healthcare worker and visitor.

First, the analysis of the whole heating system with building envelope characteristics positioned at different locations is carried out. With this respect, space heating energy demand and exergy consumption rate are calculated. For experimental part, two H/C systems are selected, conventional and LowEx system. Systems were installed into the test room and are compared regarding human body exergy consumption rate, *PMV* index and measured energy use for heating and cooling of active space (Table 3).

Table 3: *Activity procedure in Step 3 of feasibility study*

Plan of feasibility study	
Step 3: Synthesis of the possible solutions	
3.0	*Searching for alternative solutions for detached problem in the matrix of physical factors*
3.1	*Interaction among parameters of high building energy use and human body exergy balance*
3.2	*Comparison between LowEx system and conventional system*
3.3	*Final solution and design of actively regulated thermal comfort zone in relation to required conditions for individual user*

For synthesis of the possible solutions, two models are jointly used: human body exergy balance model and model of building energy use. Human body exergy balance model was developed by Shukuya et al. (2010) and further on upgraded into spreadsheet software for the calculation of human body exergy consumption rate (Iwamatsu and Asada, 2009). The calculation procedure is presented in detail in Annex B, while their comparison with energy analysis is presented in Annex C. Human body is treated as a thermodynamic system based on exergy–entropy processes. The system consists of a core and a shell and is situated in a test room with environmental temperature. The general form of the exergy balance equation for a human body as a system is represented in Eq. (1) (Shukuya et al., 2010):

$$[Exergy\ input]-[Exergy\ consumption]=[Exergy\ stored]+[Exergy\ output] \tag{1}$$

The exergy input consists of five components: 1) warm exergy generated by metabolism; 2) warm/cool and wet/dry exergies of the inhaled humid air; 3) warm and wet exergies of the liquid water generated in the core by metabolism; 4) warm/cool and wet/dry exergies of the sum of liquid water generated in the shell by metabolism and dry air to let the liquid water disperse; 5) warm/cool radiant exergy absorbed by the whole skin and clothing surfaces. The exergy output consists of four components: 1) warm and wet exergy contained in the exhaled humid air; 2) warm/cool and wet/dry exergy contained in resultant humid air containing the evaporated sweat; 3) warm/cool radiant exergy discharged from the whole skin and clothing surfaces; and 4) warm/cool exergy transferred by convection from the whole skin and clothing surfaces into surrounding air (Shukuya et al., 2010). To maintain healthy conditions, it is important that the exergy consumption and stored exergy are at optimal values with a rational combination of exergy input and output.[1–4]

The procedure of the calculation of annual energy use for heating is based on the model of energy balance according to ISO 13790, presented in Annex D. The model includes calculation of heat losses, calculation of heat gains and calculation of balance, which means energy deficit that is necessary for heating. Solar gains, interior heat sources, efficiency factor and losses via transmission and ventilation are considered. The calculation of annual energy use for heating is based on seasonal or monthly method, zoning and includes occupied or unoccupied periods (ISO 13790).

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[1] Warm radiant exergy, in: exergy transfer by radiation is warm in case $T_{surf} > T_{ao}$, the exergy is absorbed by the whole skin and clothing surfaces. Cool radiant exergy, in: exergy transfer by radiation is cool in case $T_{surf} < T_{ao}$, the exergy is absorbed by the whole skin and clothing surfaces (for our calculation $T_{ao} = T_{ai}$, $T_{surf} = T_{mr}$) (Shukuya et al., 2010).

[2] Wet exergy: Moist air system contains wet exergy, if vapour pressure of moist air system p_{vr} is higher than vapour pressure of outdoor air p_{vo} (for our calculation $p_{vr} = p_{vo}$). Dry exergy: Moist air system contains dry exergy, if vapour pressure of moist air system p_{vr} is lower than vapour pressure of outdoor air p_{vo} (for our calculation $p_{vr} = p_{vo}$) (Shukuya et al., 2010).

[3] There are six cases of warm/cool and outgoing/incoming convective exergies, depending on the conditions among the T_{cl}, T_{ai}, T_{ao}. Besides that, clothing temperature T_{cl} varies with the conditions of six variables: T_{mr}, T_{ai}, v_a, effective clothing insulation, metabolic rate. Warm convective exergy, in: exergy transfer by convection is warm in case $T_{ao} < T_{cl} < T_{ai}$, the exergy is incoming onto the human body (from the surrounding air onto the whole skin and clothing surfaces). Warm convective exergy, out: exergy transfer by convection is warm in cases $T_{ao} < T_{ai} < T_{cl}$ and $T_{ai} < T_{ao} < T_{cl}$, the exergy is outgoing from the human body (from the whole skin and clothing surfaces into the surrounding air). Cool convective exergy, in: exergy transfer by convection is cool in case $T_{ai} < T_{cl} < T_{ao}$, the exergy is incoming onto the human body (from the surrounding air onto the whole skin and clothing surfaces). Cool convective exergy, out: exergy transfer by convection is cool in cases $T_{cl} < T_{ao} < T_{ai}$ and $T_{cl} < T_{ai} < T_{ao}$, the exergy is outgoing from the human body (from the whole skin and clothing surfaces into the surrounding air) (for our calculation we assume $T_{ao} = T_{ai}$) (Shukuya et al., 2010).

[4] Warm radiant exergy, out: exergy transfer by radiation is warm in case $T_{ao} < T_{cl}$, the exergy is discharged from the whole skin and clothing surfaces (for our calculation we assume $T_{ao} = T_{ai}$). Cool radiant exergy, out: exergy transfer by radiation is cool in case $T_{ao} > T_{cl}$, the exergy is discharged from the whole skin and clothing surfaces (for our calculation we assume $T_{ao} = T_{ai}$) (Shukuya et al., 2010).

The doctoral dissertation presents innovative design of H/C systems for individual user in hospital environment. It includes hierarchical procedure of solving problems: from the analysis of building envelope characteristics at different locations, to evaluation of indoor conditions and thermal comfort of their users. Analysis of human body exergy balance is linked together with building energy use for heating and cooling. Analyses are taken in normal and extreme room conditions, for average and individual persons. Thermal comfort conditions are separated from the required conditions for patients.

6.2 Experimental settings

6.2.1 Space geometry

Experimental part of the doctoral dissertation represents continuation of the developed method presented in section 6.1.4.3 and it is part of Step 3: Synthesis of the possible solutions. Because of specifics of hospital environment and ethic codex, the combination of experiment and calculations in test environment was carried out.

Experimental part of the doctoral dissertation was carried out in real test room and in simulated model room for burn patients. The real test room is located at the Chair for Buildings and Constructional Complexes, Faculty of Civil and Geodetic Engineering, University of Ljubljana (UL FGG, KSKE) (Figures 8–10). The model room for burn patient was used for calculations of space heating energy demand, exergy consumption patterns and thermal comfort calculations. The model room had the same room dimensions and composition of constructional complexes as the real test room.

Fig 8: *Test room, UL FGG, KSKE*

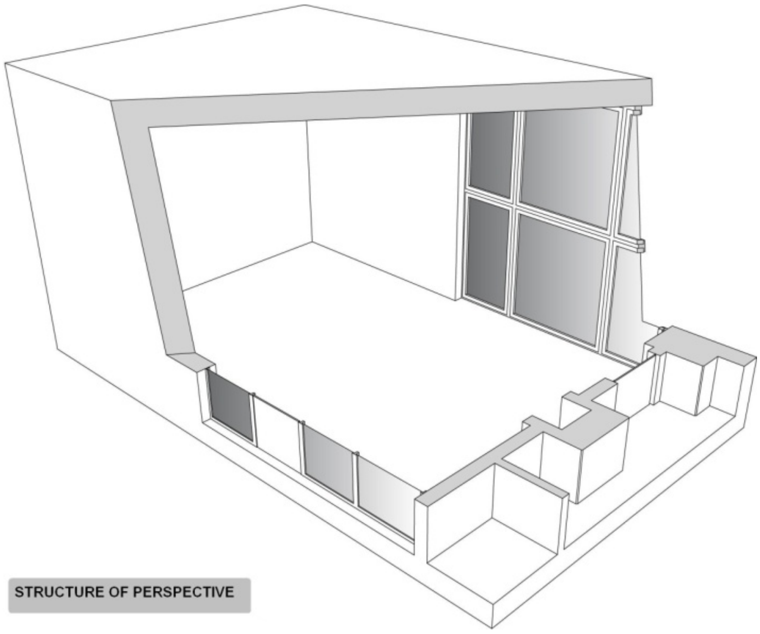


Fig 9: *Perspective view of the test room*

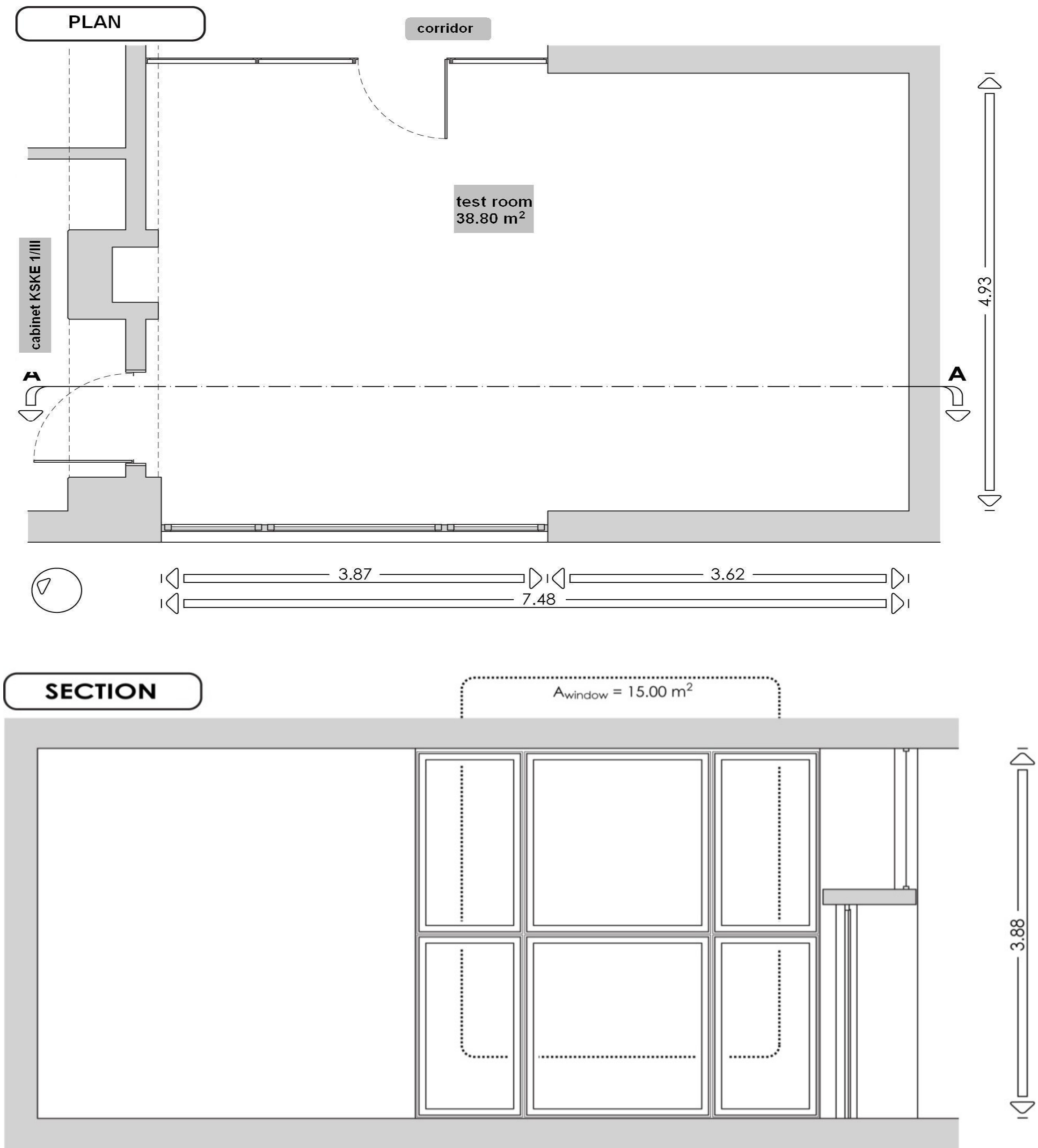


Fig 10: *Plan and section of test room*

6.2.2 Constructional complexes

Composition of non–transparent and transparent constructional complexes of the test room is presented in Tables 4 and 5.

Table 4: *Characteristics and composition of non–transparent constructional complexes*

	d [m]	***λ [W/(mK)]***	***R_l [m^{2}K/W]***	***U [W/(m^{2}K)]***
CEILING, FLOOR CONSTRUCTION				*0.830*
R_i			*0.170*	
Linoleum	*0.003*	*0.190*	*0.015*	
Concrete topping	*0.050*	*1.60*	*0.043*	
Acoustic insulation	*0.010*	*0.041*	*0.244*	
Concrete slab	*0.050*	*2.040*	*0.025*	
Super 40	*0.400*	*0.610*	*0.656*	
Plaster	*0.010*	*0.990*	*0.010*	
R_e			*0.040*	
EXTERIOR WALL				*1.290*
R_i			*0.130*	
Plaster	*0.010*	*0.870*	*0.011*	
Lightweight concrete blocks	*0.200*	*0.350*	*0.571*	
Concrete façade plates	*0.050*	*2.040*	*0.025*	
R_e			*0.040*	
INTERIOR WALL				*1.170*
R_i			*0.130*	
Plaster	*0.010*	*0.870*	*0.011*	
Lightweight concrete blocks	*0.200*	*0.350*	*0.570*	
Plaster	*0.010*	*0.870*	*0.011*	
R_e			*0.130*	

Table 5: *Characteristics and composition of transparent constructional complexes*

	$\tau_{ao\ v}$ [ ]	***U [W/(m^{2}K)]***
WINDOW	*0.80*	*2.90*
Glass		
Air layer		
Glass		
Al frame		
INTERIOR WINDOW	*0.85*	*5.90*
Glass		
Wooden/steel frame		

6.2.3 Heating–cooling system

The test room was equipped with both H/C systems with time separation (conventional and LowEx system). The position of LowEx system and conventional system is presented in Figure 11.

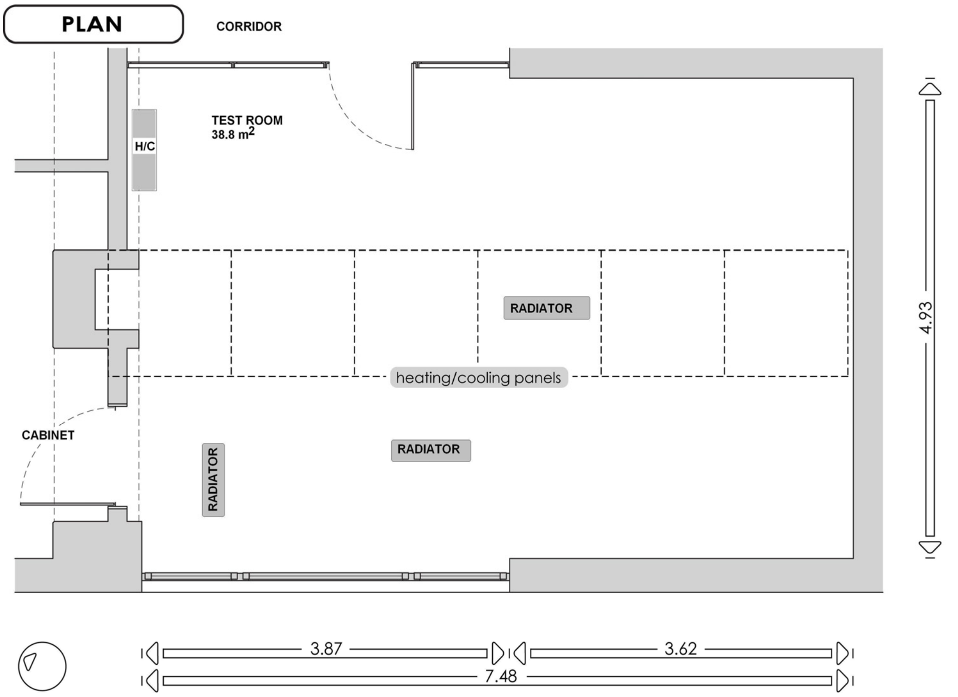


Fig 11: *Plan of test room with positions of LowEx system (heating and cooling panels) and conventional system (split system with indoor unit and oil–filled electric room radiators)*

Conventional system presents three oil–filled electric heaters and split system with indoor unit (Figure 12).

Fig 12: *Indoor unit*

Oil–filled electric heaters are type Heller (230 V – 50 Hz, 2000 W) and enables automatic temperature regulation with six heating stages and ventilating fan (Table 6). H/C split–system has two connected blocks: internal and external. They are connected with pipes and electric wires. External blocks present two cooling aggregates, mounted on the top of the roof of the faculty. The characteristics of cooling aggregates are presented in Table 7. Cooled water (9 °C) is distributed with pipe system into floors and cabinets. Internal block presents wall−type room conditioner, mounted on the top of the south wall (indoor unit). It enables automatic temperature regulation. Water temperature on the entry valve is 10 °C (Table 7). In burn patient room air is heated with heat lamps and 3000 W radiant heaters (Arturson et al., 1978; Wallace et al. 1994; Herndon, 1996: 140). Infrared radiation is a practical but very inexpensive way of distributing energy from the environment to the patient (Arturson et al., 1978). For our purposes, oil–filled electric heaters are used.

Table 6: *Characteristics of heating system*

Oil–filled electric heater (No)	*3*
Type	*Heller*
Model	*HRO 2009*
Power	*2000 W*
Voltage	*230 – 50 Hz*

Table 7: *Characteristics of cooling system*

Cooling aggregate *(No)*	*2*
Type	*Ecoflamclima*
Model	*COD.653188–A*
Cooling capacity	*30.9 kW*
Heating capacity	*35.3 kW*
Refrigerant	*R407*
Refrigerant change	*9.80 kg*
Max pressure	*28 – 20 bar*
Power supply	*400–3N~50V~Hz*
Power supply aix circuit	*23°~50V~Hz*
Full load motor input fli	*13.30 kW*
Full load motor amper fla	*26.9A*

to be continued…

…continuation

Cooling aggregate *(No)*	*2*
Lock rotor ampere era	*163.0 A*
Level protection	*IPX4*
Wiring diagram	*5478002600 N°*
Weight in operation	*330 kg*
Manufacture year	*2002*
Indoor unit *(No)*	*1*
Type	*Ecoflamclima*
Cooling power	*2 kW*
Manufacture year	*1998*
Medium	*Water*
T_{water} *(entry valve)*	*10 °C*

LowEx system presents six low–temperature–heating and high–temperature–cooling ceiling radiative panels. Panels are positioned 72 cm under the ceiling or 3.16 m above the floor, along in the centre of the ceiling. For the purpose of our experiment 9 m^2 of ceiling are covered with panels (Figure 13).

Fig 13: *Heating and cooling panels*

Dimensions of heating–cooling ceiling panels are 125 cm by 125 cm. They are constructed of 10 cm tick polystyrene thermal insulation with engraved pipes for hot or cold water (Figure 14). The final layer of four panels is contact stuck 1.25 cm gypsum board, and two panels have stone plates (one is compact 2 cm marble plate and another composite 12 mm Al–honeycomb with 3 mm tick stone plate). All panels are fixed with four steel screws into ceiling construction. Panels are connected into H/C system with valves for switching off every panel separately, and with a pump on thermostatic mixing valve. Switching between hot and cool water entering into the panels is manual (Figures 15, 16).

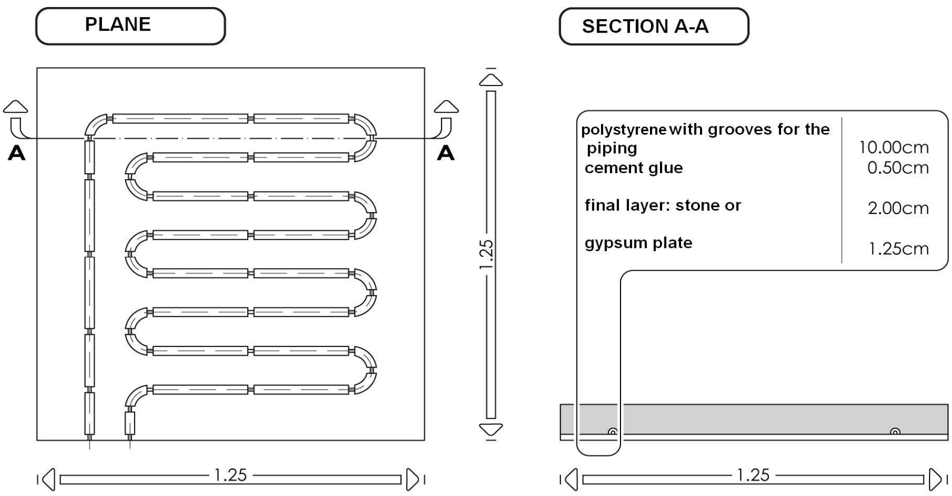


Fig 14: *Plan and section of heating–cooling LowEx panels. In the plan the grooves for the piping are shown*

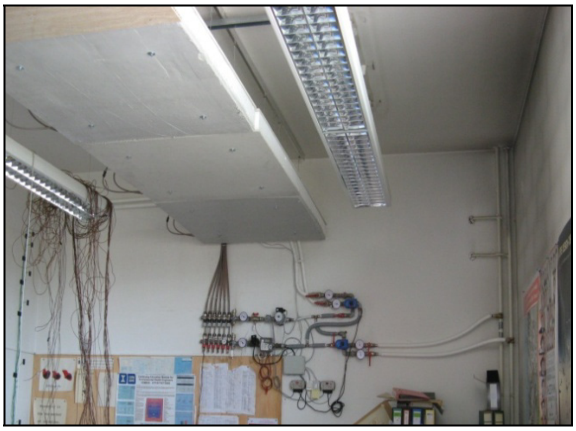


Fig 15: *Panels with control–measuring system*

Fig 16: *Panel assembly*

6.2.4 Control and monitoring system

For the purpose of on–line monitoring and control of indoor parameters, an integrated control system of internal environment on the basis of fuzzy logic (hereinafter ICSIE) was used. ICSIE system was developed by Trobec–Lah (2003) and upgraded by Košir et al. (2006) and Košir (2008). It enables the control of indoor air temperature, CO_2 and illuminance under the influence of outdoor environment and users` requests. The ICSIE system is divided in three parts: sensor network system, regulation system and actuator system. The basic architecture of the system is presented in Figure 17.

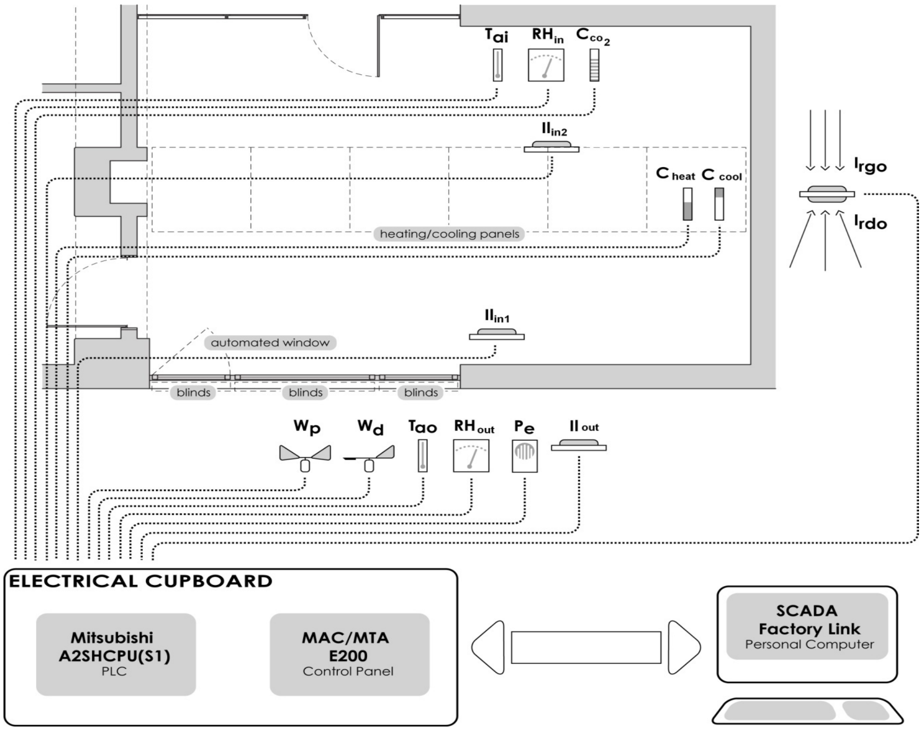


Fig 17: *Basic architecture of the ICSIE system. The presented sensor array consists of the following sensors:* T_{ai}*–room air temperature,* T_{ao}*–outdoor air temperature,* RH_{in}*–relative humidity of indoor air,* RH_{out}*–relative humidity of outdoor air,* Il_{in1} *&* Il_{in2}*–internal work plane illumination (workplace 1 and 2),* Il_{out}*–external illumination,* C_{CO2}*–concentration of* CO_2*,* I_{rgo}*–direct solar radiation,* I_{rdo}*–reflected solar radiation,* W_p*–wind speed,* W_d*–wind direction,* P_e*–precipitation detection,* C_{heat}*–energy use for heating,* C_{cool}*–energy use for cooling*

Fig 18: *Building management system, hardware*

Fig 19: *Building management system, software*

Sensor network system enables recording of room air temperature (T_{ai}), outdoor air temperature (T_{ao}), relative humidity of indoor air (RH_{in}), relative humidity of outdoor air (RH_{out}), internal surface temperatures, black globe temperatures, temperature of medium in panels, internal work plane illumination (workplace 1 and 2), external illumination, concentration of CO_2, direct solar radiation, reflected solar radiation, wind speed, wind direction, precipitation detection, energy use for heating, energy use for cooling. The selected time step was 0.5 s. The selected parameters for monitoring were: T_{ai}, T_{ao}, RH_{in}, RH_{out}, surface temperatures, black globe temperature, temperature of the medium in panels, energy use for heating and cooling (Figures 20–26).

Fig 20: *Outdoor sensors*

Fig 21: *Black globe thermometer and data logger*

Fig 22: *T_{ai}, RH_{in} sensor*

Fig 23: *Thermocouples on panels and ceiling*

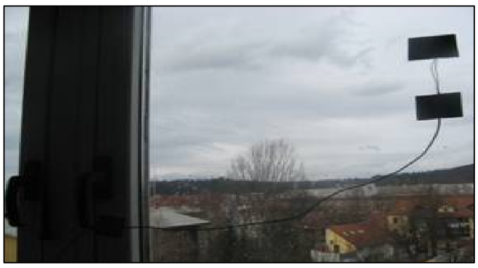


Fig 24: *Thermocouples on window*

Fig 25: *Ultrasonic compact heat and cooling meter*

Fig 26: *Electricity counter*

The characteristics of the measurement equipment for selected parameters are presented in Table 8 and Table 9.

Table 8: *Measurement equipment characteristic for microclimate parameters*

Microclimatic parameter/ measurement equipment	T_{ao}, RH_{out}	T_{ai}, RH_{in}	T_{ai}, RH_{in}	T_{ai}
	Sensor	***Sensor***	***Data logger***	***Thermocouples***
Manufacturer	*E+E*	*E+E*	*Hangzhou Ontime Electronic Technology Co.*	*/*
Type/model	*EE21–FT3A25/T02*	*EE80–2CTF3*	*HD206–1*	*K, T*
Working range	*T: –40 °C… +60 °C RH: 0…100 %*	*T: 0 °C… +50 °C RH: 0…100 %*	*T: – 30 °C… +80 °C RH: 5…98 % Resolution 0.1°C; 0.1 % RH*	*K: –200 °C… +1250 °C T: – 200 °C… +350 °C*
Accuracy	*T: ±0.7°C at – 40 °C, ±0.2°C at +20 °C RH: ±3 % at 30–70 % ±5 % at 10–90 % RH*	*T: ±0.3 °C at 0–50 °C RH: ±3 % at 30–70 % RH, ±5 % at 10–90 % RH*	*T: ±0.4 °C RH: ±2.5 %*	*K: 0.2 °C or 0.75 % above 0 °C, 2.2 °C or 2.0 % below 0 °C T: 1.0 °C or 0.75 % above 0 °C, 1.0 °C or 1.5 % below 0 °C*

Table 9: *Measurement equipment characteristic for energy use*

Ultrasonic compact heat and cooling meter	
Manufacturer	*Allmess GmbH*
Type/model	*CF Echo II 2009–09814368 EN 1434 CLASS 3*
Optical interface	*M–BUS/EN60870–5*
Voltage	*230 V, 50 Hz, 8 VA*
Temperature range	*0…180 °C Return 5…130 °C 3…100 K*
Cable, P_t	*100*
Flow (nominative q_p***, max–***q_s***, min–***q_i***)***	$q_p = 2.5\ m^3/h$, $q_s = 5.0\ m^3/h$, $q_i = 0.025\ m^3/h$

Energy use for heating and cooling, T_{ao}, T_{ai}, T_{mr}, RH_{in} and RH_{out} were continuously monitored for both systems with ICSIE system. Sensor positions are presented in Figure 27. Individual thermal comfort conditions were analyzed by calculated human body exergy consumption rate valid for thermoregulation and predicted mean votes (PMV) index[1] with spreadsheet software developed by Hideo Asada Rev 2010 (Iwamatsu and Asada, 2009).

For the calculation of human body exergy consumption rate in a room with conventional system the conditions with T_{ai} equal to T_{mr} were assumed and RH_{in} varies in the range 30–96 % (Figures 28–31). In a room with LowEx system T_{ai} differed from T_{mr} and RH_{in} was constant 80 % (recommended value, Herndon, 1996: 492). For both systems, the reference environmental temperature (the outdoor environmental temperature) and RH_{out} were assumed to be equal to T_{ai} and RH_{in}.

[1] Predicted mean vote (PMV) is an index that predicts the mean value of the votes of a large group of persons on the 7–point thermal sensation scale (+3 hot, +2 warm, +1 slightly warm, 0 neutral, –1 slightly cool, –2 cool, –3 cold), based on the heat balance of the human body. Thermal balance is obtained when the internal heat production in the body is equal to the loss of heat to the environment (ISO 7730). By setting PMV = 0 predicts combinations of activity, clothing and environmental parameters which on average will provide a thermally neutral sensation.

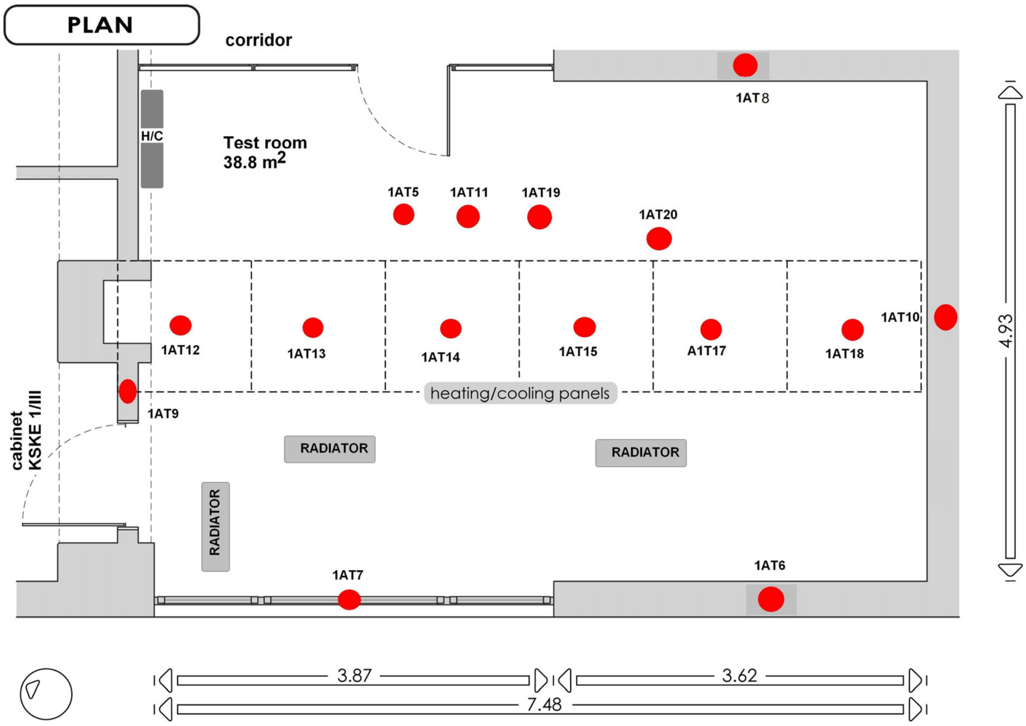


Fig 27: Sensor position

Fig 28: Humidifiers

Fig 29: Humidification

Fig 30: Humidification

Fig 31: Humidification

6.2.5 Subject characteristics and experimental conditions

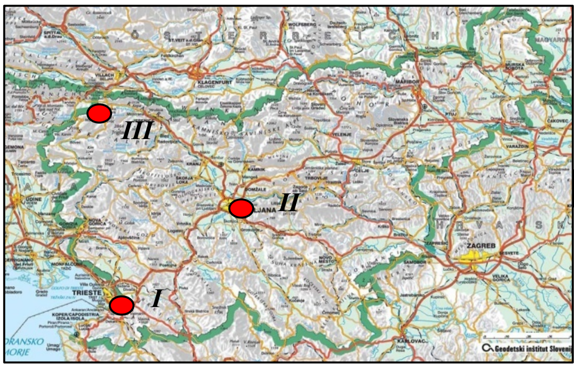
Test room was equipped with LowEx system and conventional system with time separation where T_{ai} and T_{mr} differed in the range of 15–35 °C and relative humidity changed from 30 to 96 %.

Three test subjects (burn patient, healthcare worker, visitor) were assumed for the calculation of human body exergy balance. Individual input data and experimental conditions were collected from the relevant literature (Cope et al., 1953; Caldwell et al., 1966; Fanger, 1970; Zawacki et al., 1970; Wilmore et al., 1973a, 1973b, 1974a, 1974b, 1975; Neely et al., 1974; Caldwell, 1976, 1991; Danielsson et al., 1976; ASHRAE Handbook & product …, 1977; Arturson et al., 1978; Caldwell et al., 1981, 1992; Herndon, 1981, 1996; Herndon et al., 1987a, 1987b, 1988; Cone et al., 1988; Simmers, 1988; Carlson at al., 1992; Fleming et al., 1992; Herndon and Parks, 1992; Martin et al., 1992; Shiozaki et al., 1993; Wallace et al., 1994; Kelemen et al., 1996; Wedler et al., 1999; LeDuc et al., 2002; Ramos et al., 2002; Herndon and Tompkins, 2004; Skoog et al., 2005; Atmaca and Yigit, 2006; ASHRAE Handbook HVAC applications …, 2007; Corralo et al., 2007; ISO 7730; ISO 8996; Sudoł–Szopińska and Tarnowski, 2007) and are presented in Table 10. According to the literature (Cope et al., 1953; Caldwell et al., 1966; Zawacki et al., 1970; Wilmore et al., 1973a, 1973b, 1974a, 1974b, 1975; Neely et al., 1974; Caldwell, 1976, 1991; Danielsson et al., 1976; Arturson et al., 1978; Caldwell et al., 1981, 1992; Herndon, 1981, 1996; Herndon et al., 1987a, 1987b, 1988; Cone et al., 1988; Carlson at al., 1992; Fleming et al., 1992; Herndon and Parks, 1992; Martin et al., 1992; Shiozaki et al., 1993; Wallace et al., 1994; Kelemen et al., 1996; Wedler et al., 1999; Ramos et al., 2002; Herndon and Tompkins, 2004; Atmaca and Yigit, 2006; ASHRAE Handbook HVAC applications …, 2007; Corralo et al., 2007) a burn patient with 80 % TBSA, hypermetabolic state (2 met = 116 W/m^2 of body surface) and hypothermia (T_{cr} 35.5 °C, T_{sk} 37.0 °C) was assumed. For the calculation of human body exergy balance T_{sk} and T_{cr} were constant for burn patient, and varied in case of visitor and healthcare worker. T_{cl} was calculated considering experimental conditions.

Table 10: *Subject characteristics and experimental conditions*

Subject characteristics					
Subjects	***Metabolic rate [met]***	***Effective clothing insulation [clo]***	T_{cr} **[°C]**	T_{sk} **[°C]**	T_{cl} **[°C]**
Healthcare worker	*1.1*	*0.6*	*36.8–37.2*	*29.5–36.8*	*23.3–36.2*
Burn patient	*2*	*0*	*35.5*	*37.0*	*25.3–37.1*
Visitor	*2*	*0.6*	*36.9–38.0*	*31.9–38.6*	*23.3–36.4*
Experimental conditions					
RH_{in} [1] **[%]**	v_a [1] **[m/s]**	T_{ai} [1] **[°C]**	T_{mr} [1] **[°C]**		
30–96	*0.1*	*15.0–35.0*	*15.0–35.0*		

The scheme for experimental work procedure is presented in Figures 32 and 33.



Calculation of annual energy use for heating [MJ/a] = ?
Conversion in average heating energy demand [W] = ?
Exergy consumption rate of the whole heating system [W] = ?
Radiant exergy flow rate from building wall [mW/m²] = ?
Thermal exergy flow rate [W/m²] = ?
Heat flow rate [W/m²] = ?

Fig 32: *The influence of site characteristics and building envelope on average heating energy demand and exergy consumption rate of a whole heating system. Red marks present three locations: Koper (Zone I, Mediterranean), Ljubljana (Zone II Continental), Rateče (Zone III, Alps)*

[1] For all calculations T_{ai} and T_{mr} were assumed to be equal. T_{ao} and RH_{out} were equal to T_{ai} and RH_{in}.

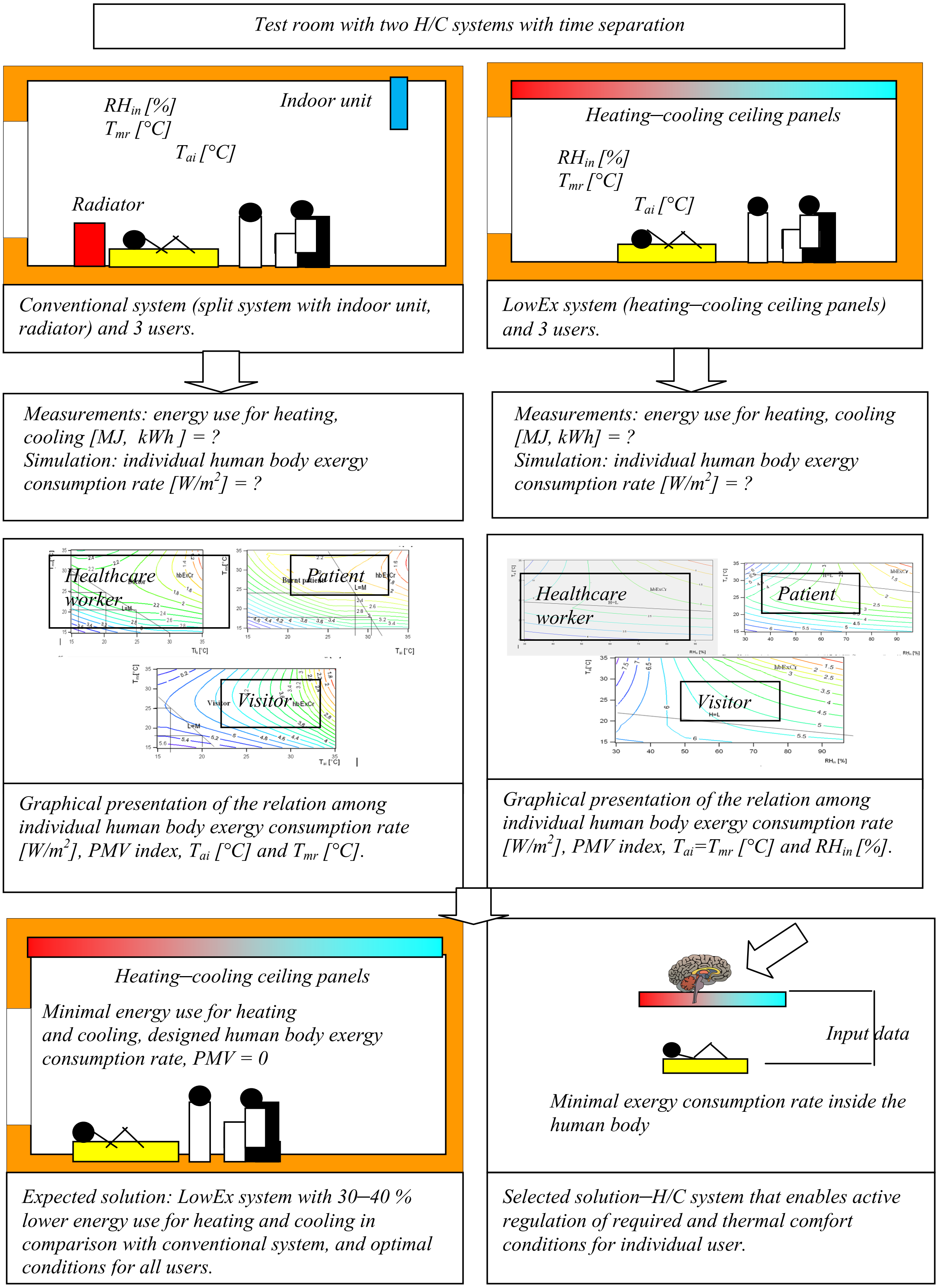


Fig 33: *Scheme of experimental work procedure*

7 RESULTS AND DISCUSSION

Results are divided into three parts. In the first part, all results of feasibility study that enables selection of the H/C system are presented. The second part presents the results of analysis of exergy consumption patterns for space heating. The third part presents the results of systems` comparison: calculated human body exergy consumption rate, *PMV* index for average test subject and individual test subject exposed to normal and extreme conditions, as well as measured energy use for heating and cooling. At the end, active regulation of required and thermal comfort conditions for individual user is presented.

7.1 Results of Feasibility study

7.1.1 Analyses of real state conditions, results of step 1

State–of–art analysis was focused on searching for data from the most relevant literature, location (Slovenia, EU, world), type of literature (articles, studies, statistic data, regulations, guidelines, standards, manuals, scientific reports, monographs, etc.). 721 sources of literature on the level of individualization, physical factors, chemical factors and biological factors were analyzed (Table 11). The list of references for the analyses of real state conditions is presented in Annex E.

Table 11: *Number of recorded references regarding literature type and research field up to now*

Research field /Literature type	*Individualization*	*Physical factors*	*Chemical factors*	*Biological factors*	*Total*
Article	*58*	*118*	*90*	*277*	*543*
Monograph	*7*	*17*	*1*	*22*	*47*
Guidelines	*1*	*13*	*3*	*15*	*32*
Regulation, Directive	*0*	*14*	*0*	*21*	*35*
Manual	*3*	*8*	*2*	*4*	*17*
Standard	*2*	*11*	*2*	*3*	*18*
Scientific Report	*5*	*8*	*1*	*3*	*17*
Other	*0*	*9*	*0*	*0*	*9*
Instructions	*0*	*0*	*0*	*3*	*3*
Total	*76*	*198*	*99*	*348*	*721*

The relevant data on physical, chemical, biological factors and individualization were organized into tables that present the databank. The databank is archived at UL, FGG, KSKE. Columns of the tables present real state conditions, demand–recommendations, health impacts and cause–actions, while rows present relevant parameter. Dimensions of the tables differ due to the number of recorded references and research field (pages: 14–57, words: 4098–19750, paragraphs: 321–1414, lines: 1134–4365).

Table 12 presents an example of table (cutting from the databank) of physical factors for RH_{in} as a relevant parameter. In such manner tables for other parameters, such as room air temperature, relative humidity, air flow rates, surface temperature; ventilation, pressurization, filtration; interactive impact among factors; release rate of thermal energy, etc., were created.

Table 12: *Example of the table of state–of–art analyses for physical factors: cutting from the databank. Original table has 14 pages, 4098 words, 321 paragraphs*

PHYSICAL FACTORS IN HOSPITAL ENVIRONMENT

Param	Real-time conditions	Regulation demands	Health impact	Cause-Actions
Relative humidity (RH, %) and humidity ratio (HR, g/kg DA)	**SICKROOM, WINTER, no humidifiers:** RH= <50% (av RH=32.8%, 29-37% 24h) mean (SD) of RH 32.8 (6.6)% HR=5g/kg DA (Hashiguchi et al., 2008) **SICKROOM, WINTER, humidifiers:** RH= <50%, (av RH=43.9%,< 40% 24h) mean (SD) of RH 43.9 (7.2)% HR<5 g/kg DA (Hashiguchi et al., 2008) **NURSE STATION, no humidifiers:** RH>30% 24h (Hashiguchi N. et al., 2008) mean (SD) of RH 33.1 (6.6)% HR=5 g/kg DA (Hashiguchi et al., 2008) **NURSE STATION, humidifiers:** mean (SD) of RH 36.9 (7.5) % HR<5 g/kg DA (Hashiguchi et al., 2008)	The law for the maintenance of sanitation in buildings in Japan states that the indoor relative humidity should be maintained at higher than 40% (Ogawa, 1999). Design manual limit 30-60% (ASHRAE HVAC design ..., 2003) 30–60% (Guidelines for design..., 1996) 45–55% (ASHRAE HVAC design ..., 2003) 30–60% (ANSI/ASHRAE/ASHE S 5 5 G 1 T fo	**Low RH:** **HOSPITAL ENVIRONMENT:** complaints of thermal discomfort and dryness of air among staff and patients, dry and itchy skin, static electricity (Hashiguchi et al., 2008) **INDOOR ENVIRONMENT:** Dehydration, irritation of skin, mucosa, eye, nose; stuffiness; electroshock effect at carpets; (Lee et al., 2000; Rayman, 1997); discomfort, symptoms (Nagda and Hodgson, 2001; Wyon et al., 2006); health problems (Arundel et al., 1986); RH<20% Japanese factory workers complained of dry air and	**Compromise:** Optimization for man, building and equipment, procedures. **Cause:** working of heating system in hospitals and nursing homes as well as offices, factories etc. (Matsui, 1981). The indoor air has to be humidified (Balaras et al., 2007). Stand-alone, console-type humidifiers that recirculate water for humidification should not be used because the water in these systems ith microorganisms ked to outbreaks of alaras et al., 2007). conditions the indoor air og, 2006).

- *Room air temperature, relative humidity, air flow rates, surface temperature;*
- *Ventilation, pressurization, filtration;*
- *Interactive impact among factors;*
- *Release rate of thermal energy, etc.*

Table 13 presents an example of the table (cutting from the databank) of chemical factors for anaesthetic gases, disinfection and sterilization substances as a relevant parameter. In such manner tables for other parameters, such as CO, CO_2, total volatile organic compounds (TVOC), etc., were created. In the field of chemical factors, also exterior pollutants are included.

Table 13: *Example of the table of state–of–art analyses for chemical factors: cutting from the databank. Original table has 15 pages, 5284 words, 419 paragraphs*

CHEMICAL FACTORS IN HOSPITAL ENVIRONMENT				
Param	Real-time conditions	Regulation demands	Health impact	Cause-Actions
Anaesthetic gases, disinfection and sterilizing substances	**Anesthetic gases:** General anesthesia is delivered using either gases or volatile liquids that are vaporized and inhaled with oxygen, medical air, nitrous oxide or a combination, and drugs delivered intravenously (Tortora et al., 2003; Doi et al., 2001). **HOSPITAL OPERATING ROOMS, HELLENIC ORS:** Chemicals in OR indoor air: waste medical gases used for anesthesia, disinfection, sterilizing substances, other particles (skin squames, lint, respiratory droplets, aerosols) (Pa... and Corn, 2002; Mierdl et al., 200... Commonly used inhalational anesthetics include: halothane ($CF_3CClBrH$), enflurane, isoflurane...	Number of air changes per hour (ACH)-it is widely recommended to use about 20 ACH (Balaras et al., 2007) **Halogenated anaesthetic agents** TLV (Waste anesthetic gases …, 2007), 8h (TWA): 2 ppm without contaminant N_2O exposure, 0.5 with contaminant N_2O exposure. **Nitrous oxide**: TLV for hospital environments (Waste anesthetic	Adverse health effects and discomfort have been detected to OR personnel by the occupational exposure to anaesthetic gases (e.g. nitrous protoxide and halogenated agents) and personnel exposure should be limited (Panni and Corn, 2002; Mierdl et al., 2003) Anesthetic gases must be removed because they may cause health problems or	Heating, ventilating and air-conditioning (HVAC) installations control indoor environmental quality (IEQ) and aseptic conditions, and secure healthy, safe and suitable indoor air quality (IAQ), for surgeons and medical staff, and of course, the patients (Balaras et al., 2007). Indoor air quality can improve considerably by using scavenging equipment to recover anaesthetic gases (Balaras et al., 2007). Establish a hazard communication program. Develop and implement a safety and health plan that includes information about exposure hazards and methods to control them.

- *Exterior and interior pollutants;*
- *CO, CO_2, TVOC, anaesthetic gases, disinfection substances, sterilization substances;*
- *The most frequent used disinfection substances and cleansers (trade name, active substance, application), etc.*

In the field of biological factors data about most frequent microorganisms, concentration, sources, transmission mode, risk factors, general and specific actions, reservoirs, normal microflora, specific areas, hand contamination, procedures and type of infection were organized into tables (cutting from the matrix) (Table 14).

Table 14: *Example of the table of state–of–art analyses for biological factors: cutting from the databank. Original table has 57 pages, 19750 words, 1414 paragraphs*

BIOLOGICAL FACTORS IN HOSPITAL ENVIRONMENT					
Param		Real-time conditions	Regulation demands	Health impact	Cause-Actions
Bacterial infections	G+ bacteria	**(a) S. aureus and MRSA (Methicillin-resistant *Staphylococcus aureus*):** At any one time approximately 30% of healthy people are carriers of S. aureus (Arbuthnott, 1992). MRSA infections are generally associated with person to-person contact. Notwithstanding this, airborne transmission of S. aureus and MRSA ...	Guidelines have been developed by the **Centers for Disease Control and Prevention (CDC)**, which recommend that hospitals 'thoroughly clean and disinfect environmental medical equipment surfaces on a regular basis' (Carling et al., 2008) . CDC guideline (Guideline for environmental …,	In the form of a bioaerosol can contaminate air and cause airborne infection (Solberg, 2000; Williams, 1966; Lidwell et al., 1975). MRSA was mainly carried on larger particles, 4±8 mm in size but some were on<4 mm particles, (respirable-reach the lung, possibly cause infection) (Shiomori	Personal hygiene of patients and staff, glows, protective clothes, hygiene of bed making, changing nappies, cleaning procedures of surfaces, equipment, rectal thermometers, etc.. CD: Hand hygiene, usage of gloves, cleaning of surfaces, with sporocide agents, isolation of asymptomatic patients; avoid the usage of rectal thermometers, lamination of AB during epidemics.

- *The most frequent microorganisms organized into groups (G+ and G–bacteria, viruses, fungus, protozoa), particulate matter (PM) and dust;*
- *Concentration, sources, transmission mode, risk factors, general and specific actions, reservoirs, normal microflora, specific areas, hand contamination, procedures, etc.;*
- *Type of infection (intestinal infection, infection via enteral food, endoscopic infections bacteriemia, urological infections, wound infections), etc.*

7.1.2 Definition of individual human needs, results of step 1

Data about individual human needs present an important source that has to be considered and analyzed regarding interactions among different parameters. It also presents the basis for the definition of subjects` characteristics and experimental conditions for the experimental part. In the field of individualization, all hazard groups of people are included: burn patients, women in labour, children, neonatal, hematological patients, cancer patients, elderly patients, dialysis patients, patients with urine catheter, frequent pathogens, risk factors, mode of transmission, actions, etc. (Table 15).

Table 15: *Example of the table of state–of–art analyses for individualization: burn patient. Original table has 18 pages, 5404 words, 622 paragraphs*

INDIVIDUALIZATION	Param	Real-time conditions	Regulation demands	Health impact	Cause-Actions
Burn patients	T_{sk}	"…the body try to raises the skin and core temperature for 2°C secondary to a hypothalamic reset…"(Herndon, 1996: 422) "…burn patient strive for temperatures of about 38°C…" (Herndon, 1996: 492)	"…ambient temperature and humidity should be maintained at 30-33 °C and 80% respectively, in order to decrease energy demands and evaporative heat losses…" (Herndon, 1996:492) "…the hypermetabolic response may be reduced by warming the ambient temperature to thermal neutrality (33°C), at which point the heat for evaporation is derived from the environment, taking the	"..specific subject from thermodynamic point of view " (Herndon and Parks, 1992). "…the worst trauma problems: larger a burn injury→more severe the consequences +the higher the chance of an adverse outcome/even death" (Herndon and Parks, 1992). "…in the US, approx. 1.4 million individuals/a sustain burn injuries Of this group, 54000	..."these changes caused by the burn wound can be altered by environmental, nutritional, and pharmacologic means… " (Cope et al., 1953; Epstein et al. 1963; Zawacki et al., 1970; Wilmore et al. 1973a, 1973b, 1974a, 1974b, 1975; Neely et al., 1974; Danielsson et al., 1976; Caldwell et al. 1981, 1992; Caldwell, 1991; Herndon et al., 1987a, 1987b, 1988; Herndon and Parks, 1992; Gore et al., 2005;
	T_{cr}	"…core temperature is generally expected to be 0.5°C higher than body surface temperatures…" (Herndon, 1996: 530) "…in normal individuals the threshold range is generally near 36.5°C-37.5°C. In burn patient the threshold set point is higher and the increase is proportionally to the size of the burn, 0.003°C / % total body size area regarding the size of the burns…" (Herndon, 1996:			

- *Women in labor, children, neonatal, hematological patients, cancer patients, patients, elderly patients, dialysis patients, patients with urine catheter, etc.;*
- *The frequent pathogens, risk factors, mode of transmission, actions, etc.*

Burn patient is from thermodynamic point of view a very specific subject. Burns are among the worst trauma problems, which can befall man. The larger a burn injury, the more severe are the consequences and the higher the chance of an adverse outcome or even death (Herndon and Parks, 1992). Each year in the United States, approximately 1.4 million individuals sustain burn injuries. Of this group, 54000 patients need hospitalization and ultimately about 5000 die from burn–related injuries (Health …, 1990; Mortality …, 2004). There is no detailed information about the present number of burn victims in Europe. Back in year 1995, 24986 patients (16800 adults and 8186 children) were hospitalized with burn injuries in Europe (24 countries with a total population of 512 million) (Wedler et al., 1999). In Slovenia the number of admissions was 32.69/100000 (Wedler et al., 1999). The Number of admitted patients at Ljubljana Burns Unit was 1821 in 1973–1976 and 762 in 1995–1998 (Janežič, 1999). The number of occupational burns registered by the Institute for Health Protection Republic of Slovenia (Number and percent …, 2010) for 2008 is 700. It includes thermal burns, chemical burns, effects of low temperatures and chilblains and presents 2.8 % of all occupational injuries regarding the nature of injuries (overall 25036).

The burn wound is a serious injury, progressive in nature, causing a myriad of effects far beyond its bounds (Herndon et al., 1987a; Herndon et al., 1988; Herndon and Parks, 1992), i.e. collateral influences. Much knowledge has been gleaned regarding its pathophysiology, yet more remains to be uncovered. Thermal injury results in significant pathophysiological changes that start with an initial critical phase, continue with an intermediate phase and end with a second critical phase. During intermediate phase, there appear metabolic changes that lead to increased metabolic rates, and higher or lower body core temperature due to reset thermostatic control centre in the hypothalamus (Herndon, 1996: 69; Ramos et al., 2002; Shiozaki et al., 1993). Hormonal and metabolic response[1,2] of burn injury leads to hypermetabolism, progressive weight loss, increased susceptibility to infection, and poor wound healing (Cope et al., 1953; Wilmore et al., 1973a, 1973b, 1974b, 1975; Danielsson et al., 1976; Arturson et al., 1978; Martin et al., 1992; Wallace et al., 1994). Burn patients are by far the most susceptible to intra– and post–operative hypothermia, since the damaged skin is no longer able to prevent the loss of body heat (Ramos et al., 2002). Study by Fleming et al. (1992) found out that increased ambient temperature attenuates hypermetabolism and increase patient comfort. Patients with extensive thermal injuries have a tremendous, long–lasting increase in transcutaneous heat loss by increased radiation, convection and evaporation (Cope et al., 1953; Wilmore et al., 1975; Caldwell, 1976, 1991; Arturson et al., 1978; Caldwell et al., 1981, 1992; Cone at al., 1988; Martin et al., 1992; Wallace et al., 1994).

[1] Secretion of stress hormones and metabolic mediators (i.e. corticosteroids, catecholamine, glucagon) cause changes in catabolism of proteins, ureagenesis, fat mobilization, glycogenolysis, gluconeogenesis, elevated glucose flow–weight loss.

[2] ↑100–300 % resting energy expenditure, ↑50 % heart rate, ↑blood pressure, ↑minute ventilation (Cope et al., 1953; Wilmore et al., 1973a, 1973b, 1974b, 1975; Danielsson et al., 1976; Arturson et al., 1978; Martin et al., 1992; Wallace et al., 1994).

Ramos et al. (2002) concluded that post–surgical cool sensation and discomfort were greater in hypothermic patients than in non–hypothermic patients. In addition to the effect of injury on metabolic rate, patients have been reported to experience an upward shift in comfort temperature compared with that of normal individuals. Although post−burn hypermetabolism was observed to be sensitive to ambient temperature, burn size appeared to be the primary determinant of the magnitude of the hypermetabolic response (Kelemen et al., 1996). These changes caused by the burn wound can be altered by environmental, nutritional, and pharmacologic means (Cope et al., 1953; Epstein et al., 1963; Zawacki et al., 1970; Wilmore et al., 1973a, 1974a, 1974b, 1975; Neely et al., 1974; Caldwell, 1976; Danielsson et al., 1976; Arturson et al., 1978; Caldwell et al., 1981; Herndon et al., 1987a, 1988; Fleming et al., 1992; Herndon and Parks, 1992; Caldwell et al., 1992; Carlson et al., 1992; Martin et al., 1992; Wallace et al., 1994; Birk and Cunningham, 1996; Kelemen et al., 1996; Gore et al., 2005). Occlusive dressing can significantly lower the radiation heat loss from burn wounds, but evaporative heat loss is not reduced significantly (Caldwell et al., 1981). To minimize hypermetabolic response of a burn patient an environmental warming is one of the most important actions (Neely et al., 1974; Wilmore et al., 1975; Herndon, 1996: 237).

Ambient temperatures and humidity should be maintained at 28–33 °C and 80 %, respectively, in order to decrease energy demands, minimize metabolic expenditure and lead to a decrease in the hypermetabolic response to thermal injury and evaporative water losses (Caldwell, 1976, 1991; Herndon et al., 1987a, 1988; Cone at al., 1988; Caldwell et al., 1992; Herndon and Parks, 1992; Wallace et al., 1994). Mortality, morbidity and hospitalization can be significantly decreased (Herndon et al., 1987a, 1988; Herndon and Parks, 1992).

Regarding specifics related to hospital environment, a burn patient with 80 % TBSA, 2 met, hypermetabolism and hypothermia was considered for the experimental part. The individual characteristics with references are collected in Table 50, Annex I.

7.1.3 Definition and selection of impact factors with matrix construction, results of step 1

After evaluation of physical, chemical and biological factors, all relevant parameters were selected and organized into matrix. In the field of physical factors the selected relevant parameters are:

> ***Physical impact factors***
>
> ***1. Elements/parameters:*** *building envelope (constructional complex, structure, surface, installations); active space (physical space, air quality, patient room volume)*
>
> ***2. Factors of user comfort:*** *heat exchange (convection, conduction, radiation, evaporation, respiration); air quantity and stratification.*

Table 16: *Example of constructed matrix for physical factors in HE*

HEAT TRANSMISSION/ AIR QUANTITY BUILDING ENVELOPE: CONSTRUCTIONAL COMPLEXES		USER: HEAT TRANSMISSION MODE					AIR QUANTITY	
		1	2	3	4	5	6	7
		Convection	Conduction	Radiation	Evaporation	Human respiration	Air quantity (ACH) and stratification quality-neutral	User
1	FLOOR: structure	1/1	2/1		4/1			7/1
2	FLOOR: surface	1/2	2/2	3/2	4/2			7/2
3	FLOOR: installations	1/3	2/3	3/3	4/3		6/3	7/3
4	WALL: structure	1/4	2/4		4/4			7/4
5	WALL: surface	1/5	2/5	3/5	4/5			7/5
6	WALL: installations	1/6	2/6	3/6	4/6		6/6	7/6
7	CEILING: structure	1/7	2/7		4/7			7/7
8	CEILING: surface	1/8	2/8	3/8	4/8			7/8
9	CEILING: installations	1/9	2/9	3/9	4/9		6/9	7/9
10	BED: structure	1/10	2/10		4/10			7/10
11	BED: surface	1/11	2/11	3/11	4/11			7/11
12	BED: installations	1/12	2/12	3/12	4/12		6/9	7/12
13	Physical space	1/13		3/13				7/13
14	User					5/11		

On the level of chemical factors the selected relevant parameters are:

> ***Chemical impact factors***
>
> ***1. Types of contaminants*** *(alcohols, ketones, acids, aldehydes, etc.)*
>
> ***2. Sources of contaminants:*** *contaminants from external sources (natural, anthropogenic sources); contaminants from internal sources (patients, personnel, visitors, building materials, furniture, disinfection substances and cleaning agents, anaesthetic gases, sterilization substances, etc.).*

Table 17: *Example of constructed matrix for chemical factors in HE (cutting from the original matrix of the size 23 rows and 6 columns)*

CHEMICAL FACTORS		1.0 External sources	2.0 Internal sources				
		1.1	2.1	2.2	2.3	2.4	2.5
Contaminant/ Source		Anthropogenic, natural sources	Patients, visitors, personnel	Building materials, furniture	Disinfection, cleaning agents	Anaesthetic gases	Sterilization substances
1	CO	1/1.1	1/2.1				
2	CO_2	2/1.1	2/2.1				
3	NOx	3/1.1					
4	SOx	4/1.1					
5	O_3	5/1.1					
6	PM	6/1.1	6/2.1				
7	VOC	7/1.1	7/2.1	7/2.2	7/2.3	7/2.4	
8	Water vapour	8/1.1	8/2.1				
9	PAH	9/1.1					
10	Soot	10/1.1					
11	Alcohols	11/1.1	11/2.1		11/2.3		
12	Pesticides	12/1.1					
13	CH	13/1.1	13/2.1	13/2.2			
14	Ketones	14/1.1	14/2.1				
15	Mercaptans, sulfids		15/2.1				
16	Acids		16/2.1		16/2.3		
17	NH_3		17/2.1				
18	Phosphates		18/2.1				
19	Ftalates			19/2.2			
20	Aldehydes			20/2.2	20/2.3		
21	Org. N Comp.		21/2.1				

On the level of biological factors the selected parameters are:

Biological impact factors

1. Types of contaminants *(bacteria, viruses, fungi, protozoa, etc.)*

2. Sources of contaminants*: contaminants from internal sources (patients, staff, visitors, ventilation system, surfaces, diagnostic instruments, water, liquids, soil, food, dust, etc.)*

3. Mode of transmission: *as part of the chain of infection, possible intervention (direct contact, indirect contact, vehicle–borne, airborne).*

Table 18: *Example of constructed matrix (cutting form the original matrix of the size 37 rows and 16 columns) for biological factors in HE*

BIOLOGICAL FACTORS IN HOSPITAL EVNIRONMENT			1.0 Internal sources												2.0 Mode of transmission			
			1.1	1.2	1.3	1.4	1.5	1.6	1.7	1.8	1.9	1.10	1.11	1.12	2.1	2.2	2.3	2.4
Contaminants/ Reservoir			Patients, staff, visitors	Ventilation system	Surfaces	Diagnostic instruments	Water	Liquids	Environment	Objects,	Devices	Soil	Food	Dust	Airborne	Vehicle	Contact-direct (hand, instrument, clothes)	Contact-indirect (medicine, food, water)
1.0 Bacteria	1.1	*Staphylococcus aureus, MRSA, VRE*	x						x	x			x			x	x	x
	1.2	*Staphylococcus epidermidis*	x														x	
	1.3	*Enterococcus spp.*	x															
	1.4	*Enterobacteriae*	x															
	1.5	*Mycobacterium tuberculosis*	x			x									x		x	
	1.6	*Mycoplasma pneumoniae*	x												x	x		x
	1.7	*Neisseria meningitidis*	x												x	x		x
	1.8	*Streptococcus pyogenes*	x												x	x		x
	1.9	*Streptococcus agulactiae*	x													x		x
	1.10	*Corinebacterie dipteriae*																
	1.11	*Pseudomonas spp.*				x	x	x	x		x						x	
	1.12	*Enterococcus viridans*																
	1.13	*Intestinal pathogens*											x		x	x		x
	1.14	*Listria monocitogenes*							x				x				x	
	1.15	*Legionella spp.*					x		x						x	x		x
	1.16	*Aeromonas spp.*				x	x	x	x								x	
	1.17	*Salmonella spp.*								x			x		x		x	

7.1.4 Design of the problem, results of step 2

The doctoral thesis was focused on searching for alternative solutions for a problem of high energy use in hospital buildings. With this respect, the matrix of physical parameters (Table 16) was further upgraded into Table 19, where the problem was detected and classified per partes. The high energy use for heating and cooling is strongly related with three different fields that are in constant interactions: building envelope (constructional complexes), heat transmission mode (user comfort), and H/C system (results from interactions of the other two fields). Alternative solutions were revealed from the interactions among parameters in columns and rows.

Table 19: *Matrix of physical parameters with detected and classified problem per partes (fields 1/4, 2/4, 3/4…): cutting from the matrix*

HEAT TRANSMISSION/ AIR QUANTITY		USER: HEAT TRANSMISSION MODE					AIR QUANTITY	
		1	2	3	4	5	6	7
BUILDING ENVELOPE: CONSTRUCTIONAL COMPLEXES		Convection	Conduction	Radiation	Evaporation	Human respiration	Air quantity (ACH) and stratification quality-neutral	User
1	FLOOR: structure	1/1	2/1		4/1			7/1
2	FLOOR: surface	1/2	2/2	3/2	4/2			7/2
3	FLOOR: installations	1/3	2/3	3/3	4/3		6/3	7/3
4	WALL: structure							
5	WALL: surface							
6	WALL: installations							
7	CEILING: structure							
8	CEILING: surface							
9	CEILING: installations							
10	BED: structure							
11	BED: surface							
12	BED: installations							
13	Physical space							
14	User							

HEAT TRANSMISSION/ AIR QUANTITY		**USER: HEAT TRANSMISSION MODE**				
		1	2	3	4	5
BUILDING ENVELOPE: CONSTRUCTIONAL COMPLEXES		Convection	Conduction	Radiation	Evaporation	Human respiration
3	FLOOR:installations	Conventional room heaters.	Conventional room heaters.	Conventional room heaters.	Conventional room heaters.	Conventional room heaters.
4	WALL: structure	Thermally well insulated building complex.	Thermally well insulated building complex.	Thermally well insulated building complex.	Thermally well insulated building complex.	Thermally well insulated building complex.
8	CEILING: surface	Application of LowEx system.	Application of LowEx system.	Application of LowEx system.	Application of LowEx system.	Application of LowEx system.
14	User	Impact on thermal comfort.	Impact on thermal comfort.	Impact on thermal comfort.	Impact on thermal comfort.	Impact on thermal comfort.

For example, fields from 1/4 to 5/4 present the interactions between structure of constructional complex and user's heat transmission mode. The high building energy use is closely related with not so well insulated test room that could affect thermal comfort of its users. An important aspect that has to be considered is also improvement of thermal insulation of constructional complexes in relation to the location characteristics.

Fields from 1/3 to 5/3 present interactions between constructional complex installations and user's heat transmission mode, while fields from 1/8 to 5/8 present interactions between surfaces of constructional complexes and user's heat transmission mode. H/C system could be included in the test room as a part of installation or as a surface application. Type of heating system considerably affects user's heat transmission mode and consequently leads to improvement or worsening of thermal comfort conditions. With this respect, the conventional system (i.e. installed oil–filled electric heaters) and LowEx system (i.e. heating–cooling ceiling panels) are in the sequel compared and analyzed.

Fields from 1/14 to 5/14 present interactions between user and heat transmission mode and are closely related with the type of H/C system and building envelope characteristics.

7.1.5 Synthesis of possible solutions, results of step 3

Towards holistic solution of the problem of high energy use the alternative solutions are: improvement of building envelope together with effective H/C system that enables optimal conditions for individual user with minimal possible energy use for heating and cooling.

The solution of building envelope based on the existing state is presented in Table 20 which is upgraded Table 19. The relevant parameters that are included in further analysis were selected from Table 20.

On the level of building envelope the selected relevant parameters are:

U–value for constructional complex without thermal insulation
U–value for constructional complex with thermal insulation
U–value for constructional complex with additional thermal insulation

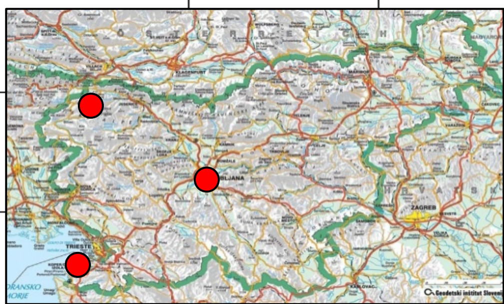
On the level of location the selected relevant parameters are:

Zone I, Mediterranean, $T_{ao,avg}$
Zone II, Continental, $T_{ao,avg}$
Zone III, Alps, $T_{ao,avg}$

After the selection of relevant parameters, a matrix was constructed. The columns include relevant parameters of location and the rows include building envelope's characteristics. The designed matrix is very convenient for the definition of priority actions and further analysis. The most important advantage of the developed matrix is its integrity. It covers all possible interactions among parameters that have been identified on the level of the first step of design (Table 20).

After the selection of relevant parameters that have to be further considered, the next step is the detection of all possible interactions, such as *U–value* of constructional complexes and building energy use, etc.. Detection of possible interaction among parameters was based on relevant studies and is further on elaborated in the analysis of exergy consumption patterns for space heating (section 7.2).

Table 20: *Matrix of interactions among characteristics of building envelope, site, and building energy use*

BUILDING ENERGY USE			Location		
			1	2	3
			Zone I, Mediterranean $T_{ao, avg}$	Zone II, Continental $T_{ao, avg}$	Zone III, Alps $T_{ao, avg}$
Building envelope/constructional complexes	1	U-value no insulation			
	2	U-value good insulation			
	3	U-value additional insulation			

LowEx systems were compared with conventional system from building and user point of view. Table 21 presents the part of the matrix of interactions between parts of human body exergy balance and building energy use. The construction of the matrix was based on two models. The first model presents human body exergy balance developed by Shukuya et al. (2010). The second model presents the model of building energy use that was developed within IEA ECBCS Annex 37 (Guidebook to IEA ECBCS Annex 37 …, 2006), IEA ECBCS Annex 49 (Guidebook to IEA

ECBCS Annex 49 …, 2011), ISO 13790. After precise analysis of models and their parameters, all relevant parameters for further analyses are selected.

On the level of human body exergy model the selected relevant parameters are:

Cool/warm radiant exergy rate in [W/m^2]
Cool/warm convective exergy rate in [W/m^2]
Breath air in (the rate of exergies of the inhaled humid air) [W/m^2]
Inner part in (metabolic thermal exergy rate) [W/m^2]
Exergy consumption rate [W/m^2]
Rate of stored exergy [W/m^2]
Exergy rate of exhalation and evaporation of sweat [W/m^2]
Cool/warm radiative exergy rate out [W/m^2]
Cool/warm convective exergy rate out [W/m^2]

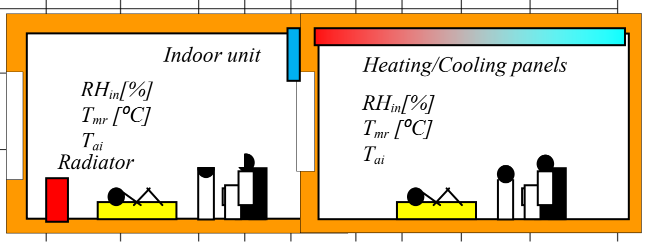
On the level of building energy use model the selected relevant parameters are:

T_{ai}, T_{mr}, RH_{in}

The columns include relevant parameters of human body exergy model and the rows include parameters of building energy use. The designed matrix is very convenient for developing LowEx systems (Table 21). After the selection of relevant parameters, the next step is the detection of interactions, such as human body exergy consumption rate and air temperature, etc.. Detection of possible interaction among parameters was based on relevant studies (Annex E) and is further on analyzed in the study of comparison between conventional and LowEx system.

Table 21: *Matrix of interactions between parts of human body exergy balance and building energy use*

HUMAN BODY EXERGY BALANCE			Individual characteristics									
			1	2	3	4	5	6	7	8	9	10
BUILDING ENERGY USE			PMV index	Cool (C), Warm (W) radiant exergy in [W/m^2]	Cool (C), Warm (W) convective exergy in [W/m^2]	Breath air in [W/m^2]	From inner part [W/m^2]	Exergy consumption rate [W/m^2]	Stored exergy [W/m^2]	Exhalation, sweat out [W/m^2]	Cool (C), Warm (W) radiation out [W/m^2]	Cool (C), Warm (W) convection out [W/m^2]
Building characteristics	1	T_{ai}										
	2	T_{mr}										
	3	RH_{in}										



The final goal is beside minimal possible energy use for heating and cooling, to create such LowEx system that enables optimal conditions for individual user, and attains the designed human body exergy consumption rate (Figure 34).

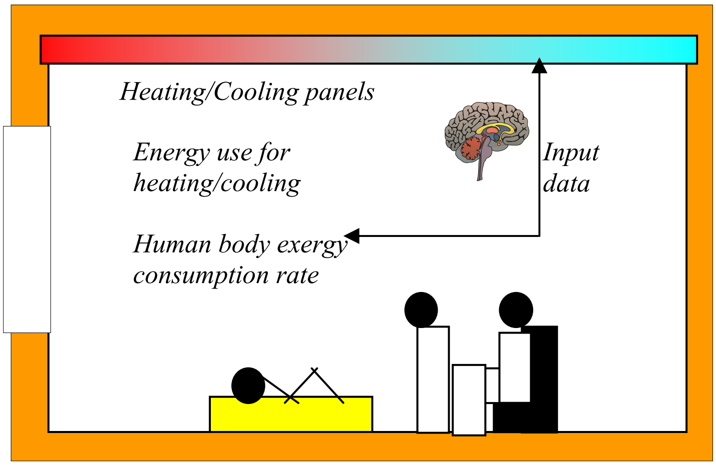


Fig 34: *Design of LowEx system*

Step 3 (the synthesis of possible solutions) revealed research fields for further analyses:

- comparison of energy supply for three cases of exterior walls at different locations and analysis of exergy consumption patterns for space heating;
- comparison of conventional and LowEx systems on the basis of exergy analysis of thermal comfort conditions for average and individual test subjects exposed to normal and extreme conditions, and measured energy use for heating and cooling.

7.2 Results of analysis of exergy consumption patterns for space heating

Our test room has no thermal insulation and its response depends significantly on the outdoor conditions. The design of building envelope has to be based on location characteristics. The most effective solution for our test room could be found with holistic consideration of all aspects of the whole heating system. In this respect the analysis of exergy consumption patterns for space heating was done. The results of the analysis were also published in our publications (Dovjak et al., 2010a, 2010b, 2011e, 2011g).

7.2.1 Methodology for the analysis of exergy consumption patterns for space heating

Steady–state exergy analyses were done for three cases of our model room located in three different Slovenian climate zones during winter conditions. In addition to exergy analyses, also energy analyses were performed. Results of both groups of analyses were compared and discussed. Average building heating energy demand was calculated as a result of steady–state energy analyses; while on the other hand, the exergy consumption rate was analyzed from the supply side to the demand side, i.e. from boiler to building envelope, as a result of steady–state exergy analyses. On the level of building walls, heat flow rate, exergy flow rate through building wall and radiant exergy flow rate from the interior surfaces of building envelopes were also compared.

The calculation of average building heating energy demand was carried out with a computer program called TOST (Jereb et al., 2008). Calculation procedure in TOST is based on building annual energy balance defined in Thermal performance of buildings – Calculation of energy use for space heating and cooling ISO 13790. TOST was developed and tested at UL FGG, KSKE. The reason for the development of such a program was EPBD (Directive 2002/91/EC), recast EPBD (Directive 2010/31/EU) and Slovenian regulations (Energy act, 1999, 2007, 2008, 2010; Rules on the methodology of construction …, 2009, Rules on efficient …, 2010) that require a certification procedure of energy efficiency for new and renovated buildings. The building energy certificate is also obligatory for presenting energy efficiency as part of building project documentation.

TOST enables the calculation of building annual heating energy use for steady–state conditions. Calculation considers climatic data, local characteristics, interior microclimatic demands and characteristics of all systems, subsystems and elements that have an influence on efficient use of building's energy. Building's energy use can be calculated per month or per heating season, for residential or public buildings. The program enables the consideration of unoccupied period of time, when the heating system is off. The main program outputs are calculated values of heat gains, heat losses and heating energy use. Results are compared with regulated demands by

Slovenian regulations and are presented in the form of Energy performance certificate for building (Construction act, 2002; Rules on efficient …, 2010; Jereb et al., 2008).

Annual energy use for heating was calculated for our test room of 163.4 m^3, having one exterior wall with a 15 m^2 glazed window. Others are interior walls. Using the TOST program, steady–state calculation for the whole period of heating season was made, neglecting the effect of solar heat gains and interior heat sources in order to be comparable to exergy calculation developed by Shukuya (1994). The annual energy use for heating of our test room [MJ/a] was converted in average heating energy demand. The average heating energy demand presents the whole amount of energy use divided by the period of time of the heating season (Dovjak et al., 2010a). It is expressed in watts [W] and will be further on compared with exergy consumption rate [W].

Steady–state exergy consumption in the whole chain of supply and demand for space heating is calculated in the following respective five sub–systems: 1) building envelope; 2) room air with fan; 3) heat exchanger; 4) water circulation with pump and 5) boiler. The same test room as for energy analysis is used for calculation. Input data are presented in Annex F. To achieve comparability of results between exergy and energy analyses, the solar heat gains and interior heat sources were eliminated also in the steady–state exergy calculation.

The exergy calculation of space heating from boiler to building envelope, calculation of heat flow rate [mW/m^2] and exergy flow rate [mW/m^2] through building wall and calculation of radiant exergy flow rate [mW/m^2] from the interior surfaces of building envelopes were carried out by following the detailed calculation procedure described by Shukuya (1994), Shukuya and Hammache (2002) and published in IEA ECBCS Annex 37 and IEA ECBCS Annex 49 (Guidebook to IEA ECBCS Annex 37 ..., 2006; Guidebook to IEA ECBCS Annex 49 …, 2011). Their brief explanations are given in Annex F, G and H.

Existing methodology for building exergy and energy calculations was applied for buildings in Slovenia situated in the central Europe, exactly at the cross–section of four big European geographic units: Alps, Pannonian basin, Dinaric mountain chain and the Mediterranean part. Slovenia has three main climate zones, so that exergy and energy analyses for these three zones were performed. Different response of our model test room was expected, when moved on different location. Moreover, the size of response depends considerably on thermal characteristics of room's envelope.

Based upon the real state of our test room and also upon the overview of the current status of Slovenian building regulation, several variants of exterior building walls were assumed. Case 1 corresponds to a wall having *U–value* according to our in situ test room, which represents a wall with no thermal insulation (U_{cu} = 1.29 W/(m^{2}K)). Case 2 corresponds to a wall having *U–value* according to current rules (U_{reg} = 0.28 W/(m^{2}K)). Case 3 represents a wall with additional thermal insulation (U_{add} = 0.15 W/(m^{2}K)). *U–value* for windows is 2.90 W/(m^{2}K) in Case 1 (double glazed widow, 16 mm air, without frame), 1.14 W/(m^{2}K) in Case 2 and Case 3 (double glazed Low–e, 16 mm Ar, without frame). For the building in climate zone I the selected location was Koper (Mediterranean part), climate zone II Ljubljana (Continental part), climate zone III Rateče (Alps). However, Case 1 located in the Continental part, Ljubljana, presents our real test room. The climatic data of these locations for steady–state calculation of average heating energy demand were taken from the Ministry of the Environment and Spatial Planning, Government of Republic of Slovenia (RS–MOP) (Climate data, 2009; Climate indicators …, 2009). The latest climatic data from a particular location that can be downloaded with on–line system from RS–MOP (Climate data, 2009; Climate indicators …, 2009) are 8.4 °C for Koper, 5.5 °C for Ljubljana and 4.9 °C for Rateče and they represent the average outdoor temperatures during heating period. Room air temperature was assumed to be constant at 20 °C in all calculations.

7.2.2 Comparison of energy supply

Figure 35 shows the average heating energy demand [W] for three climate zones in Slovenia and three different cases of test room (Case 1: U_{cu} = 1.29 W/(m^{2}K), U_{window} = 2.90 W/(m^{2}K), n = 1.0; Case 2: U_{reg} = 0.28 W/(m^{2}K), U_{window} = 1.14 W/(m^{2}K), n =

0.7; Case 3: U_{add} = 0.15 W/(m^{2}K), U_{window} = 1.14 W/(m^{2}K), n = 0.7). It presents the whole amount of energy use for heating season. Steady–state calculation for the whole period of a heating season neglects the effect of solar heat gains and interior heat sources due to comparability of results between exergy and energy analyses.

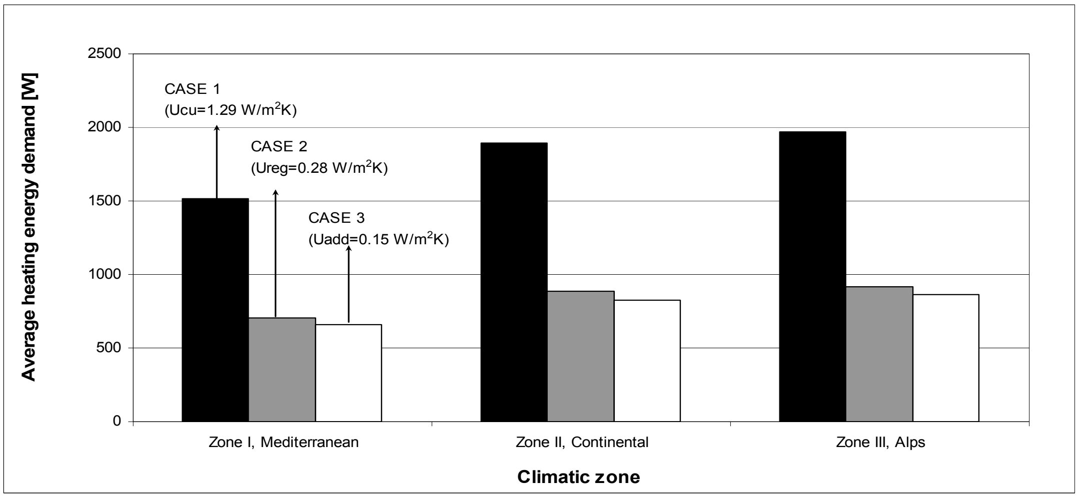


Fig 35: *Average heating energy demand [W], Case 1–3, Zone I–III*

The test room in Ljubljana (Case 1 in Zone II, Continental part) has no thermal insulation and due to that very high rate of transmission losses (1252 W) and average energy demand (1898 W). Additional thermal insulation decreases the rate of transmission losses by 70 % (875 W) and results in 56 % lower (1068 W) average heating energy demand (Case 1–Case 3, Zone II, Continental part).

Maximal average heating energy demand appears in Case 1 in Zone III, Alps (1968 W), mainly due to high rate of transmission losses through poorly insulated building envelope (1299 W). Minimal average heating energy demand appears in Case 3 in Zone I, Mediterranean (663 W), mainly due to low rate of transmission losses through well–insulated building envelope (302 W). Maximal decrease of average heating energy demand appears in the Alps, because of additional thermal insulation, improved windows and air tightness (Case 1–Case 3: 56 % or 1108 W decrease of average heating energy demand and 70 % or 907 W decrease of the rate of transmission losses).

It can be concluded that additional thermal insulation, improved windows and air tightness are important, especially in colder climates. We can find out that in the Mediterranean part the average heating energy demand is by 23 % lower than in the Alps in Case 1, Case 2 and Case 3. From this perspective, it is very important that the design of building envelope is based on the characteristics of respective locations and at the same time, the air tightness has to maintain good indoor air quality. However, from the results of calculations with included solar heat gains and interior heat sources it can be seen that in the most severe climate zone and with poor insulation the solar and interior heat gains cover more than 40 % of cumulated heat losses, while in the Mediterranean zone with improved insulation and with better window and air tightness this ratio exceeds 90 %. The utilized fragment of the gains has considerable effect on the final results. However, the aspect of the comparability of the results, the same approach for energy and analysis exergy analysis will be used and utilized fragment of the gains will be excluded from calculations.

7.2.3 Results of exergy consumption patterns

Table 22 summarizes the input data for the calculation in three Cases for exergy calculation. To prevent any mix–up of results in this phase, it is assumed that there is no solar radiation transmitted though glass window and absorbed by interior surfaces of building envelope and no heat generation from occupants, electric lighting, or electric appliances.

Table 22: *Case characteristics for exergy consumption analysis for 4 stages of space heating*

CASE, ***Room characteristics***	***C1*** ***(U_{cu})***	***C2*** ***(U_{reg})***	***C3*** ***(U_{add})***	***C2/1*** ***($U_{reg+eff}$)***	***C3/1*** ***($U_{add+eff}$)***
V [m^3]	*163.4*	*163.4*	*163.4*	*163.4*	*163.4*
A_{window} [m^2]	*15*	*15*	*15*	*15*	*15*
U_{window} [W/(m^{2}K)]	*2.90*	*1.14*	*1.14*	*1.14*	*1.14*
U_{wall} [W/(m^{2}K)]	*1.29*	*0.28*	*0.15*	*0.28*	*0.15*
n [h^{-1}]	*1*	*0.7*	*0.7*	*0.7*	*0.7*

to be continued...

...continuation

CASE, ***Room characteristics***	***C1*** ***(U_{cu})***	***C2*** ***(U_{reg})***	***C3*** ***(U_{add})***	***C2/1*** ***($U_{reg+eff}$)***	***C3/1*** ***($U_{add+eff}$)***
T_h outlet air temperature [K]	*303*	*303*	*303*	*303*	*303*
Inlet water temperature [K]	*343*	*343*	*343*	*343*	*343*
Outlet water temperature [K]	*333*	*333*	*333*	*333*	*333*
Electric power of the fan [W]	*30*	*16*	*16*	*16*	*16*
Electric power of the pump [W]	*23*	*12*	*12*	*12*	*12*
Ratio of chemical energy to the higher heating value of liquidified natural gas	*0.94*	*0.94*	*0.94*	*0.94*	*0.94*
Ratio of produced electricity energy to the higher heating value of liquidified natural gas	*0.35*	*0.35*	*0.35*	*0.35*	*0.35*
Thermal efficiency of boiler	*0.80*	*0.80*	*0.80*	*0.95*	*0.95*

In Case 1 (U_{cu}) the thermal insulation of the building envelope system is poor; i.e. classical double glazed widow and an exterior wall without insulation and a boiler with a moderate thermal efficiency. It presents in situ test room. In Case 2 (U_{reg}), Case 3 (U_{add}) the thermal insulation of the building envelope is improved by a combination of double–glazed window with a Low–E glass and argon gas fill and an exterior wall with improved insulation, while the boiler efficiency remains unchanged. In Case 2/1 ($U_{reg+eff}$), Case 3/1 ($U_{add+eff}$) additionally boiler's efficiency is improved from 0.80 to 0.95.

Figure 36 shows the comparison of cumulated exergy consumption rate [W] for four stages of space heating, for Case 1, Case 2 and Case 2/1 in the Continental part. However, Case 3 and Case 3/1 were also analyzed and discussed for all climates (Tables 23–25), but were not included into the figure, considering that there are no significant differences between exergy consumption profiles in Case 2 and Case 2/1.

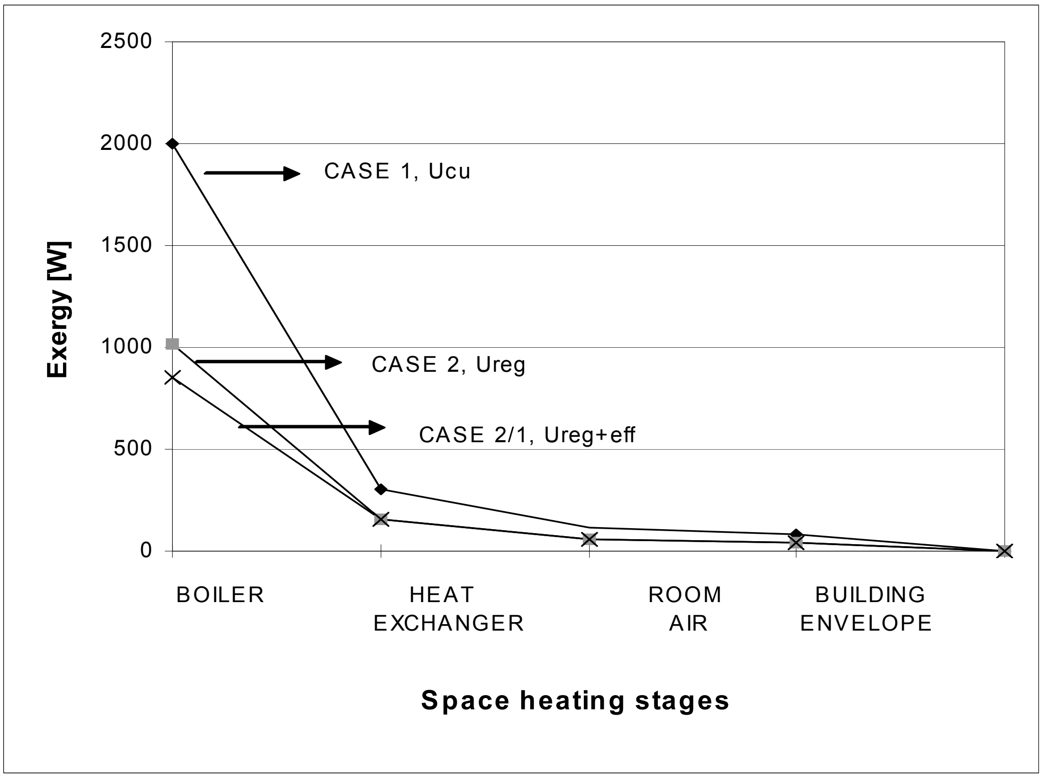


Fig 36: *Comparison of exergy consumption rate [W] for 4 stages of space heating, Continental part, Slovenia*

Table 23: *Comparison of exergy consumption rate [W] for 4 stages of space heating, Continental part, Slovenia*

Continental, Ljubljana	***BOILER*** *[W]*	***HEAT EXCHANGER*** *[W]*	***ROOM AIR*** *[W]*	***BUILDING ENVELOPE*** *[W]*
Case 1 U_{cu}	*2003.42*	*299.29*	*110.97*	*84.34*
Case 2 U_{reg}	*1015.73*	*151.74*	*56.26*	*42.76*
Case 3 U_{add}	*982.51*	*146.78*	*54.42*	*41.36*
Case 2/1 $U_{reg+eff}$	*855.35*	*151.74*	*56.26*	*42.76*
Case 3/2 $U_{add+eff}$	*827.37*	*146.78*	*54.42*	*41.36*

Table 24: *Comparison of exergy consumption rate [W] for 4 stages of space heating, Alps, Slovenia*

Alps, Rateče	***BOILER*** *[W]*	***HEAT EXCHANGER*** *[W]*	***ROOM AIR*** *[W]*	***BUILDING ENVELOPE*** *[W]*
Case 1 U_{cu}	*2086.32*	*314.83*	*119.14*	*91.46*
Case 2 U_{reg}	*1057.76*	*159.62*	*60.40*	*46.37*
Case 3 U_{add}	*1023.16*	*154.40*	*58.43*	*44.85*
Case 2/1 $U_{reg+eff}$	*890.75*	*159.62*	*60.40*	*46.37*
Case 3/2 $U_{add+eff}$	*861.61*	*154.40*	*58.43*	*44.85*

Table 25: *Comparison of exergy consumption rate [W] for 4 stages of space heating, Mediterranean part, Slovenia*

Mediterranean, Koper	***BOILER*** *[W]*	***HEAT EXCHANGER*** *[W]*	***ROOM AIR*** *[W]*	***BUILDING ENVELOPE*** *[W]*
Case 1 U_{cu}	*1602.74*	*227.73*	*75.50*	*53.98*
Case 2 U_{reg}	*812.58*	*115.46*	*38.28*	*27.37*
Case 3 U_{add}	*786.01*	*111.68*	*37.03*	*26.47*
Case 2/1 $U_{reg+eff}$	*684.28*	*115.46*	*38.28*	*27.37*
Case 3/2 $U_{add+eff}$	*661.90*	*111.68*	*37.03*	*26.47*

Exergy consumption is represented by the difference between exergy input and output. For example, in Case 1 (U_{cu}) the exergy rate of 2003 W is supplied to the boiler and the "warm" exergy rate of 299 W is produced and delivered to the heat exchanger by hot water circulation. The difference, i.e. 1704 W, is consumed inside the boiler due to combustion. The exergy input rate of 2003 W (Case 1, U_{cu}) also presents the total exergy consumption rate in the whole process of space heating. The value of the average heating energy demand calculated by TOST (1898 W) could be compared with the space heating exergy demand, q (1705 W) that causes all other portions of the exergy consumption rate in the whole process of space heating (Annex F). As can be seen, the results of exergy analysis are the same as those of energy analysis, if the comparison is made only for the energy supply and exergy supply. The small difference in the amount of supply between exergy and energy appears, because of calculation approaches; calculation of the average heating energy demand is based on ISO 13790, and the exergy calculation followed the procedure described by Shukuya (1994).

Exergy consumption within the boiler is the largest among the sub–systems and could not be avoided. This is the nature of the process. The basic principle of the boiler is to transfer energy stored in the fuel by chemical form to thermal energy to be contained by water in the boiler. In a typical fire–tube boiler, hot gas from the fire passes through one or more tubes running through a sealed container of water. The temperature of the burning gas is around 1200 °C, while on the other hand, the temperature of the water is 20–25 °C. The thermal energy for heating water passes from the gases through the sides of the tubes by thermal conduction. This large temperature difference inevitably results in a large amount of exergy consumption. Boiler efficiency in energy concept, i.e. the ratio of the thermal energy output to chemical energy input may be over 80 %, but the corresponding efficiency cannot be so high due to the large temperature difference. Case 2/1 ($U_{reg+eff}$) presents the situation after boiler improvement. As it can be seen, consumption of large quantities of exergy is unavoidable when extracting thermal exergy by a combustion process from the chemical exergy contained in liquidified natural gas (LNG). In Case 2 (U_{reg}) the line shows the result of the improvement of boiler efficiency from 0.8 to 0.95 in Case 2/1 ($U_{reg+eff}$). The decrease of exergy consumption rate is marginal (160 W) (Figure 36).

It may be considered that by increasing the outlet water temperature of the boiler, the exergy output from the boiler is larger and hence the boiler is more efficient. This, however, results in the consumption of more exergy within the water–to–air heat exchanger and also within the room air, where the required temperature is 20 °C. It is obvious that the biggest problem in the chain represents the exergy input into the boiler, which is extremely inefficient in case of non–renewable primary energy source. The answer to this problem is LowEx system. The design of a building itself or urban structure has nothing to do with the boiler stage. Real intervention must be treated in the next three stages with the main stress on room air and building envelope (Figure 37).

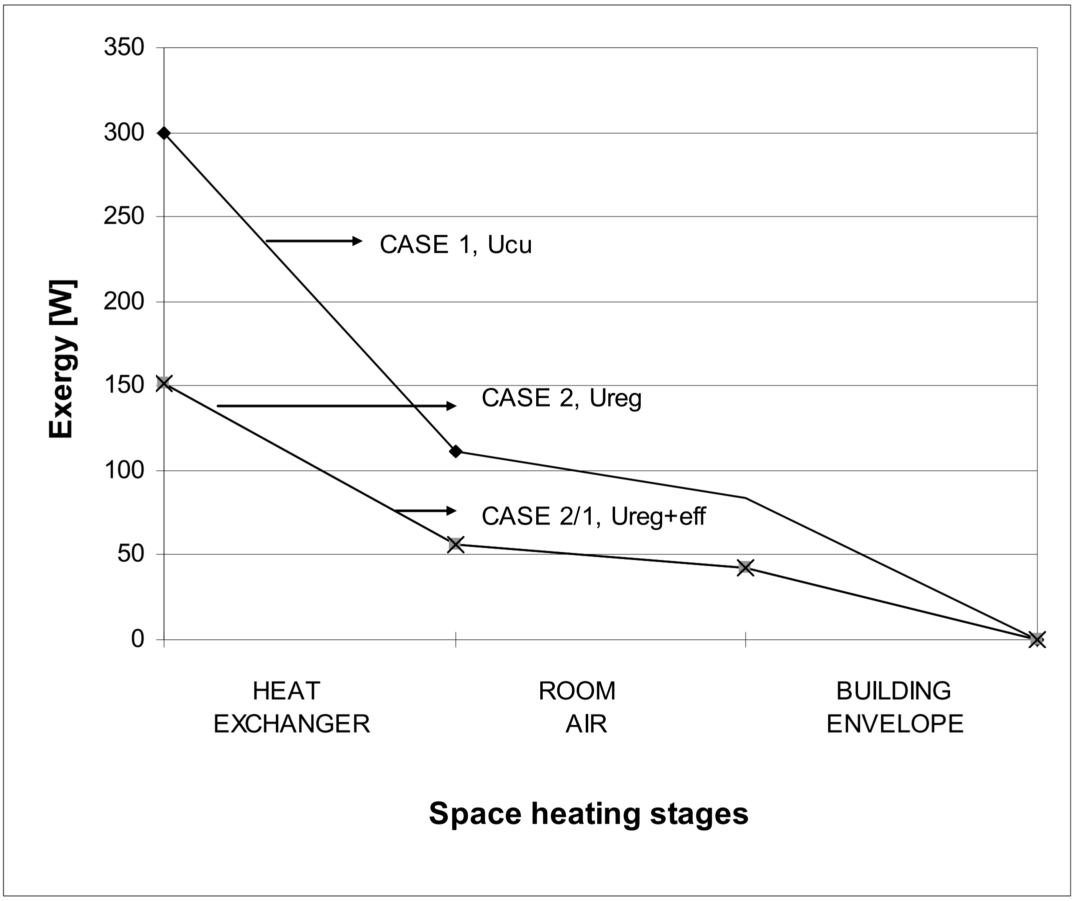


Fig 37: *Comparison of exergy consumption rate [W] for 3 stages of space heating (heat exchanger, room air and building envelope), Continental part, Slovenia*

The impact of building envelope improvement is shown in Figures 36–39. Case 1 (U_{cu}) presents the result of the situation before the intervention in the building envelope, and Case 2 (U_{reg}) and Case 3 (U_{add}) present the result after the intervention. The rate of input exergy in Case 1 (U_{cu}) is the highest among these cases (2003 W). Improvement of building envelope in Case 2 (U_{reg}) causes dramatic decrease of the rate of supplied exergy to the boiler (from 2003 to 1016 W of exergy consumption rate). The rate of output exergy in Case 1 (U_{cu}) is also the highest among these cases (299 W). Improvement of building envelope in Case 2 (U_{reg}) also causes decrease of "warm" exergy rate that is produced and delivered to the heat exchanger by hot water (from 299 W to 152 W) (Figure 36). Exergy consumption rate inside the boiler to combustion is decreased with improved thermal insulation (from 1704 W to 864 W).

How exergy consumption rate varies among subsystems in the Alpine area and the Mediterranean part for Case 1, U_{cu} is presented in Figures 38 and 39.

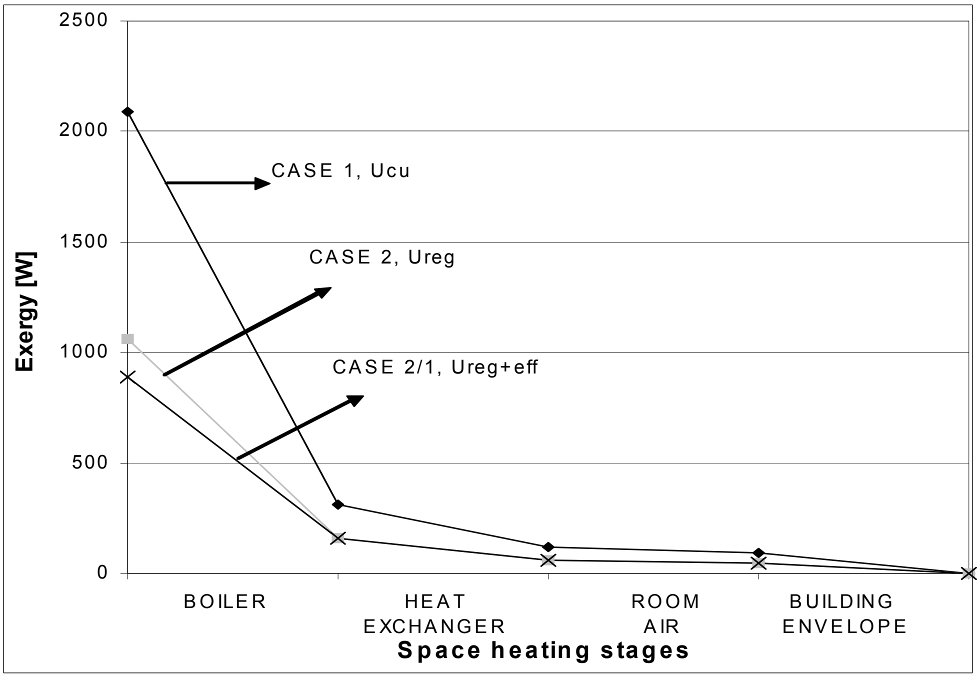


Fig 38: *Comparison of exergy consumption rate [W] for 4 stages of space heating, Alps, Slovenia*

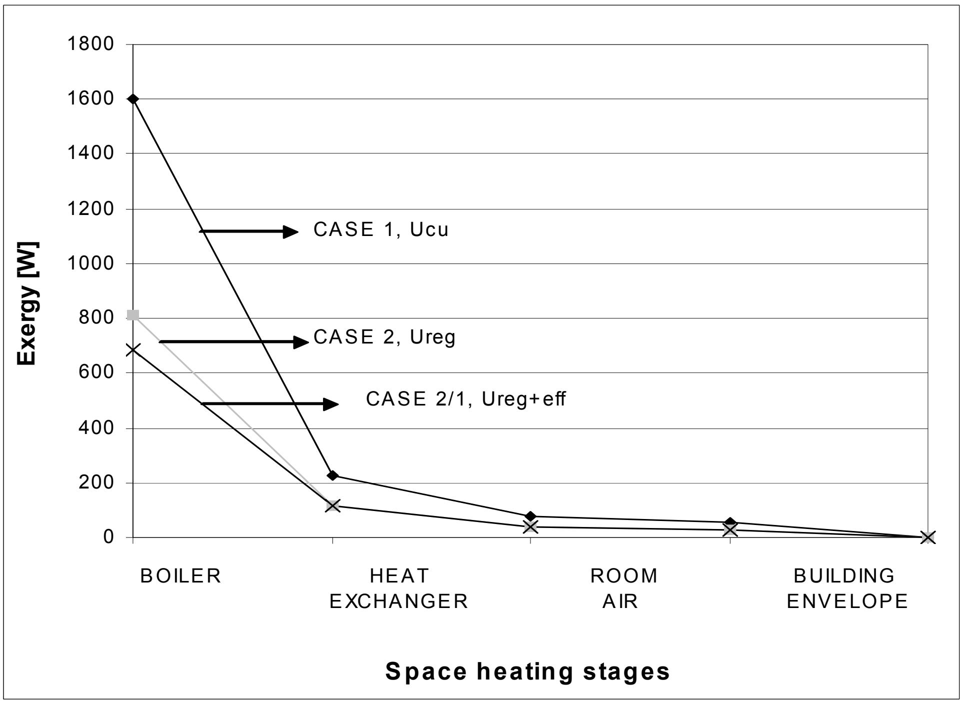


Fig 39: *Comparison of exergy consumption rate [W] for 4 stages of space heating, Mediterranean part, Slovenia*

In the Alpine area 2086 W of exergy rate are supplied to the boiler and in the Mediterranean part 1603 W. In the Alpine area 315 W of "warm" exergy rate are produced and delivered to the hot water heat exchanger and in the Mediterranean part 228 W. 1771 W of exergy rate are consumed inside the boiler due to combustion in the Alpine area and 1375 W in the Mediterranean part. These values are significantly different from those in the Continental part due to regarding different environmental temperatures.

It should be pointed out that extremely high boiler efficiency alone cannot make any significant contribution to the reduction of exergy consumption rate in the whole process of space heating. By focusing on Figure 36, it can be established that the heating exergy load, which is the rate of exergy output from the room air and the rate of exergy input to the building envelope, is 84 W in Case 1 (U_{cu}), 43 W in Case 2 (U_{reg}) and Case 2/1 ($U_{reg+eff}$), 41 W in Case 3 (U_{add}) and Case 3/1($U_{add+eff}$). It is only 6 to 7 % of the rate of chemical exergy input to the boiler, so that the introduction of a measure to reduce the heating exergy load may be considered as marginal. However, as can be seen from the difference in the whole exergy consumption profile between Case 1 (U_{cu}) and Case 2 (U_{reg}), it is more beneficial to reduce the heating exergy load by installing thermally well–insulated glazing and exterior walls (difference between Case 1–Case 2: 988 W reduction in the Continental part, 1029 W in the Alps, and 790 W in the Mediterranean) than to develop a boiler with an extremely high thermal efficiency (difference between Case 2–Case 2/1: 160 W reduction in the Continental part, 167 W in the Alps, and 128 W in the Mediterranean part) in order to decrease the total exergy consumption rate (Figures 36, 38, 39). The reduction in exergy consumption rate of the boiler sub–system due to the improvement in boiler efficiency turns out to be essentially meaningful together with the improvement of building–envelope's thermal insulation (difference between Case 1–Case 2/1: 1148 W reduction in the Continental part, 1196 W in the Alps, 919 W in the Mediterranean part) (Figures 36, 38, 39). The improvement of boiler efficiency becomes meaningful once the thermal insulation of the envelope is improved. Moreover, beside boiler efficiency also the problem of high energy value of primary energy source has to be emphasized.

If heat flow rate and exergy flow rates through building wall are compared for walls with different levels of thermal insulation, it can be established that there exist considerable differences (Case 1: 18.71 W/m^2 of heat flow rate and 0.925 W/m^2 of exergy flow rate; Case 2: 4.06 W/m^2 of heat flow rate and 0.201 W/m^2 of exergy flow rate). The difference is even more important, if besides improved thermal insulation also the location is taken into consideration (Case 1 in the Alps: 19.48 W/m^2 of heat flow rate and 1.003 W/m^2 of exergy flow rate; Case 2 in the Mediterranean: 3.3 W/m^2 of heat flow rate and 0.129 W/m^2 of exergy flow rate).

The impact of climate zones can be seen by comparing Figures 36, 38, 39. The highest total exergy consumption rate through all phases of space heating and also the average heating energy demand appears in the Alps (2086 W of the total exergy consumption rate or 1968 W of average heating energy demand in Case 1, 1058 W and 918 W in Case 2, 1023 W and 860 W in Case 3) and the lowest in the Mediterranean part (1603 W or 1516 W in Case 1, 813 W or 707 W in Case 2, 786 W or 663 W in Case 3).

The total exergy consumption rate through all phases of space heating and the average heating energy demand in the Continental part is 2003 W or 1898 W in Case 1, 1016 W and 885 W in Case 2, 983 W and 829 W in Case 3.

However, exergy analyses show us that the highest difference exists between exergy consumption rate before and after the improvement of thermal insulation in the Alps (1029 W exergy savings, 1050 W energy savings calculated with TOST). Not so well insulated building envelope and low boiler efficiency reflect in the highest exergy consumption rate, especially in colder climates (Case 1, U_{cu}: 2003 W in the Continental part, 2086 W in the Alps, 1603 W in the Mediterranean) (Figures 36, 38, 39).

7.2.4 Radiant exergy flow rate from interior surfaces

Beside heat flow and exergy flow rates through building envelope, another aspect from the viewpoint of comfort should also be considered. This is "warm" radiant exergy available from the wall surfaces.

The amount of exergy contained by a substance varies with its temperature and also with its environmental temperature. Air has a certain amount of exergy, both when the air temperature is higher than that of the environment and when the air temperature is lower than that of the environment (Shukuya and Hammache, 2002). The exergy contained by the air at a temperature higher than its environment is an ability of thermal energy contained by the air to disperse into the environment. It is called "warm" exergy and it usually happens during winter season (Shukuya, 1996, 2006a; Shukuya et al., 2006b). On the other hand, the exergy contained at a temperature lower than its environment is the ability of the air, in which there is lack of thermal energy compared to the environment, to let the heat flux into the environment. It is called "cool" exergy and it usually happens during summer season (Shukuya, 1996, 2006a; Shukuya et al., 2006b).

In winter season, when the room space is heated and the room air temperature is higher than that of the outdoor environment, the room air has "warm" exergy as a quantity of state. The role of heating systems is to supply and consume exergy for keeping "warm" exergy contained by room space at a certain desired level. The exergy supply is to transfer exergy either by flows of conduction, convection, or radiation to the indoor environment space. "Warm" radiant exergy flow rate depends significantly on the thickness of thermal insulation and temperature difference between indoor and outdoor environment. Thermally well–insulated walls have higher surface temperatures and thereby contribute more to comfort conditions inside the heated indoor environment space. Thermal radiative exchange between the human body and the surrounding surfaces has important influence on thermal comfort. It is very important to use both absorption and emission of warm radiant exergy by raising the interior surface temperature. The fact that the radiant exergy flow rate coming from the interior surfaces of the exterior building walls becomes larger is related to higher surface temperatures on thermally well insulated building walls. Higher surface temperatures on the building walls result in higher temperatures of the skin and clothing surfaces and cause higher warm radiant exergy flow rate that is absorbed and emitted by the whole skin and clothing surfaces (Shukuya, 2006a, 2008b). It is also very important to understand the role of thermal insulation in order to assure sufficient amount of warm radiant exergy in winter. For the analysis of radiant exergy flow rate, coming from the interior surfaces of the exterior building walls also three cases of walls and three different climate zones were compared. Calculation of thermal radiant exergy flow rate from a unit area of building wall surfaces is described in Annex H and is based on equations given by Shukuya (2006a).

Figures 40, 41 and 42 show a quantitative relationship between warm radiant exergy flow rates [mW/m^2] as a function of surface temperature under the condition of outdoor air temperature.

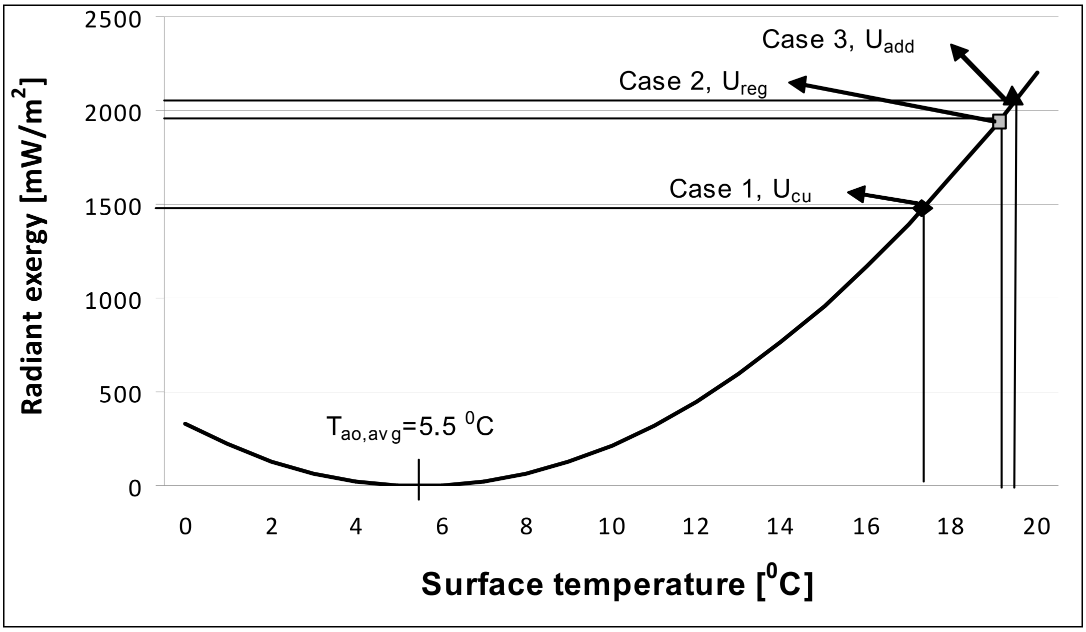


Fig 40: *Radiant exergy flow rate from the interior surfaces of building envelopes [mW/m^2] under winter conditions, 3 cases, Continental part, Ljubljana*

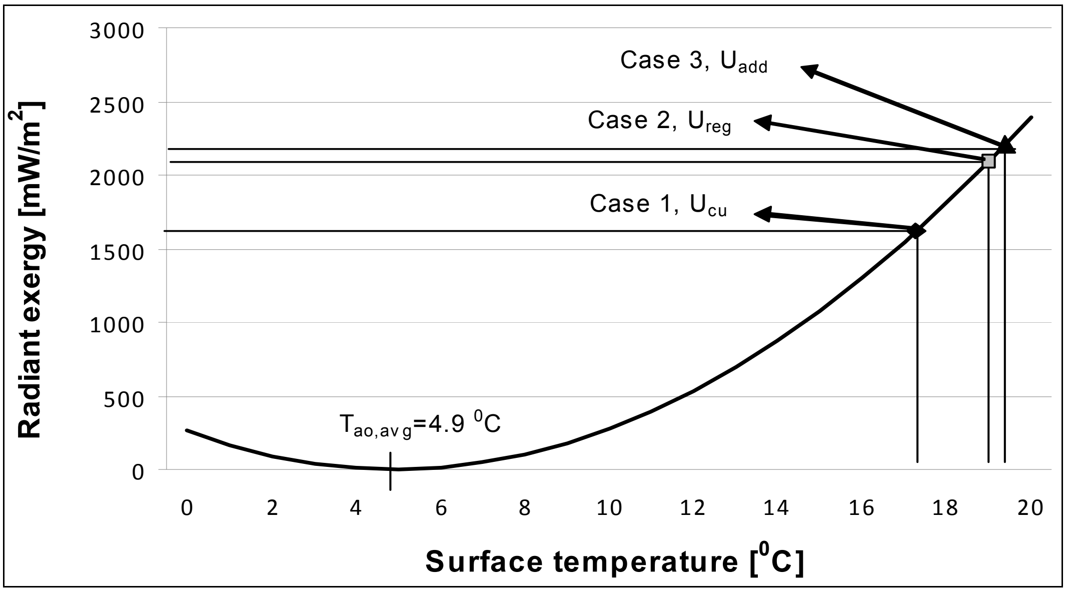


Fig 41: *Radiant exergy flow rate from the interior surfaces of building envelopes [mW/m^2] under winter conditions, 3 cases, Alps, Rateče*

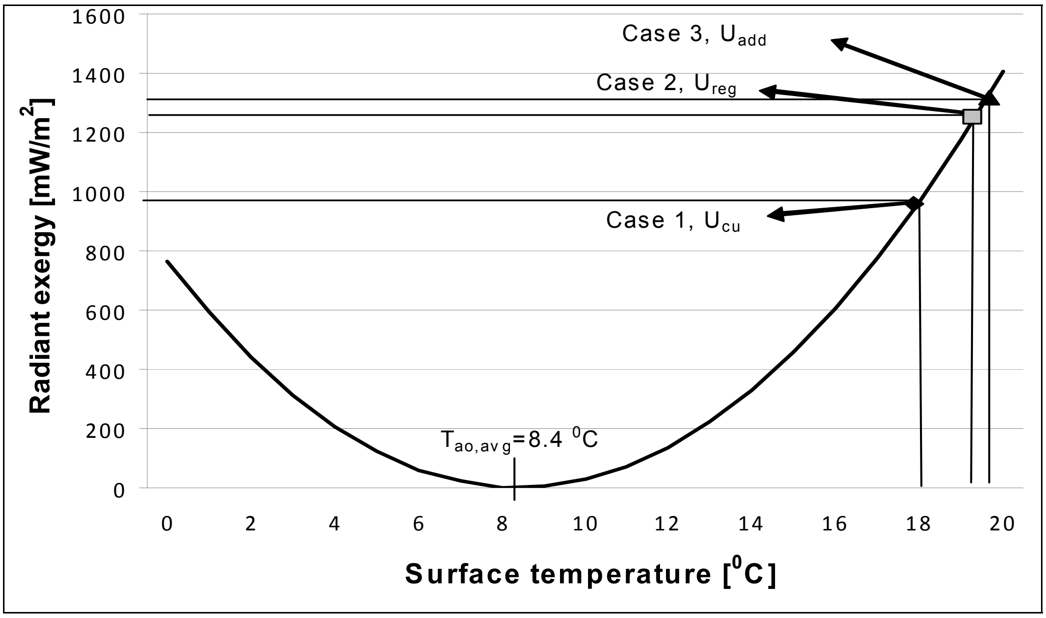


Fig 42: *Radiant exergy flow rate from the interior surfaces of building envelopes [mW/m^2] under winter Conditions, 3 cases, Mediterranean part, Koper*

In the Continental part (Ljubljana), the outdoor temperature as the environmental temperature for exergy calculation is assumed to be 5.5 °C. For the Alps (Rateče), the outdoor temperature is assumed to be 4.9 °C. For the Mediterranean part (Koper) the outdoor temperature is assumed to be 8.4 °C.

Figure 43 shows the warm radiant exergy flow rates [mW/m^2] for three specific cases of exterior walls. In any of the three climate zones, the amount of warm radiant exergy flow rates available from interior surfaces ranges from 500 to 2500 mW/m^2, as can be seen in Figures 40–43. The amount of warm radiant exergy flow rates available from interior surfaces of our test room in Ljubljana is low (1458 mW/m^2) due to poor insulation (U_{cu} = 1.29 W/(m^{2}K)).

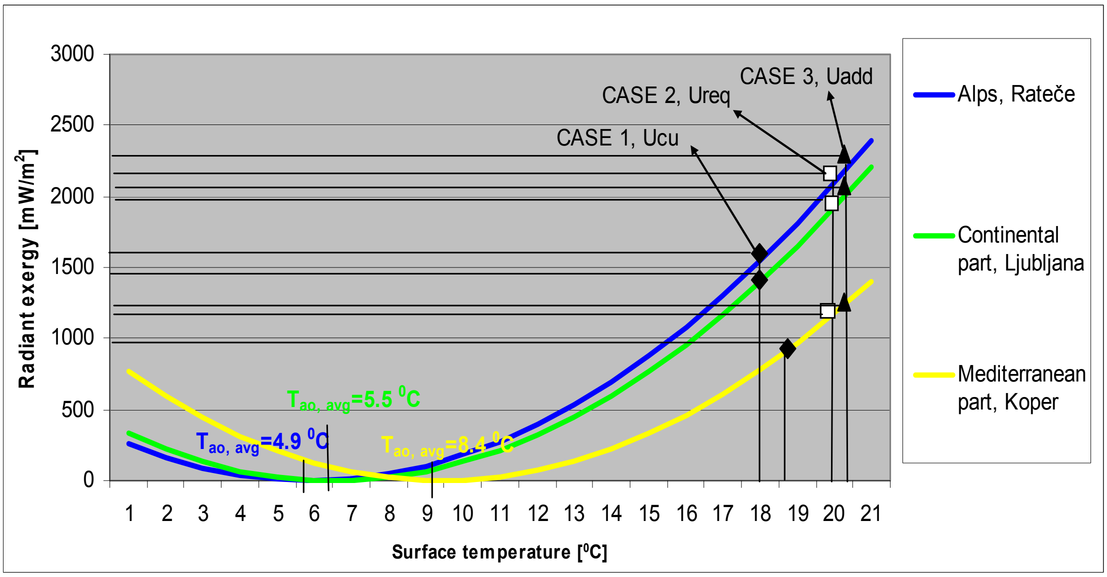


Fig 43: *Radiant exergy flow rates [mW/m^2] for three cases of exterior walls in three climate zones*

It can be established that better insulation makes possible a larger amount of warm radiant exergy flow rate, especially in the Alps (2189 mW/m^2 in Case 3) and also in the Continental part (2016 mW/m^2 in Case 3). The lowest value of warm radiative exergy flow rate is in Case 1 (U_{cu}) in the Mediterranean part (928 mW/m^2). Higher surface temperatures are brought by thermally well insulated wall in the manner of linearity. The surface temperatures in the Continental part are 17.6 °C in Case 1 (U_{cu}), 19.5 °C in Case 2 (U_{reg}) and 19.9 °C in Case 3 (U_{add}). The surface temperatures in the Alps are 17.5 °C in Case 1 (U_{cu}), 19.5 °C in Case 2 (U_{reg}) and 19.7 °C in Case 3 (U_{add}). The surface temperatures in the Mediterranean are 18.1 °C in Case 1 (U_{cu}), 19.6 °C in Case 2 (U_{reg}) and 19.8 °C in Case 3 (U_{add}).

If the three cases at one particular location are taken into consideration, the differences between surface temperatures or warm radiant exergy flow rates are quite significant (for example in the range of 0.4 °C–2.3 °C or 68–558 mW/m^2 among the cases in the Continental part). However, if changes in the surface temperatures together with warm radiant exergy flow rates for three cases at different locations are taken into the consideration, the differences also become considerable. The difference between Case 1 in the Alps and Case 3 in the Mediterranean is almost 2.3 °C for surface temperatures and 300 mW/m^2 for warm exergy flow rates.

However, warm radiant exergy flow rate coming from the interior surfaces doubles, since it is a function of the square of temperature difference, as shown in Annex H (Eq. (1)). Therefore, it should be once again confirmed that the first priority of improvement should come with the increase of the warm radiant exergy flow rate coming from interior surfaces by adding thermal insulation and consequently raising the surface temperatures. The walls with good thermal insulation have positive influence on warm exergy emission. One improved variant is a LowEx system with discrete thermal emitters, separated from the basic structure of the wall. It was implemented and tested in Slovenian Ethnographic Museum (Museums energy efficiency …, 1999; Krainer et al., 2007).

Results of preliminary analysis of exergy consumption patterns for space heating show that location with characteristics of building envelope has an important effect on average heating energy demand, exergy consumption rate of the whole heating system and thermal comfort conditions (surface temperature and warm/cool radiant exergy rate absorbed by the whole skin and clothing surfaces/discharged from the whole skin and clothing surfaces). It could be concluded that the first priority action that has to be carried out in future is an improvement of boiler efficiency and change of primary energy source on one side, and thermally well insulated building envelope on the other side.

7.3 Results of exergy analysis of thermal comfort conditions for average test subjects

The study considers the effect of variations between effective clothing insulation, T_{ai} and RH_{in} on human body exergy consumption rate and predicted mean votes (*PMV*) index of average test subjects. Subjective characteristics and experimental conditions were taken from the study by Toftum et al. (1998). The results were also published in our publication (Dovjak et al., 2011b).

7.3.1 Methodology for the analysis

7.3.1.1 Analyzed data

Toftum et al. (1998) studied the effect of different fabric materials (cotton, microfiber, nylon, Gore–Tex) on the skin and environmental temperature/effective clothing insulation together with perceived discomfort at a high level of skin humidity. The experiment was carried out in a climate chamber (4.8 m long, 3.6 m wide, 2.6 m high) in which T_{ai} and RH_{in} were controlled. In total, 40 sitting subjects were exposed to different combinations of experimental conditions (Table 26). Skin humidity was evaluated considering the combination of the vapour permeability of the clothing and thermal environment parameters. The relative skin humidity was in the range of 32–75 %, the skin wettedness within 0.09–0.48, and the moisture permeability from 0.12 to 0.40.

Table 26: *Subject characteristics and experimental conditions*

Subject characteristics						
Subjects	***gender***	***age***	***A_{du} (m^2)***			
20	*male*	*22.8±2.6*	*1.89±0.14*			
20	*female*	*22.8±2.6*	*1.89±0.14*			
Experimental conditions						
RH_{in}[1] [%]	***v_a [m/s]***	***T_{ai}[1] [°C]***	***T_{mr}[1] [°C]***	***Metabolic rate [met]***	***Effective clothing insulation [clo]***	***Mean skin humidity [%]***
50–80	*0.1*	*25.0–25.5*	*25.0–25.5*	*1*	*0.63–0.9*	*33.7*

[1] For all calculations T_{ai} and T_{mr} were assumed to be equal. T_{ao} and RH_{out} were equal to T_{ai} and RH_{in}.

Table 26 presents the subject characteristics and experimental conditions, and Table 27 characteristics of additional 5 treated cases. Human body exergy calculation was performed for average test subjects assuming that they are exposed to the same conditions inside the test space as in the experimental study conducted by Toftum et al. (1998)[1]. For all calculations, the reference environmental temperature (the outdoor environmental temperature, T_{ao}) and the outdoor relative humidity (RH_{out}) were assumed to be equal to T_{ai} and indoor relative humidity (RH_{in}). This assumption was made because exergy analyses are carried out for test subjects that are positioned inside an active space that presents a thermal environment insulated from outside conditions. Moreover, no data were available regarding the outdoor conditions in the analyzed data from Toftum et al. (1998). Furthermore, to assume the same conditions as in Toftum et al. (1998), T_{ai} was equal to T_{mr}. The assumptions $T_{ao} = T_{ai}$, $T_{surf} = T_{mr}$, $p_{vr} = p_{vo}$ are considered also in Eqs. (2)–(12) of Annex B.

Table 27: *Case characteristics*

Case analysis	***Clothing material***	***Moisture permeability [ ]***	***Air velocity [m/s]***	***Effective clothing insulation [clo]***	T_{ai}[1] ***[°C]***	T_{mr}[1] ***[°C]***	RH_{in}[1] ***[%]***
Case 1	*Cotton+ Gore−Tex*	*0.40*	*0.1*	*0.63*	*25.5*	*25.5*	*50*
Case 2	*Cotton+ Microfiber*	*0.32*	*0.1*	*0.90*	*25.0*	*25.0*	*80*
Case 3	*Cotton+ PU nylon*	*0.12*	*0.1*	*0.89*	*25.0*	*25.0*	*50*
Case 4	*Cotton + PU nylon*	*0.12*	*0.1*	*0.89*	*25.0*	*25.0*	*80*
Case 5	*Cotton + PU nylon*	*0.12*	*0.1*	*0.89*	*25.5*	*25.5*	*80*

[1] For all calculations T_{ai} and T_{mr} were assumed to be equal. T_{ao} and RH_{out} were equal to T_{ai} and RH_{in}.

7.3.2 Results of human body exergy consumption rate and PMV index for average test subjects

First, the results of human body exergy balance for given conditions in Case 1 (Cotton*Gore−Tex, 25.5 °C, 50 %, 0.63 clo) are presented.

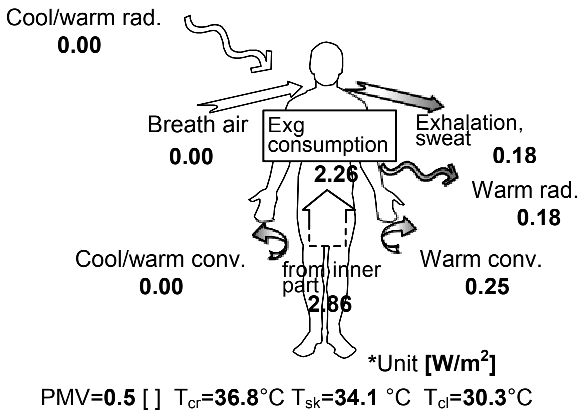


Fig 44: *Exergy balance of the human body for Case 1 (Cotton*Gore−Tex, 25.5 °C, 50 %, 0.63 clo). The exergy values are expressed as rates, W/m² (body surface)*

Figure 44 shows the example of the whole human body exergy balance in the following conditions: 25.5 °C, 50 %, 0.63 clo. The input exergy presents thermal radiative exergy exchange between the human body and the surrounding surfaces of active space, which influences on thermal comfort. Warm/cool radiant exergy rate absorbed by the whole skin and clothing surfaces is zero, because T_{ai} is equal to T_{mr}[1,2]. The sum of exergy rates contained by the inhaled humid air is also zero (breath air in Figure 44), because room T_{ai} and RH_{in} are equal to outside conditions T_{ao} and RH_{out} ($p_{vr} = p_{vo}$)[3]. Warm/cool convective exergy rate absorbed by the whole skin and clothing surfaces is also zero, because T_{ai} is equal to T_{ao}, and even temperature of clothing (hereinafter T_{cl}) is higher than T_{ai}[1]. The main input exergy (100 %) is presented by metabolic thermal exergy rate[4] (inner part in Figure 44). This means that the thermal exergy rate of 2.86 W/m² is generated by bio–chemical reactions inside the human body.

[1] T_{ao} was assumed to be equal to T_{ai}. In Eqs. (2)–(12), Annex B: $T_{ao} = T_{ai}$.

[2] In Eq. (6) Annex B: $T_{mr} = T_{surf.}$

[3] p_{vr} was assumed to be equal to p_{vo}. In Eqs. (2)–(12), Annex B: $p_{vr} = p_{vo}$.

[4] Metabolic thermal exergy rate (inner part) is the sum of warm exergy rate generated by metabolism, warm and wet exergy rates of liquid water generated in the core by metabolism, warm/cool and wet/dry exergy rates of the sum of liquid water generated in the shell by metabolism and dry air to let the liquid water disperse and the warm exergy rate stored in the core and the shell. In our case it appears as warm exergy rate of the sum of liquid water generated in the shell by metabolism because $T_{sk} > T_{ai}$ ($T_{ai} = T_{ao}$) and wet exergy rate of the sum of liquid water generated in the shell by metabolism because $p_{vs}(T_{ai}) > p_{vr}$.

As can be seen from Eqs. (2), (4), (5), (8) of Annex B with considered assumptions[1–3], it is mainly influenced by the metabolic rate, the ratio between T_{ai}[1] and body core temperature (hereinafter T_{cr}), the difference and the ratio between T_{cr} and T_{ai}[1], the ratio and the difference between skin temperature (hereinafter T_{sk}) and T_{ai}[1], the ratio between saturated water–vapour pressure at the room air temperature (hereinafter p_{vs} (T_{ai})[1] and water vapour pressure in the room space (hereinafter p_{vr})[2], as well as by the ratio between p_{vr} and water vapour pressure of the outdoor air (hereinafter p_{vo})[2]. It is important to keep the body structure functioning and to get rid of the generated entropy. Thus, 2.86 W/m^2 has to be released into ambient environmental space by radiation, convection, evaporation and conduction that present output exergies. The rate of warm exergy stored in the core and in the shell is 1 mW/m^2 and presents a part of metabolic thermal exergy rate, which is influenced by the ratio between T_{ai}[1] and T_{cr} and by the radio between T_{ai}[1] and T_{sk} (Eq. (8), Annex B). Because the moisture contained in the room air is not saturated, the water secreted from sweat glands evaporates into the ambient environmental space. The exergy rate of exhalation and evaporation of sweat[4] is 0.18 W/m^2 (6.2 % of output and consumed exergies). As can be seen from Eqs. (9) and (10) of Annex B, with the assumed conditions, they are influenced by the difference and the ratio between T_{cr} and T_{ai}[1], the ratio between $p_{vs}(T_{cr})$ and p_{vr}[1,2], the difference and the ratio between T_{cl} and T_{ai}[1], and by the ratio between p_{vr} and p_{vo}[1,2]. Warm radiant exergy rate discharged from the whole skin and clothing surfaces emerges because of higher T_{cl} than T_{ai}[1] and presents 0.18 W/m^2 (6.2 % of output and consumed exergies). Exergy rate of 0.25 W/m^2 (8.6 % of output and consumed exergies) is transferred by convection from the whole skin and clothing surfaces into surrounding air, mainly due to the difference between T_{cl} and T_{ai}[1,2]. The rate of exergy consumption that presents the difference between the rate of input exergy, the rate of stored exergy and the rate of output exergy is 2.26 W/m^2 (79 % of output and consumed exergies) for the conditions in Case 1.

[1] T_{ao} was assumed to be equal to the T_{ai}. In Eqs. (2)–(12), Annex B: $T_{ao} = T_{ai}$.

[2] p_{vr} was assumed to be equal to p_{vo}. In Eqs. (2)–(12), Annex B: $p_{vr} = p_{vo}$.

[3] In Eq. (6), Annex B: $T_{mr} = T_{surf}$.

[4] The exergy rate of exhalation and evaporation of sweat is the sum of warm and wet exergy rates of exhaled humid air, warm/cool exergy rate of the water vapour originating from the sweat and wet/dry exergy rate of the humid air containing the evaporated sweat. In our case there appears warm exergy rate of the water vapour originating from the sweat, because $T_{cl} > T_{ai}$ ($T_{ao} = T_{ai}$) and zero wet/dry exergy rate of the humid air containing the evaporated sweat because $p_{vr} = p_{vo}$.

If a test subject was exposed to the conditions from Case 2 (Cotton+Microfiber, 25 °C, 80 %, 0.9 clo), the rates of input and output exergies, stored exergy and exergy consumption would differ. Slightly lower T_{ai} and higher RH_{in} than at conditions from Case 1 cause lower input exergy rate by metabolic thermal exergy rate (due to smaller value of $p_{vs}(T_{ai})/p_{vr}$)[1,2]. Also in Case 2, the metabolic thermal exergy rate presents the only input exergy. Cool/warm radiant exergy rate absorbed by the whole skin and clothing surfaces is zero, because T_{mr} was equal to T_{ai}[1,3]. Cool/warm convective exergy rate absorbed by the whole of skin and clothing surfaces is zero, because T_{ai} was equal to T_{ao}. Higher humidity in Case 2 causes that the test subject cannot easily remove the resultant entropy by the evaporation and exhalation of sweat (due to smaller value of $p_{vs}(T_{cr})/p_{vr}$, smaller difference of $T_{cl}–T_{ai}$ and smaller value of T_{cl}/T_{ai}). However, lower input exergy consequently results in lower exergy consumption rate in Case 2 (2.06 W/m^2). As mentioned in the introduction, at thermally comfortable conditions equal to thermal neutrality (*PMV* index close to 0) lower exergy consumption rates appear, when only the effect of temperature is taken into consideration. It is interesting that conditions in Case 1 are more acceptable comfort environment (*PMV* index = 0.5) with higher exergy consumption rate (2.26 W/m^2) than in Case 2 where *PMV* index is 1.4 and exergy consumption rate is 2.06 W/m^2. By considering this, it is very important to make in–depth analysis of the whole chain of human body exergy balance and to include the effect of RH_{in} (80 % in Case 2; 50 % in Case 1).

Figure 45a presents exergy inputs and Figure 45b exergy outputs, exergy stored and exergy consumption for the conditions in Case 1. Exposure to different experimental conditions in Cases 2–5 affects individual parts of human body exergy balance. However, metabolic thermal exergy rate presents the only exergy input in all cases (100 %) and varies from 2.42 W/m^2 in Case 5 to 3.07 W/m^2 in Case 3 (Table 28).

[1] T_{ao} was assumed to be equal to the T_{ai}. In Eqs. (2)–(12), Annex B: $T_{ao} = T_{ai}$.

[2] p_{vr} was assumed to be equal to p_{vo}. In Eqs. (2)–(12), Annex B: $p_{vr} = p_{vo}$.

[3] In Eq. (6), Annex B: $T_{mr} = T_{surf}$.

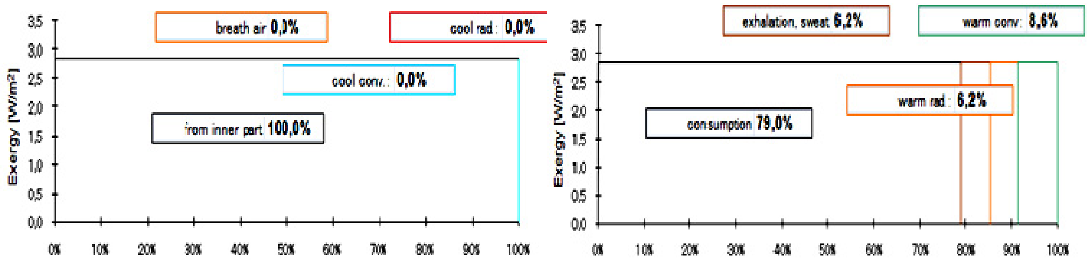


Fig 45: *a) Exergy inputs (left) and b) exergy outputs and exergy consumption (right) for the conditions in Case 1 (Cotton*Gore−Tex, 25.5 °C, 50 %, 0.63 clo). The exergy values are expressed as rates, W/m*^{2} *(body surface) and %*

Table 28 gives the calculated values of human body exergy consumption rate based on the experimental studied Cases 1–5. The exergy consumption rate presents the highest amount of output exergy rates in all cases (79.0–83.4 %). The lowest exergy consumption rate results in Case 5 (2.02 W/m^2) due to the lowest metabolic thermal exergy rate among cases (inner part in Table 28). The highest exergy consumption rate results in Case 3 (2.55 W/m^2), because of the highest metabolic thermal exergy rate (due to large difference of T_{cr}–T_{ai} and large value of T_{cr}/T_{ai}, large difference of T_{sk}–T_{ai} and large value of T_{sk}/T_{ai}). RH_{in} has important effect on exergy consumption rate and *PMV* index as well. For example, T_{ai} is in Case 3 and Case 4 the same (25.0 °C). Higher RH_{in} in Case 4 (80 %) causes lower exergy consumption rate (2.06 W/m^2, mainly due to lower metabolic thermal exergy rate) and less acceptable comfort environment (*PMV* = 1.4) than in Case 3 (at RH_{in} 50 %), where exergy consumption rate is 2.55 W/m^2 and *PMV* 0.9. In Cases 1, 2, 3 and 4 there result higher warm radiant exergy rate discharged from the whole skin and clothing surfaces (4.6–6.2 %, 0.14–0.18 W/m^2) and higher warm exergy rate transferred by convection from the whole skin and clothing surfaces (6.5–8.6 %, 0.20–0.25 W/m^2) because of larger difference of T_{cl}–T_{ai} and smaller value of T_{ai}/T_{cl} in comparison to Case 5 (warm radiative exergy rate, out: 5.4 %, 0.13 W/m^2 and warm convective exergy rate, out: 7.6 %, 0.18 W/m^2). The exergy rate of exhalation and evaporation of sweat is 0.18 W/m^2 (6.0–6.2 %) in Case 1 and 3 and 0.09 W/m^2 (3.6–3.8 %) in Cases 2, 4 and 5 due to almost the same T_{ai} and RH_{in} conditions (mainly due to almost equal ratio p_{vs} $(T_{cr})/p_{vr}$). The highest exergy rate of exhalation and evaporation of sweat results in Case 1 and 3 with 50 % RH_{in} (mainly due to the largest value of p_{vs} $(T_{cr})/p_{vr}$). Warm exergy rate transferred by convection from the whole skin and clothing surfaces is 0.20 in Cases 2, 3, 4 (6.5–8.1 %) and 0.25 W/m^2 (8.6 %) in Case

1. The lowest warm exergy rate transferred by convection results in Case 5 due to the smallest difference of $T_{cl}–T_{ai}$ and the largest value of T_{ai}/T_{cl}.

Table 28: *Calculated human body exergy balance for average test subjects. The exergy values are expressed as rates, W/m^2 (body surface)*

Case	***T_{ai}, T_{mr} [°C], RH_{in} [%]*** [1]	***PMV index***	***T_{cr}, T_{sk}, T_{cl} [°C]***	***Cool (C), Warm (W) radiant exergy in [W/m^2]***	***Cool (C), Warm (W) convective exergy in [W/m^2]***	***Breath air in [W/m^2]***	***From inner part [W/m^2]***	***Exergy consumption [W/m^2]***	***Stored exergy [W/m^2]***	***Exhalation, sweat out [W/m^2]***	***Cool (C), Warm (W) radiation out [W/m^2]***	***Cool (C), Warm (W) convection out [W/m^2]***
1	*25.5°C* *50 %*	*0.5*	*36.8* *34.1* *30.3*	*0*	*0*	*0*	*2.86*	*2.26*	*0.001*	*0.18*	*C=0* *W=* *0.18*	*C=0* *W=* *0.25*
2	*25.0°C* *80 %*	*1.4*	*36.8* *34.3* *29.3*	*0*	*0*	*0*	*2.50*	*2.06*	*0.001*	*0.09*	*C=0* *W=* *0.14*	*C=0* *W=* *0.20*
3	*25.0°C* *50 %*	*0.9*	*36.8* *34.2* *29.3*	*0*	*0*	*0*	*3.07*	*2.55*	*0.001*	*0.18*	*C=0* *W=* *0.14*	*C=0* *W=* *0.20*
4	*25.0°C* *80 %*	*1.4*	*36.8* *34.2* *29.3*	*0*	*0*	*0*	*2.50*	*2.06*	*0.001*	*0.09*	*C=0* *W=* *0.14*	*C=0* *W=* *0.20*
5	*25.5°C* *80 %*	*1.4*	*36.8* *34.2* *29.3*	*0*	*0*	*0*	*2.42*	*2.02*	*0.001*	*0.09*	*C=0* *W=* *0.13*	*C=0* *W=* *0.18*

Based on the above presented analysis it can be confirmed that RH_{in} has an important effect on the whole human body exergy balance and thermal comfort conditions. This is the starting–point for further analyses, i.e. individual test subjects.

[1] For all calculations T_{ai} and T_{mr} were almost equal. T_{ao} and RH_{out} were assumed to be equal to T_{ai} and RH_{in}.

7.4 Results of exergy analysis of thermal comfort conditions for individual test subject

Three test subjects (healthcare worker, burn patient, visitor) were exposed to normal and extreme conditions created in a room with LowEx and conventional system. Systems were compared from individual thermal comfort conditions and measured building energy use for heating and cooling. The analysis considers the effect of variations between T_{ai}, T_{mr} and RH_{in} on individual human body exergy consumption rate, *PMV* index and energy use for heating and cooling. The results were also published in our publication (Dovjak et al., 2010c, 2011a, 2011c, 2011d, 2011f).

7.4.1 Methodology for the analysis

7.4.1.1 Analyzed data

Test room (163.4 m^3) was equipped with LowEx system, where T_{ai} and T_{mr} differed in the range of 15–35 °C. The same room was equipped with electric heaters and split system with indoor unit used one at the time, where T_{ai} and T_{mr} varied in the range of 15–35 °C. Relative humidity differed in the range 30–96 %. Three test subjects (healthcare worker, burn patient, visitor) were selected for the individual thermal comfort calculations. Individual input data and experimental conditions were collected from the relevant literature (Cope et al., 1953; Caldwell et al., 1966; Fanger, 1970; Zawacki et al., 1970; Wilmore et al., 1973a, 1973b, 1974a, 1974b, 1975; Neely et al., 1974; Caldwell, 1976, 1991; Danielsson et al., 1976; ASHRAE Handbook & product …, 1977; Arturson et al., 1978; Herndon, 1981, 1996; Caldwell et al., 1981, 1992; Herndon et al., 1987a, 1987b, 1988; Cone et al., 1988; Simmers, 1988; Carlson at al., 1992; Fleming et al., 1992; Herndon and Parks, 1992; Martin et al., 1992; Shiozaki et al., 1993; Wallace et al., 1994; Kelemen et al., 1996; Wedler et al., 1999; LeDuc et al., 2002; Ramos et al., 2002; Herndon and Tompkins, 2004; Skoog et al., 2005; Atmaca and Yigit, 2006; ASHRAE Handbook HVAC applications …, 2007; Corralo et al., 2007; ISO 7730; ISO 8996; Sudoł–Szopińska and Tarnowski, 2007) and are presented in Table 29 and in more detail in Annex I.

Energy use for heating and cooling, T_{ao}, T_{ai}, T_{mr}, RH_{in} and RH_{out} were continuously monitored for both systems. Individual thermal comfort conditions were analyzed by calculated human body exergy consumption rates and predicted mean votes (*PMV*) index with spreadsheet software developed by Hideo Asada Rev 2010 (Iwamatsu and Asada, 2009). The calculation procedures followed the human body exergy model by Shukuya et al. (2010). Human body was treated as a thermodynamic system consisting of two systems: a core and a shell, based on exergy–entropy processes. For exergy calculations, the reference environmental temperature (the outdoor environmental temperature, T_{ao}) and the outdoor RH_{out} were assumed to be equal to the indoor T_{ai} and indoor RH_{in}. The assumptions $T_{ai} = T_{ao}$, $T_{mr} = T_{surf}$, $RH_{in} = RH_{out}$ were considered also in Eqs. (2)–(12) of Annex B.

Table 29: *Subject characteristics and experimental conditions*

Subject characteristics[1]					
Subjects	***Metabolic rate [met]***	***Effective clothing insulation [clo]***	T_{cr} **[°C]**	T_{sk} **[°C]**	T_{cl} **[°C]**
Healthcare worker	*1.1*	*0.6*	*36.8–37.2*	*29.5–36.8*	*23.3–36.2*
Burn patient	*2*	*0*	*35.5*	*37.0*	*25.3–37.1*
Visitor	*2*	*0.6*	*36.9–38.0*	*31.9–38.6*	*23.3–36.4*
Experimental conditions[2]					
RH_{in} **[%]**	v_a **[m/s]**	T_{ai} **[°C]**	T_{mr} **[°C]**		
30–96	*0.1*	*15.0–35.0*	*15.0–35.0*		

7.4.2 Results of individual human body exergy consumption rate and PMV index in normal room conditions

Table 30 presents the comparison between LowEx and conventional system in normal room conditions. Human body exergy consumption rates [W/m^2] and *PMV* index are calculated for healthcare worker, burn patient and visitor, exposed to different combinations of T_{ai} and T_{mr} that result in the same operative temperature

[1] In case of healthcare worker and visitor T_{sk} and T_{cr} varied due to experimental conditions, in case of burn patient values were constant due to hypothermic state and 80 % TBSA. T_{cl} varied due to experimental conditions among subjects.

[2] For all calculations, T_{ai} and T_{mr} were almost equal. T_{ao} and RH_{out} were assumed to be equal to T_{ai} and RH_{in}.

22.5 °C. RH_{in} is assumed to be 60 %. The data show that the human body exergy rates and *PMV* index vary among individuals for both systems, even if they are exposed to the same environmental conditions. Optimal conditions are created in the room with LowEx system, where higher surface than air temperatures (T_{mr} 27 ° C, T_{ai} 18 °C) are reflected in more acceptable comfort conditions for healthcare worker and visitor (*PMV* = –0.1, 0.1) than in the room with conventional system (*PMV* = –0.2, 0.3). In case of burn patient more acceptable comfort conditions are created in a room with LowEx system, where lower surface temperatures than air temperatures (T_{mr} 18 °C, T_{ai} 27 °C) are reflected in more acceptable comfort conditions (*PMV* = –0.8) than in room with conventional system (*PMV* = –1.0). However, comfortable conditions in the room with LowEx system do not always result in lower human body exergy consumption rates (healthcare worker: 2.23 W/m^2, 2.78 W/m^2; burn patient: 3.23 W/m^2, 3.52 W/m^2; visitor: 5.69 W/m^2, 5.56 W/m^2) than in a room with conventional system (healthcare worker 2.38 W/m^2; burn patient: 3.18 W/m^2; visitor: 5.50 W/m^2).

Table 30: *Comparison between LowEx system and conventional system*

Subject	***T_{ai} [1] [°C]***	***T_{mr} [°C]***	***PMV index***	***Exergy consumption rate [W/m^2]***
LowEx system				
Healthcare worker	*18.0*	*27.0*	*– 0.1*	*2.23*
Burn patient	*18.0*	*27.0*	*– 1.3*	*3.23*
Visitor	*18.0*	*27.0*	*0.1*	*5.69*
Healthcare worker	*27.0*	*18.0*	*– 0.2*	*2.78*
Burn patient	*27.0*	*18.0*	*– 0.8*	*3.52*
Visitor	*27.0*	*18.0*	*0.5*	*5.56*
Conventional system				
Healthcare worker	*22.5*	*22.5*	*– 0.2*	*2.38*
Burn patient	*22.5*	*22.5*	*– 1.0*	*3.18*
Visitor	*22.5*	*22.5*	*0.3*	*5.50*

The human body exergy consumption rates depend significantly on individual characteristics and will be considered in the sequel.

[1] For our calculations T_{ai} was assumed to be equal to $T_{ao.}$ RH_{in} was assumed to be 60 % and equal to $RH_{out.}$

The results presented in Table 30 can blur the differences between two systems and do not give detailed information about the influences on the separate parts of human body exergy balance between subjects. The whole human body exergy balance for three subjects exposed to normal conditions in a room with conventional and LowEx system were analyzed.

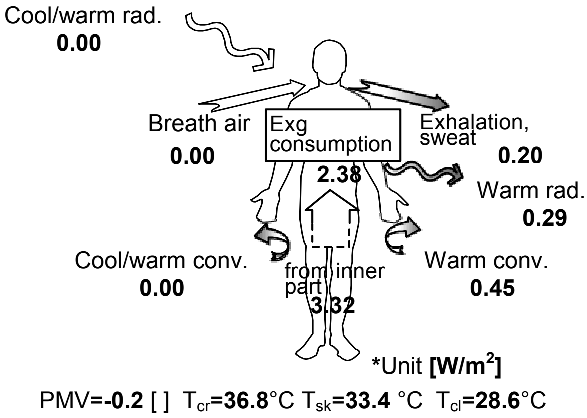


Fig 46: *Exergy balance of the human body for healthcare worker in a room with conventional system (22.5 °C T_{ai} and 22.5 °C T_{mr}, 60 % RH_{in}). The exergy values are expressed as rates, W/m^2 (body surface)*

Figure 46 shows the example of the whole human body exergy balance for healthcare worker in a room with conventional system (22.5 °C T_{ai}, 22.5 °C T_{mr}, 60 % RH_{in}). Input exergy presents thermal radiative exergy exchange between the human body and the surrounding surfaces of the active space and its influences on thermal comfort. Cool/warm radiant exergy rate absorbed by the whole skin and clothing surfaces is zero, because T_{ai} is equal to T_{mr}[1,2]. The sum of the exergy rates contained by the inhaled humid air (breath air in Figure 46) is also zero, because room T_{ai} and RH_{in} are equal to outside conditions T_{ao} and RH_{out} (p_{vr} = p_{vo})[3]. The warm/cool convective exergy rate absorbed by the whole skin and clothing surfaces is also zero, because T_{ai} is equal to T_{ao}. The main input exergy (100 %) is presented by metabolic thermal exergy rate (inner part in Figure 46). This means that the thermal exergy rate of 3.32 W/m^2 is generated by bio–chemical reactions inside the human body. The metabolic thermal exergy rate is influenced by the metabolic rate, the value of T_{ai}/T_{cr},

[1] T_{ao} were assumed to be equal to T_{ai}. In Eqs. (2)–(12), Annex B: $T_{ao} = T_{ai}$.

[2] In Eq. (6) Annex B: $T_{mr} = T_{surf}$.

[3] p_{vr} was assumed to be equal to p_{vo}. In Eqs. (2)–(12), Annex B: $p_{vr} = p_{vo}$.

the difference of $T_{cr}–T_{ai}$ and the value of T_{cr}/T_{ai}, the difference of $T_{sk}–T_{ai}$ and the value of T_{sk}/T_{ai}, the value of $p_{vs} (T_{ai})/p_{vr}$, and the value of p_{vr}/p_{vo} (Eqs. (2), (4), (5), (8) of Annex B)[1–3]. It is important to keep the body structure and function and to get rid of the generated entropy. Thus, 3.32 W/m^2 have to be released into ambient environmental space by radiation, convection, evaporation and conduction, which present output exergies. The rate of warm exergy stored in the core and in the shell is 40 μW/m^2, it presents part of metabolic thermal exergy rate and is influenced by the value of T_{ai}/T_{cr} and by the value of T_{ai}/T_{sk} (Eq. (8), Annex B)[1,2]. Because the moisture contained in the room air is not saturated, the water secreted from sweat glands evaporates into the ambient environmental space. The exergy rate of exhalation and evaporation of sweat is 0.20 W/m^2 (6.1 % of output and consumed exergies). They are influenced by the difference of $T_{cr}–T_{ai}$ and the value of T_{cr}/T_{ai}, the value of $p_{vs}(T_{cr})/p_{vr}$, the difference of $T_{cl}–T_{ai}$ and the value of T_{cl}/T_{ai}, and by the value of p_{vr}/p_{vo} (Eqs. (9), (10), Annex B)[1-3]. Warm radiant exergy rate discharged from the whole skin and clothing surfaces emerges because of higher T_{cl} than T_{ai} and presents 0.29 W/m^2 (8.6 % of output and consumed exergies). Exergy rate of 0.45 W/m^2 (13.5 % of output and consumed exergies) is transferred by convection from the whole skin and clothing surfaces into surrounding air, mainly due to the difference between T_{cl} and T_{ai}. The rate of exergy consumption that presents the difference between input, stored exergy and output exergies is 2.38 W/m^2 (71.7 % of output and consumed exergies) for healthcare worker in a room with conventional system (22.5 °C T_{ai} and 22.5 °C T_{mr} and 60 % RH_{in}).

The whole human body exergy balance for burn patient in a room with conventional system differs from that for the healthcare worker. Higher metabolic level (2 met) and lower T_{cr} for burn patient results in higher metabolic thermal exergy rate

[1] For our calculations T_{ai} was assumed to be equal to $T_{ao.}$ In Eqs. (2)–(12), Annex B: $T_{ao} = T_{ai}$. RH_{in} was assumed to be 60 % and equal to $RH_{out.}$

[2] In Eq. (6) Annex B: $T_{mr} = T_{surf}$.

[3] p_{vr} was assumed to be equal to p_{vo}. In Eqs. (2)–(12), Annex B: $p_{vr} = p_{vo}$.

(5.70 W/m^2) than in the case of healthcare worker (due to higher metabolic rate, larger difference of $T_{sk}–T_{ai}$ and larger value of T_{sk}/T_{ai}), higher exergy consumption rate (3.18 W/m^2) (due to larger rate of input exergy), zero rate of stored exergy (the exergy is released from the core and the shell), higher value of warm exergy rate transferred by convection from the whole skin and clothing surfaces into the surrounding air (1.59 W/m^2) (due to larger difference of $T_{cl}–T_{ai}$ and smaller value of T_{ai}/T_{cl}), higher value of warm radiant exergy rate discharged from the whole skin and clothing surfaces (0.57 W/m^2) (due to larger difference of $T_{cl}–T_{ai}$) and higher exergy rate of exhalation and evaporation of sweat (0.36 W/m^2) (due to larger difference of $T_{cl}– T_{ai}$ and larger value of T_{cl}/T_{ai}).

If visitor with doubled metabolic rate was exposed to normal room conditions in a room with conventional system that would result in doubled value of metabolic thermal exergy rate than for healthcare worker (6.86 W/m^2) (due to higher metabolic rate, larger difference of $T_{cr}–T_{ai}$, larger value of T_{cr}/T_{ai}, larger difference of $T_{sk}–T_{ai}$ and larger value of T_{sk}/T_{ai}) and doubled exergy consumption rate (5.50 W/m^2) (due to higher input exergies), higher rate of stored exergy (4 mW/m^2) (due to smaller value of T_{ai}/T_{cr} and smaller value of T_{ai}/T_{sk}), higher values of warm exergy rate transferred by convection from the whole skin and clothing surfaces (0.73 W/m^2) and higher exergy rate by exhalation and evaporation of sweat (0.37 W/m^2) (due to larger difference of $T_{cr}–T_{ai}$ and larger value of T_{cr}/T_{ai}) as well as lower warm radiant exergy rate discharged from the whole skin and clothing surfaces (0.26 W/m^2) (due to smaller difference of $T_{cl}–T_{ai}$).

If healthcare worker was exposed to the conditions (18.0 °C T_{ai}, 27.0 °C T_{mr}, 60 % RH_{in}) in the room with LowEx system that would result in the same operative temperature (22.5 °C) as in the room with conventional system, the whole human body exergy balance would differ. Larger difference of $T_{cr}–T_{ai}$ and larger value of T_{cr}/T_{ai}, larger difference of $T_{sk}–T_{ai}$ and larger value of T_{sk}/T_{ai} result in higher metabolic thermal exergy rate 4.28 W/m^2 (87.6 %) than in conventional system (3.32 W/m^2). Higher T_{mr} than T_{ai} causes 0.61 W/m^2 (12.4 %) of warm radiant exergy rate absorbed by the whole skin and clothing surfaces[1]. The sum of exergy rates of inhaled humid air (breath air) is zero because $T_{ai} = T_{ao}$ and $p_{vr} = p_{vo}$. Cool/warm exergy rate transferred by convection absorbed by the whole skin and clothing surfaces is also zero, because $T_{ao} = T_{ai}$. Metabolic thermal exergy rate and warm radiant exergy rate absorbed by the whole skin and clothing surfaces present the only input exergies. Output exergies are warm exergy rate transferred by convection[2] from the whole skin and clothing surfaces into the surrounding air, warm radiant exergy rate discharged[3] from the whole skin and clothing surfaces and the exergy rate of exhalation and evaporation of sweat. Higher T_{mr} than T_{ai} results in higher warm radiant exergy rate discharged from the whole skin and clothing surfaces (0.91 W/m^2, 18.6 %) (due to larger difference of T_{cl}–T_{ai}), higher warm exergy rate transferred by convection (1.45 W/m^2, 29.6 %) (due to larger difference of T_{cl} and T_{ai} and smaller value of T_{ai}/T_{cl}), slightly higher exergy rate of exhalation and evaporation of sweat (0.30 W/m^2, 6.0 %) (due to larger difference of T_{cr}–T_{ai} and larger value of T_{cr}/T_{ai}, larger difference of T_{cl}–T_{ai} and larger value of T_{cl}/T_{ai}) and lower exergy consumption rate (2.23 W/m^2, 45.7 %) (due to much higher output exergies and stored exergy) than in conventional system.

If burn patient was exposed to conditions in a room with LowEx system (18.0 °C T_{ai}, 27.0 °C T_{mr}, 60 % RH_{in}), that would result in 7.52 W/m^2 of metabolic thermal exergy rate (due to larger difference of T_{cr}–T_{ai} and larger value of T_{cr}/T_{ai}, larger difference of T_{sk}–T_{ai} and larger value of T_{sk}/T_{ai}) than in conventional system. Higher T_{mr} than T_{ai} causes 0.61 W/m^2 of warm radiant exergy rate absorbed[1] by the whole skin and clothing surfaces. T_{ai} equal to T_{ao} causes zero cool/warm convective exergy rate absorbed by the whole skin and clothing surfaces. Exergy consumption rate is 3.23 W/m^2 (it is higher due to higher input exergies), the rate of stored exergy is zero (exergy is released from the core and the shell). The output exergies are warm exergy rate transferred by convection[2] from the whole skin and clothing surfaces (3.24 W/m^2) (it is higher due to larger difference of T_{cl}–T_{ai} and smaller value of T_{ai}/T_{cl}), warm radiant exergy rate discharged[3] from the whole skin and clothing surfaces

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[1] In our case warm radiant exergy rate absorbed by the whole skin and clothing surfaces appears, because $T_{mr} > T_{ai.}$

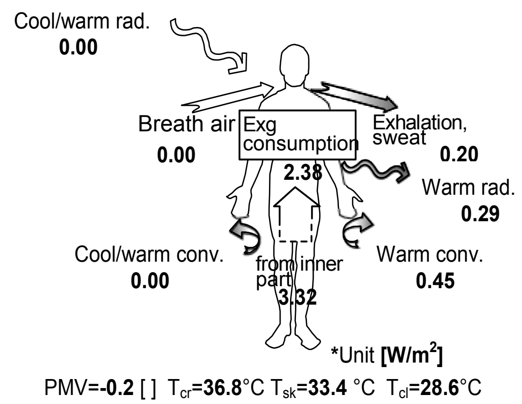
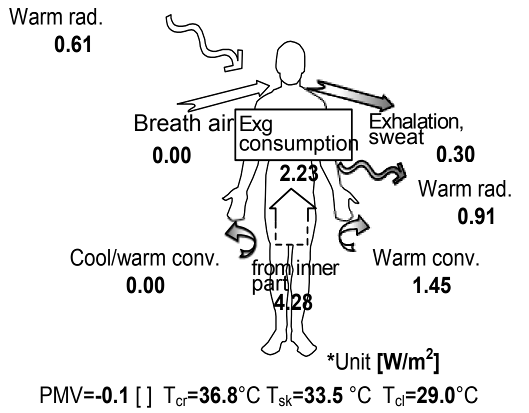
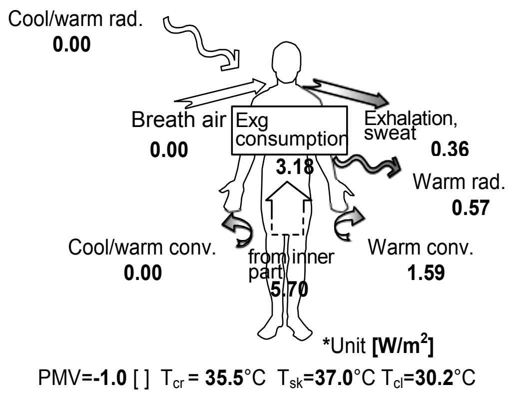
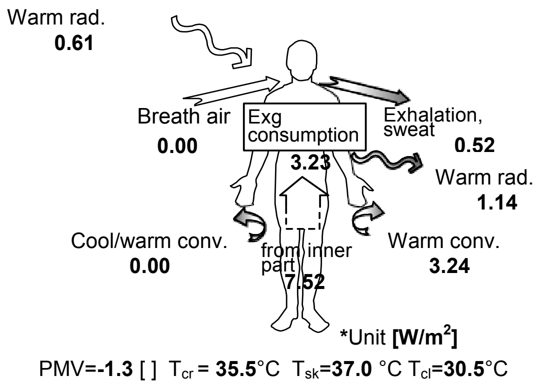
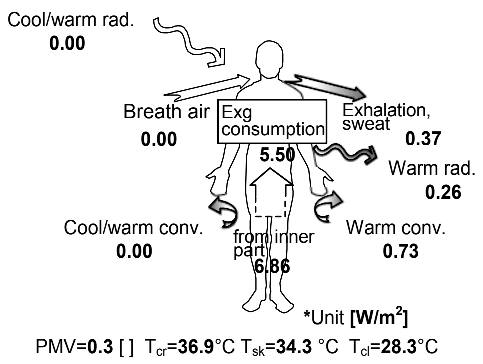
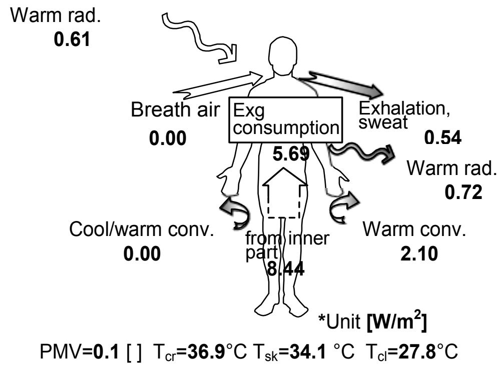
[2] In our case warm exergy rate transferred by convection from the whole skin and clothing surfaces appears, because $T_{cl} > T_{ai}$.

[3] In our case warm radiant exergy rate discharged from the whole skin and clothing surfaces appears, because $T_{cl} > T_{ai.}$

(1.14 W/m^2) (it is higher due to larger difference of T_{cl}–T_{ai}) and the exergy rate of exhalation and evaporation of sweat (0.52 W/m^2) (they are higher due to larger difference of T_{cr}–T_{ai} and larger value of T_{cr}/T_{ai}, larger difference of T_{cl}–T_{ai} and larger value of T_{cl}/T_{ai}). For visitor the same conclusions could be made as for burn patient: larger difference of T_{cr}–T_{ai} and larger value of T_{cr}/T_{ai}, larger difference of T_{sk}–T_{ai} and larger value of T_{sk}/T_{ai} result in higher metabolic thermal exergy rate (8.44 W/m^2) than in conventional system (6.86 W/m^2). Higher T_{mr} than T_{ai} causes 0.61 W/m^2 of warm radiant exergy rate absorbed[1] by the whole skin and clothing surfaces and higher output exergies than in conventional system (due to $T_{mr} > T_{ai}$). Warm exergy rate transferred by convection from the whole skin and clothing surfaces into the surrounding air is 2.10 W/m^2 (higher than in conventional system due to larger difference of T_{cl}–T_{ai} and smaller value of T_{ai}/T_{cl}), the exergy rate of exhalation and evaporation of sweat is 0.54 W/m^2 (higher due to larger difference of T_{cr}–T_{ai} and larger value of T_{cr}/T_{ai}, larger difference of T_{cl}–T_{ai} and larger value of T_{cl}/T_{ai}) and warm radiant exergy rate discharged from the whole skin and clothing surfaces is 0.72 W/m^2 in the room with LowEx system (higher due to larger difference of T_{cl}–T_{ai}). The difference among input exergies, stored exergy and output exergies is 5.69 W/m^2 and presents exergy consumption rate of a visitor in a room with LowEx system (it is the highest among subjects because of the highest input exergies).

[1] In our case warm radiant exergy rate absorbed by the whole skin and clothing surfaces appears, because $T_{mr} > T_{ai}$.

Table 31: *Individual human body exergy balance and systems comparison. The exergy values are expressed as rates, W/m² (body surface)*

<table>
<tr><td>Conventional system
(22.5 °C T_{ai} and 22.5 °C T_{mr}, 60 % RH_{in}) [1]</td><td>LowEx system
(18.0 °C T_{ai} and 27.0 °C T_{mr} , 60 % RH_{in}) [1]</td></tr>
<tr><td colspan="2">Healthcare worker</td></tr>
<tr><td>
</td><td>
</td></tr>
<tr><td colspan="2">Burn patient</td></tr>
<tr><td>
</td><td>
</td></tr>
<tr><td>Visitor</td><td></td></tr>
<tr><td>
</td><td>
</td></tr>
</table>

[1] For our calculations T_{ai} was assumed to be equal to T_{ao}, RH_{in} was equal to RH_{out}.

7.4.3 Results of individual human body exergy consumption rate and PMV index in extreme conditions

Thermal comfort conditions do not always result in lower human body exergy consumption rates. RH_{in} has an important effect with temperature conditions, as it was shown in the study presented in section 7.4.2. A series of in–depth analyses of the whole human body exergy balance shows how individual human body releases the heat and also the effects of different levels of RH_{in}. By exposing subjects to extreme environmental conditions (i.e. hot/humid, cold/humid, hot/dry, cold/dry) created in a room with conventional and LowEx system, the influence of different levels of relative humidity (30–96 %), T_{ai} and T_{mr} (15–35 °C) on human body exergy consumption rate [W/m^2] and on *PMV* index was analyzed. For the calculation of human body exergy consumption rate and *PMV* index in a room with conventional system the conditions $T_{ai} = T_{mr}$ with RH_{in} 30–96 % were assumed and for LowEx system the conditions $T_{ai} \neq T_{mr}$ with 80 % RH_{in} were assumed (recommendation, Herndon (1996: 492)). At conditions of 30 % of RH_{in}, skin wettedness was 0.09 and at conditions of 96 % of RH_{in}, skin wettedness was 0.48. The assumption $T_{ai} = T_{ao}$, $T_{mr} = T_{surf}$, $p_{vr} = p_{vo}$ was considered also in Eqs. (2)–(12) of Annex B.

7.4.3.1 Conventional system

Figure 47 presents human body exergy consumption rates [W/m^2] as a function of T_{ai} (= T_{mr}) and RH_{in} for healthcare worker in a test active space with conventional system (T_{ai}=T_{mr}, RH_{in} 30–96 %). If we change T_{ai} in the range 15–35 °C and RH_{in} 30–96 %, the exergy consumption rate for healthcare worker varies from 0.32 W/m^2 to 4.38 W/m^2. At hot and dry environmental conditions (35 °C T_{ai} and RH_{in} 30 %) the human body exergy consumption rate is the largest (4.38 W/m^2), mainly due to higher input exergy by metabolic thermal exergy rate[1], lower output exergies by warm radiant exergy rate discharged from skin and clothing surfaces[2] and warm exergy rate transferred by convection from the whole skin and clothing surface into

[1] Higher metabolic thermal exergy rate appears due to larger value of $p_{vs}(T_{ai})/p_{vr}$.

[2] Warm radiant exergy rate discharged from the whole skin and clothing surfaces is lower due to smaller difference of T_{cl}–T_{ai}.

the surrounding air[1] and lower rate of stored exergy[2] than at hot and humid environmental conditions (35 °C and 96 %), where human body exergy consumption rate is the minimal possible (0.32 W/m²).

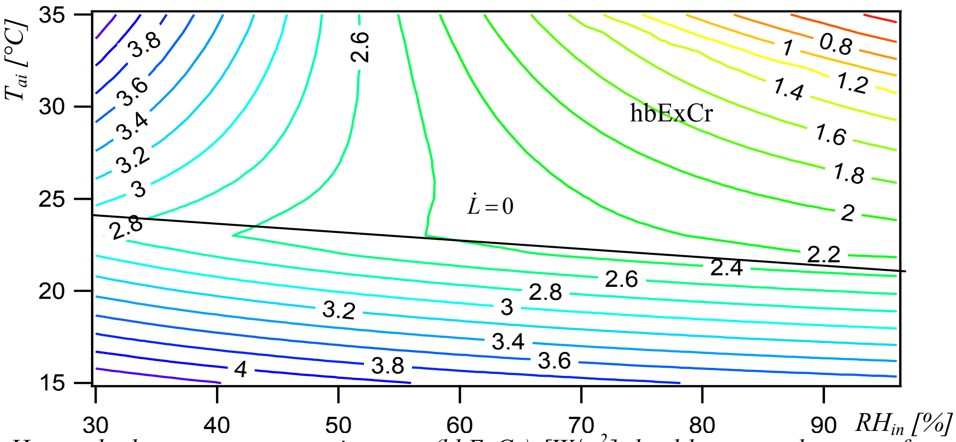


Fig 47: *Human body exergy consumption rate (hbExCr) [W/m²], healthcare worker, as a function of T_{ai} (= T_{mr}) [°C] and RH_{in} [%], conventional system. The black line presents thermal load on the body ($\dot{L}$ = 0, PMV = 0)*

The rate of output exergy by exhalation and evaporation of sweat is much lower[3] in hot and humid environmental conditions than in hot and dry environmental conditions, because the sweat cannot effectively evaporate due to high relative humidity. The black line ($\dot{L}$ = 0) in the figure presents conditions where internal heat production is equal to the sum of heat losses from the body, which results in comfort conditions (*PMV* = 0, calculated with Eq. (25) in Annex B). For a healthcare worker the conditions with T_{ai} 22–24 °C and RH_{in} 30–96 % result in comfort conditions. Hot and dry environmental conditions (35 °C and 30 %) and cold and dry environmental conditions (15 °C and 30 %) result in similar exergy consumption rates (4.378 W/m² and 4.376 W/m² for healthcare worker) (Figure 47). The difference appears if the whole human–body exergy balance is taken into consideration. High temperature and low RH_{in} (35 °C and 30 %, respectively) cause lower metabolic thermal exergy rate[4]

[1] Warm convective exergy rate discharged from the whole skin and clothing surfaces is lower due to smaller difference of T_{cl}–T_{ai} and larger value of T_{ai}/T_{cl}.

[2] The rate of stored exergy is lower due to larger value of T_{ai}/T_{cr} and larger value of T_{ai}/T_{sk}.

[3] The exergy rate of exhalation and evaporation of sweat is lower due to smaller value of $p_{vs}(T_{cr})/p_{vr}$.

[4] Metabolic thermal exergy rate is lower at conditions 35 °C and 30 % than at conditions 15 °C and 30 % due to smaller difference of T_{cr}–T_{ai} and smaller value of T_{cr}/T_{ai}, smaller difference of T_{sk}–T_{ai} and smaller value of T_{sk}/T_{ai}.

(by 1.73 W/m^2), lower exergy rate of exhalation and evaporation of sweat[1] (by 0.39 W/m^2), lower warm radiant exergy rate discharged from the whole skin and clothing surfaces[2] (by 0.50 W/m^2), lower warm convective exergy rate from the whole skin and clothing surfaces[3] (by 0.84 W/m^2) and higher rate of stored exergy[4] (by 4 mW/m^2) than at conditions with low T_{ai} and low RH_{in} (15 °C and 30 %, respectively) in case of healthcare worker.

The difference between human body exergy consumption rate for healthcare worker exposed to hot and humid environmental conditions (35 °C and 96 %) and cold and humid environmental conditions (15 °C and 96 %) is more significant (by 3.36 W/m^2). In cold and humid environmental conditions there appears higher metabolic thermal exergy rate[5] (by 4.49 W/m^2) and higher rate of output exergy by exhalation and evaporation of sweat[6] (by 0.24 W/m^2), higher warm radiant exergy rate discharged from the whole skin and clothing surfaces[7] (by 0.51 W/m^2) and higher warm exergy rate transferred by convection form the whole skin and clothing surfaces[8] (by 0.85 W/m^2) and lower rate of stored exergy[9] (by 0.235 W/m^2) than in hot and humid environmental conditions (35 °C and 96 %) in case for healthcare worker.

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[1] The exergy rate of exhalation and evaporation of sweat is lower at conditions 35 °C and 30 % than at conditions 15 °C and 30 % due to smaller difference of T_{cr}–T_{ai} and smaller value of T_{cr}/T_{ai}, smaller difference of T_{cl}–T_{ai} and smaller value of T_{cl}/T_{ai}.

[2] Warm radiant exergy rate discharged from the whole skin and clothing surfaces is smaller at conditions 35 °C and 30 % than at conditions 15 °C and 30 % due to lower difference of T_{cl} and T_{ai}.

[3] Warm convective exergy rate discharged from the whole skin and clothing surfaces is lower at conditions 35 °C and 30 % than at conditions 15 °C and 30 % due to smaller difference of T_{cl}–T_{ai} and larger value of T_{ai}/T_{cl}.

[4] Conditions 35 °C and 30 % result in 4 mW/m^2 of the rate of thermal exergy stored in the core and in the shell. Conditions 15 °C and 30 % results in zero rate of stored exergy, because of 0.885 W/m^2 of the thermal exergy rate is released from the core and the shell.

[5] Metabolic thermal exergy rate is higher at conditions 15 °C and 96 % than at conditions 35 °C and 96 % due to larger difference of T_{cr}–T_{ai} and larger value of T_{cr}/T_{ai}, larger difference of T_{sk}/T_{ai} and larger value of T_{sk}/T_{ai}.

[6] The rate of exhalation and evaporation of sweat is higher at conditions 15 °C and 96 % than at conditions 35 °C and 96 % due to larger difference of T_{cr}–T_{ai} and larger value of T_{cr}/T_{ai}, larger difference of T_{cl}–T_{ai} and larger value of T_{cl}/T_{ai}.

[7] Warm radiant exergy rate discharged from the whole skin and clothing surfaces is higher at conditions 15 °C and 96 % than at conditions 35°C and 96 % due to larger difference of T_{cl}–T_{ai}.

[8] Warm convective exergy rate discharged from the whole skin and clothing surfaces is higher at conditions 15 °C and 96 % than at conditions 35 °C and 96 % due to larger difference of T_{cl}–T_{ai} and smaller value of T_{ai}/T_{cl}.

[9] Conditions 35 °C and 96 % result in 0.235 W/m^2 of the rate of thermal exergy stored in the core and in the shell. Conditions 15 °C and 96 % result in zero rate of stored exergy, because 0.786 W/m^2 of the thermal exergy rate is released from the core and the shell.

Hot and humid environmental conditions in a burn patient room do not present comfort conditions for a healthy person such as healthcare worker. Higher or lower temperatures that recommended by ANSI/ASHRAE Standard 55 (T_{ai} between 20 °C and 25 °C, RH_{in} 30 % up to 60 %) could lead to decreased psychophysiological performance of healthy individuals. Pilcher et al. (2002) found that exposure to temperatures > 21 °C causes decrements in psychophysiological performance of 5.95 % in relation to neutral temperature (i.e. 16–21 °C), while exposure to the temperature range of 21–27 °C causes minimal changes in performance (0.8 % decrement), 27–32 °C causes a 7.5 % decrement in performance, above 32 °C causes a 14.88 % decrement in performance. Exposure to temperature below 10 °C or above 32 °C in the period less than 120 minutes reduces psychophysical capabilities by approx. 15 % (Sudoł–Szopińska and Tarnowski, 2007). Moreover, high T_{ai} and RH_{in} cause increased value of sweat rate[1] and skin wettedness[2] (Tanabe et al., 1987; Berglund and Cunningham, 1986; Atmaca and Yigit, 2006). But sweating itself does not cool the healthcare worker's body, the sweat must evaporate. For effective evaporation[3] and cooling mechanism of a healthcare worker's body much drier environment is required. For example, in hot and dry environmental conditions the exergy rate of evaporation of sweat in case of healthcare worker is 170 mW/m^2, while it is 2 mW/m^2 in hot and humid environmental conditions[4]. In such humid environment, water secreted from sweat glands cannot evaporate into the ambient environment and the body cannot be cooled. The heat has to be released with other mechanisms. For example, higher air velocity around the human body leads to convective cooling and creates better comfort conditions.

[1] Individual sweat rate depends on environmental conditions, exercise intensity, fitness level, degree of acclimatization, hydration status, health status (Plowman and Smith, 2007).

[2] Gagge et al. (1969) defined skin wettedness as the ratio of actual evaporative heat loss to maximum possible evaporative heat loss under the same environmental conditions. Skin wettedness can be estimated from a set of physiological and physical parameters, i.e. vapour pressure at the skin, saturated vapour pressure at skin, temperature and vapour pressure of the surrounding air (Kerslake, 1972; Berglund and Cunninggam, 1986). Higher skin wettedness also causes a suppression of sweat glands (Nadel and Stolwijk, 1973) and also leads to warm discomfort (Berglund and Cunningham, 1986; Tanabe et al., 1987).

[3] Evaporation is typically a mass diffusion phenomenon, where the liquid sweat is converted into vapour liquid phase. The energy required to change the phase is extracted from immediate surroundings, leading to lower temperatures, and it is called latent heat of vaporization. It is a primary defence against the heat stress and is a major mechanism of cooling the body.

[4] The exergy rate of exhalation and evaporation of sweat is higher in hot and dry environmental conditions due to larger value of $p_{vs}(T_{cr})/p_{vr}$.

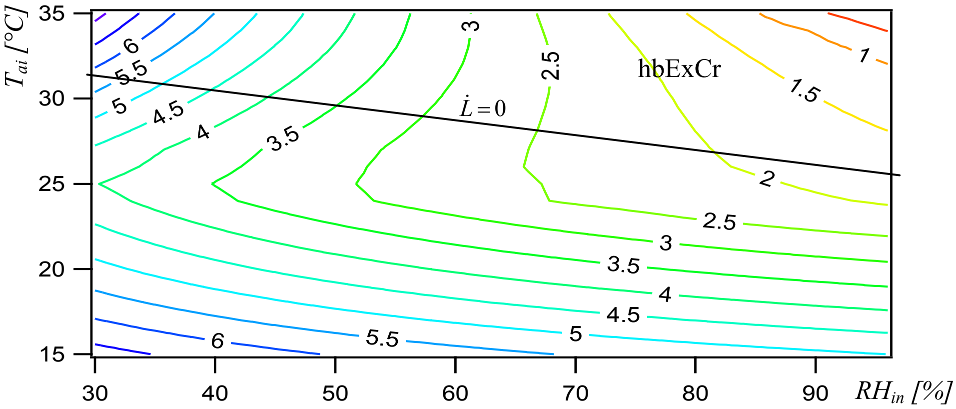

Fig 48: *Human body exergy consumption rate (hbExCr) [W/m^2], burn patient, as a function of T_{ai} (= T_{mr}) [°C] and RH_{in} [%], conventional system. The black line presents the rate of thermal load on the body surface area ($\dot{L}$ = 0, PMV = 0)*

But in case of burn patient room, higher values of air velocity are not permitted. Burn patient is from thermodynamic point of view a very specific subject. Regarding medical data (Cope et al., 1953; Caldwell et al., 1966, 1981, 1992; Zawacki et al., 1970; Wilmore et al., 1973a, 1973b, 1974a, 1974b, 1975; Neely et al., 1974; Caldwell, 1976, 1991; Danielsson et al., 1976; Arturson et al., 1978; Herndon, 1981, 1996; Herndon et al., 1987a, 1987b, 1988; Cone et al., 1988; Carlson at al., 1992; Fleming et al., 1992; Martin et al., 1992; Herndon and Parks, 1992; Shiozaki et al., 1993; Wallace et al., 1994; Kelemen et al., 1996; Wedler et al., 1999; Ramos et al., 2002; Herndon and Tompkins, 2004; Atmaca and Yigit, 2006; ASHRAE Handbook HVAC applications …, 2007; Corralo et al., 2007), he/she has higher skin temperature, lower T_{cr} and higher metabolic rate than healthy individuals. Individual specifics have great influence on separate parts of human body exergy balance and lead to higher rates. In case of burn patient, human body exergy consumption rate in a test active space with conventional system varies from 0.17 to 7.17 W/m^2, as can be seen form Figure 48. Let us focus on the required conditions for hospital room for burn patient, i.e. 32 °C, 95 %. The results of calculations show that the exergy consumption rate valid for thermoregulation is the lowest possible[1] in the required conditions. And moreover, internal heat production and the sum of heat losses from the body are in these conditions equal (marked with black line). Regarding the

[1] Required conditions result in the lowest exergy consumption rate of the burn patient body, mainly due to minimum rate of input exergy by metabolic thermal exergy rate.

equation for the calculation of PMV index (Eq. (25), Annex B), the conditions where internal heat production is equal to the sum of heat losses from the body result in $PMV = 0$. However, PMV calculation is more correct for healthy individuals. If we expose the burn patient to conditions different than the required ones, for example cold and dry environment, the rate of exergy consumption will become much higher due to higher metabolic thermal exergy rate[1]. This will result in thermally uncomfortable and unhealthy conditions. Moreover, the difference appears also between subjects exposed to the same environmental conditions. For example, at conditions (15 °C, RH_{in} 30 %) exergy consumption rate is 6.70 W/m^2 for patient and 4.38 W/m^2 for healthcare worker. The rate of stored exergy is zero for healthcare worker and patient, because exergy is released from the core and the shell. Higher difference between T_{cl} and T_{ai} in case of burn patient than healthcare worker causes higher rates of warm radiative and convective exergies that are discharged from patient's body to the surrounding space. Warm radiant exergy rate and warm convective exergy rate discharged from the patient body to the surrounding space are 0.78 W/m^2 and 2.33 W/m^2 in case of burn patient, and 0.51 W/m^2 and 0.84 W/m^2 in case of healthcare worker[2]. Moreover, metabolic thermal exergy rate[3] and the exergy rate of evaporation and exhalation of sweat[4] are approx. 1.8 times higher in case of burn patient than in case of healthcare worker, as can be seen from Table 32.

[1] Metabolic thermal exergy rate of burn patient in cold and dry environmental conditions is higher than in required conditions, mainly due to larger difference of $T_{cr}–T_{ai}$ and larger value of T_{cr}/T_{ai}, larger difference of $T_{sk}–T_{ai}$ and larger value of T_{sk}/T_{ai} and larger value of $p_{vs}(T_{ai})/p_{vr}$.

[2] Warm convective exergy rate discharged from the whole skin and clothing surfaces is higher for burn patient than for healthcare worker due to larger difference of $T_{cl}–T_{ai}$ and smaller value of T_{ai}/T_{cl}. Warm radiant exergy rate discharged from the whole skin and clothing surfaces is higher for burn patient than for healthcare worker due to larger difference of $T_{cl}–T_{ai}$.

[3] Metabolic thermal exergy rate of burn patient is higher than for healthcare worker, mainly due to higher metabolic rate, larger difference of $T_{sk}–T_{ai}$ and larger value of T_{sk}/T_{ai}.

[4] The exergy rate of exhalation and evaporation of sweat is higher for burn patient than for healthcare worker due to larger difference of $T_{cl}–T_{ai}$ and larger value of T_{cl}/T_{ai}.

Table 32: *Results of human body exergy calculations for extreme conditions. The exergy values are expressed as rates, W/m² (body surface)*

Subject	***Conditions T_{ai} [°C], RH_{in} [%]*** [1]	***PMV index***	***T_{cr}, T_{sk}, T_{cl} [°C]***	***Cool (C), Warm (W) radiant exergy in [W/m²]***	***Cool (C), Warm (W) convective exergy in [W/m²]***	***Breath air in [W/m²]***	***From inner part [W/m²]***	***Exergy consumption [W/m²]***	***Stored exergy [W/m²]***	***Exhalation, sweat out [W/m²]***	***Cool (C), Warm (W) radiation out [W/m²]***	***Cool (C), Warm convection out [W/m²]***
Healthcare worker	*35°C 30 %*	*2.1*	*37.0 36.0 35.6*	*0*	*0*	*0*	*4.56*	*4.38*	*0.004*	*0.17*	*C = 0 W = 0.003*	*C = 0 W = 0.004*
	15°C 30 %	*−1.4*	*36.8 29.5 23.3*	*0*	*0*	*0*	*6.29*	*4.38*	*0*	*0.56*	*C = 0 W = 0.51*	*C = 0 W = 0.84*
	35°C 96 %	*3.0*	*37.6 37.3 36.2*	*0*	*0*	*0*	*0.83*	*0.32*	*0.235*	*0.002*	*C = 0 W = 0.01*	*C = 0 W = 0.02*
	15°C 96 %	*−1.0*	*36.8 29.8, 23.4*	*0*	*0*	*0*	*5.32*	*3.68*	*0*	*0.24*	*C = 0 W = 0.52*	*C = 0 W = 0.87*
Burn patient	*35°C 30 %*	*0.2*	*35.5 37.0 36.2*	*0*	*0*	*0*	*7.52*	*7.17*	*0.001*	*0.31*	*C = 0 W = 0.01*	*C = 0 W = 0.03*
	15°C 30 %	*−1.7*	*35.5 37.0, 25.3*	*0*	*0*	*0*	*10.82*	*6.70*	*0*	*1.01*	*C = 0 W = 0.78*	*C = 0 W = 2.33*
	35°C 96 %	*3.0*	*35.5 37.0 37.1*	*0*	*0*	*0*	*0.40*	*0.17*	*0.048*	*0.002*	*C = 0 W = 0.04*	*C = 0 W = 0.09*
	15°C 96 %	*−1.3*	*35.5 37.0 25.8*	*0*	*0*	*0*	*8.84*	*5.00*	*0*	*0.43*	*C = 0 W = 0.86*	*C = 0 W = 2.55*
Visitor	*35°C 30 %*	*2.1*	*37.3 36.4 35.6*	*0*	*0*	*0*	*8.49*	*8.14*	*0.31*	*0.015*	*C = 0 W = 0.003*	*C = 0 W = 0.009*
	15°C 30 %	*−0.8*	*36.9 31.9 23.3*	*0*	*0*	*0*	*9.80*	*6.72*	*0*	*1.03*	*C = 0 W = 0.51*	*C = 0 W = 1.54*
	35°C 96 %	*3.0*	*38.6 38.0 36.4*	*0*	*0*	*0*	*2.09*	*0.73*	*0.648*	*0.006*	*C = 0 W = 0.02*	*C = 0 W = 0.04*
	15°C 96 %	*−0.6*	*36.9 32.6 23.7*	*0*	*0*	*0*	*8.29*	*5.61*	*0.44*	*0.0002*	*C = 0 W = 0.56*	*C = 0 W = 1.67*

Table 32 presents the results of human body exergy balance and *PMV* index in extreme environmental conditions (hot/cold; humid/dry). As mentioned in the introduction, studies (Isawa et. al., 2003; Shukuya et al., 2003; Shukuya, 2006a; Simone et al., 2010, 2011a, 2011b) show that in the framework of thermal comfort conditions of human body where only the effect of temperature is taken into consideration, the rate of exergy consumption is minimal. But in the present study, where also the effect of relative humidity and individual characteristics are taken into consideration, at thermal comfort conditions (*PMV* index close to 0) higher human body exergy consumption rates appear. This is because there is some inevitable

[1] For our calculations T_{ai} was assumed to be equal to T_{ao}. RH_{in} was assumed to be equal to RH_{out}.

chemical exergy consumption inside the human body, which causes the generation of quite a large amount of entropy that has to be discarded. Otherwise, the human body cannot maintain the required homeostatic conditions. For example, in 35 °C and 30 % the human body exergy rate is 4.38 W/m^2 and the *PMV* index is 2.1 for healthcare worker, 7.17 W/m^2 and 0.2 for burn patient and 8.14 W/m^2 and 2.1 for visitor[1]. The lowest possible exergy consumption rates 0.32 W/m^2 for healthcare worker, 0.17 W/m^2 for burn patient, 0.73 W/m^2 for visitor result at 35 °C and 96 % where most of people feel uncomfortable (*PMV* index = 3.0)[2].

From this point of view, it is very important to make an in–depth analysis of all exergy inputs, outputs and stored exergies. However, the *PMV* index is 0 when T_{ai} is equal to 22–24 °C for healthcare worker, 26–32 °C for burn patient and 17–21 °C for visitor, independently of RH_{in}, as it is shown in Figures 47–49. Koch et al. (1960) concluded that humidity had negligible effect on thermal comfort up to 60 % RH_{in} and 18 °C. Below these levels, T_{ai} alone was the governing factor.

Similar conclusions as for burn patient could also be made in the case of visitor, i.e. regarding high metabolic rate. In a test active space with conventional system human body exergy consumption rate varies from 0.73 to 8.14 W/m^2. The lowest possible human body exergy consumption rate can be found at high T_{ai} and high RH_{in} (35 °C, 96 %)[3] and the highest exergy consumption rate can be found at conditions with high T_{ai} and low RH_{in} (35 °C, 30 %)[4]. The main difference is that at conditions with the lowest possible human body exergy consumption rate, the visitor does not feel comfortable (35 °C, 96 % results in 0.73 W/m^2 and *PMV* = 3.0), as can be seen from Figure 49 and Table 32.

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[1] Conditions 35 °C and 30 % result in the highest exergy consumption rate for all subjects, due to high rate of input exergy by metabolic thermal exergy rate, and low rates of output exergies.

[2] Conditions 35 °C and 96 % result in the lowest exergy consumption rate, mainly due to minimum rate of input exergy by metabolic thermal exergy rate.

[3] Conditions 35 °C and 96 % result in the lowest exergy consumption rate for visitor than in other conditions, mainly due to lower input exergy rate by metabolic thermal exergy rate, and higher rate of stored exergy.

[4] Conditions 35 °C and 30 % result in the highest exergy consumption rate, mainly due to higher rate of input exergy by metabolic thermal exergy rate and lower rate of stored exergy than at conditions 35 °C and 96 %, where human body exergy consumption rate is minimal.

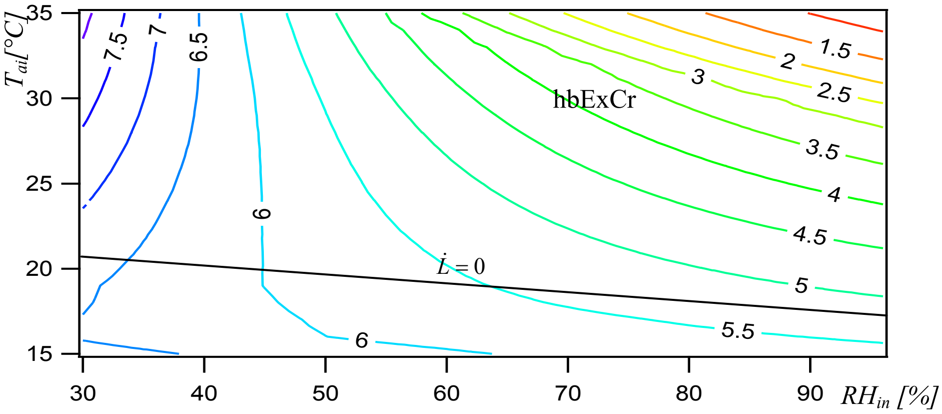

Fig 49: *Human body exergy consumption rate (hbExCr) [W/m²], visitor, as a function of T_{ai} (= T_{mr}) [°C] and RH_{in} [%], conventional system. The black line presents the rate of thermal load on the body surface area ($\dot{L}$ = 0, PMV = 0)*

The highest variations in exergy consumption rates between individuals appear in extreme conditions with high T_{ai} and low RH_{in}, i.e. 2.79 W/m^2 for burn patient–healthcare worker or 3.76 W/m^2 for visitor–healthcare worker. The highest exergy consumption rate among individuals, results in the case of visitor (8.14 W/m^2) at conditions with 35 °C and 30 %, due to higher rate of input exergy by metabolic thermal exergy rate than in the case of burn patient and healthcare worker exposed to the same conditions[1]. The lowest possible consumption rate results in the conditions of high T_{ai} and RH_{in} (35 °C, 96 %) in case of burn patient (0.17 W/m^2), due to lower rate of input exergy rate by metabolism than in the case of visitor and healthcare worker exposed to the same conditions[2].

It could be concluded that the required demands and recommendations (ASHRAE Handbook HVAC applications …, 2007; Herndon, 1996: 492) in hospital room for burn patient are health conditions for patient, where patient demands have to be taken as priority. In case of burn patient the required conditions cause the lowest possible internal production by metabolism and also the lowest evaporation beside low convection and radiation, as it was mentioned by Herndon (1981) and Herndon

[1] In case of a visitor at conditions with 35 °C and 30 % the metabolic thermal exergy rate is the highest among subjects due to higher metabolic rate, the largest difference of T_{cr}–T_{ai} and the largest value of T_{cr}/T_{ai} among subjects exposed to the same conditions.

[2] In case of a burn patient at conditions with 35 °C and 96 % the metabolic thermal exergy rate is the lowest among subjects exposed to the same conditions due to the smallest difference of T_{cr}–T_{ai} and the smallest value of T_{cr}/T_{ai}, the smallest difference of T_{sk}–T_{ai} and smallest value of T_{sk}/T_{ai}.

(1996: 492) (Table 32)[1]. The worst case is high air temperature and low RH_{in}, where the exergy consumption rate is the highest for burn patient (7.17 W/m^2)[2]. It is interesting that PMV is in this case lower (0.24).

The same conclusions were made by Wilmore et al. (1975) and Herndon (1981). Wilmore et al. (1975) found out that hypermetabolism can be attenuated by increasing the ambient temperature from 25 to 33 °C. Herndon (1981) concluded that metabolic rate is the lowest possible in ambient temperature of 35 °C (Figure 50). In these two studies, only the air temperature was taken into consideration, but not RH_{in}.

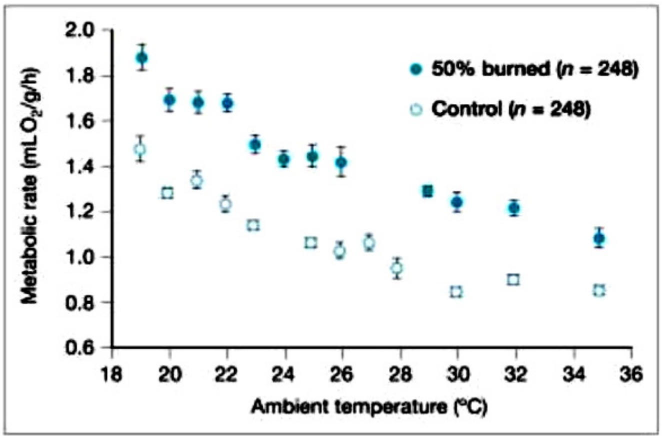

Fig 50: *Effect of T_{ai} on metabolic rate (50 % burned and control rats) (Herndon, 1981: 702)*

[1] In case of a burn patient at conditions with 35 °C and 96 % the metabolic thermal exergy rate is lower than in other conditions due to smaller difference of T_{cr}–T_{ai} and smaller value of T_{cr}/T_{ai}, smaller difference of T_{sk}–T_{ai} and smaller value of T_{sk}/T_{ai} and smaller value of $p_{vs}(T_{ai})/p_{vr}$. Warm convective exergy rate discharged from the whole skin and clothing surfaces is low due to low difference of T_{cl}–T_{ai} and large value of T_{ai}/T_{cl}. Warm radiant exergy rate discharged from the whole skin and clothing surfaces is low due to low difference of T_{cl}–T_{ai}. The exergy rate of exhalation and evaporation of sweat is low due to small difference of T_{cl}–T_{ai} and small value of T_{cl}/T_{ai}, and small difference of T_{cr}–T_{ai} and small value of T_{cr}/T_{ai}, and small value of $p_{vs}(T_{cr})/p_{vr}$. Conditions 35 °C and 96 % results in 48 mW/m^2 of the rate of thermal exergy stored in the core and in the shell.

[2] In case of a burn patient conditions with 35 °C and 30 % result in the highest human body exergy consumption rate due to high rate of input exergy by metabolic thermal exergy rate mainly due to large value of $p_{vs}(T_{ai})/p_{vr}$. Warm convective exergy rate discharged from the whole skin and clothing surfaces is low due to low difference of T_{cl}–T_{ai} and large value of T_{ai}/T_{cl}. Warm radiant exergy rate discharged from the whole skin and clothing surfaces is low due to low difference of T_{cl}–T_{ai}. The exergy rate of exhalation and evaporation of sweat is high due to large value of $p_{vs}(T_{cr})/p_{vr}$. Conditions 35 °C and 30 % results in 1 mW/m^2 of the rate of thermal exergy stored in the core and in the shell.

7.4.3.2 LowEx system

The data from Table 33 show that the human body exergy rates [W/m^2] also vary between individuals, like in the test active space equipped with conventional system. Human body exergy consumption rates for burn patient and healthcare worker are lower, if the surfaces are heated to higher temperature than that of air. Such conditions result also in *PMV* index closer to 0. It means that better comfort conditions could be created with LowEx systems. For example, the human body exergy consumption rate in burn patient is 2.55 W/m^2 at conditions T_{ai} = 15 °C, T_{mr} = 35 °C, and 3.66 W/m^2 at conditions T_{ai} = 35 °C, T_{mr} = 15 °C. Even more, the exergy consumption rate in case T_{ai} = 15 °C, T_{mr} = 35 °C is almost the same for healthcare worker (2.59 W/m^2) and for burn patient (2.55 W/m^2). If the surfaces are heated, the exergy consumption rate is lower, especially due to much larger rates of output exergies by exhalation and evaporation of sweat, warm radiation and warm convection discharged from the human body into the surrounding space than at conditions with $T_{mr} < T_{ai}$[1]. This affects considerably the comfort and health conditions of human body described in detail in section 7.2.4 (Radiant exergy flow rate from interior surfaces). Moreover, conditions with higher T_{mr} and lower T_{ai} result in warm radiant exergy rate absorbed by the whole skin and clothing surfaces as an input exergy, and warm radiant exergy rate and warm convective exergy rate transferred from the whole skin and clothing surfaces as an output exergies. Conditions with lower T_{mr} and higher T_{ai} result in cool radiant exergy rate absorbed by the whole skin and clothing surfaces as an input exergy, and cool radiant exergy rate and cool convective exergy rate transferred from the whole skin and clothing surfaces as an output exergies.

[1] At conditions with $T_{mr} > T_{ai}$ much higher metabolic thermal exergy rate appears for all subjects due to much larger difference of $T_{cr}–T_{ai}$ and larger value of T_{cr}/T_{ai}, larger difference of $T_{sk}–T_{ai}$ and larger value of T_{sk}/T_{ai} than at conditions with $T_{mr} < T_{ai}$. Warm radiant exergy rate absorbed by the whole skin and clothing surfaces appears due to $T_{mr} > T_{ai}$. Warm radiant exergy rate and warm convective exergy rate discharged from the whole skin and clothing surfaces appear due to $T_{cl} > T_{ai}$. The exergy rate of exhalation and evaporation of sweat is higher due to larger difference of $T_{cr}–T_{ai}$ and larger value of $T_{cr}/T_{ai,}$ larger difference of $T_{cl}–T_{ai}$ and larger value of T_{cl}/T_{ai}. The rate of stored exergy is lower at conditions with $T_{ai} < T_{mr}$ in case of healthcare worker, visitor and burn patient than at conditions with $T_{ai} > T_{mr}$. In case of burn patient conditions $T_{ai} < T_{mr}$ results in zero rate of stored exergy, because exergy is released from the core and the shell.

Table 33: *Human body exergy balance and PMV index, 80 % RH_{in}, LowEx system. The exergy values are expressed as rates, W/m² (body surface)*

Subject	T_{ai}, T_{mr} [°C][1]	T_{cr}, T_{sk}, T_{cl} [°C]	*PMV index*	*Cool (C), Warm (W) radiant exergy in [W/m²]*	*Cool (C), Warm (W) convective exergy in [W/m²]*	*Breath air in [W/m²]*	*From inner part [W/m²]*	*Exergy consumption [W/m²]*	*Stored exergy [W/m²]*	*Exhalation, sweat out [W/m²]*	*Cool (C), Warm (W) radiation out [W/m²]*	*Cool (C), Warm (W) convection out [W/m²]*
Health. worker	15.0 35.0	36.9 34.5 31.1	0.4	C = 0 W = 3.06	0	0	4.85	2.59	0.004	0.29	C = 0 W = 1.94	C = 0 W = 3.07
Burn patient	15.0 35.0	35.5 37.0 31.7	−1.0	C = 0 W = 3.06	0	0	8.16	2.55	0	0.51	C = 0 W = 2.11	C = 0 W = 6.03
Visitor	15.0 35.0	36.9 35.0 29.2	0.4	C = 0 W = 3.06	0	0	8.95	5.55	0.019	0.53	C = 0 W = 1.51	C = 0 W = 4.37
Health. worker	35.0 15.0	36.9 34.3 29.7	2.6	C = 3.06 W = 0	0	0	0.58	3.07	0.081	0.01	C = 0.22 W = 0	C = 0.33 W = 0
Burn patient	35.0 15.0	35.5 37.0 34.6	1.2	C = 3.06 W = 0	0	0	0.63	3.66	0.0003	0.02	C = 0.001 W = 0	C = 0.003 W = 0
Visitor	35.0 15.0	37.3 36.3 31.4	3.0	C = 3.06 W = 0	0	0	1.63	4.15	0.063	0.02	C = 0.11 W = 0	C = 0.28 W = 0

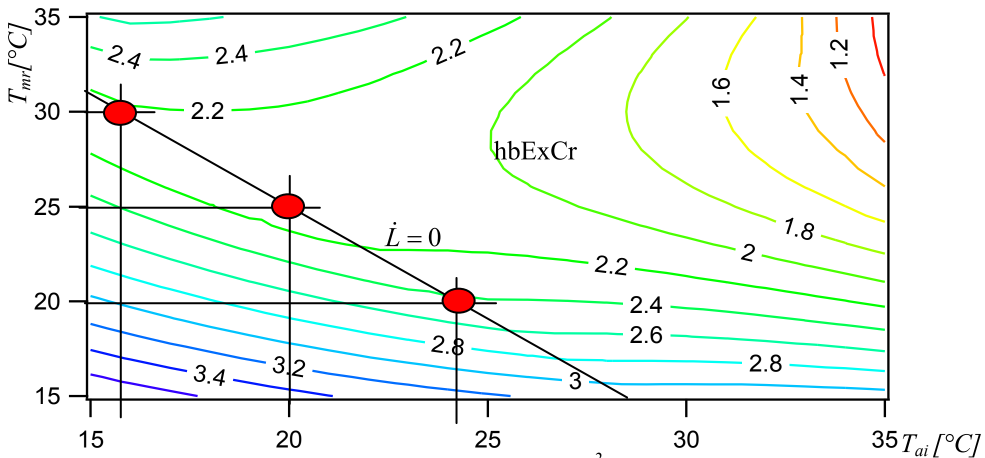

Fig 51: *Human body exergy consumption rate (hbExCr) [W/m²], healthcare worker, as a function of T_{ai} and T_{mr}, 80 % RH_{in}, LowEx system. The black line presents the rate of thermal load on the body surface area ($\dot{L}$ = 0, PMV = 0)*

Figure 51 presents human body exergy consumption rates as a function of T_{ai} and T_{mr} for healthcare worker in a test room with LowEx system (T_{ai} differs from T_{mr}, 80 % RH_{in}). If T_{ai} is changed in the range 15–35 °C and T_{mr} in the range of 15–35 °C, the exergy consumption rate for healthcare worker varies from 0.9–3.8 W/m².

[1] For our calculations T_{ai} was assumed to be equal to T_{ao}. RH_{in} was assumed to be equal to RH_{out}.

The lowest exergy consumption rate results at conditions of high T_{ai} and high T_{mr} (35 °C, 35 °C)[1] and the highest at conditions of low T_{ai} and low T_{mr} (15 °C, 15 °C)[2]. Moreover, $\dot{L} = 0$ ($PMV = 0$) at conditions of T_{ai} 16 °C, T_{mr} 30 °C with 2.2 W/m^2 of exergy consumption rate; at 20 °C T_{ai} and 25 °C T_{mr} with 2.1 W/m^2 of exergy consumption rate and also at 24 °C T_{ai} and 20 °C T_{mr} with 2.4 W/m^2 of exergy consumption rate (Figure 51). From this perspective it can be concluded that exergy consumption rate is not the lowest possible in comfort conditions, where $\dot{L} = 0$ ($PMV = 0$).

Comfort conditions could be designed with combination of T_{ai} and T_{mr} and also human body exergy consumption rate could be regulated with combination of T_{mr} and T_{ai}. This aspect is further presented in section part 7.6 (Active regulation).

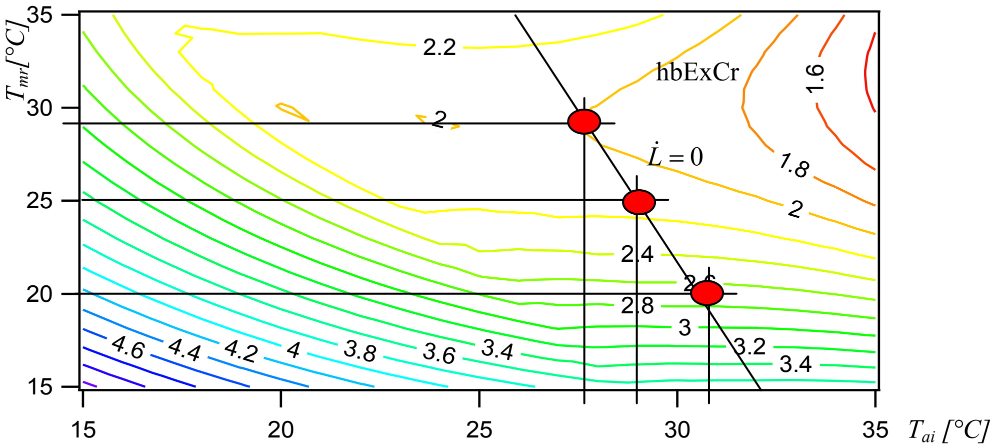

Fig 52: *Human body exergy consumption rate (hbExCr) [W/m^2], burn patient, as a function of T_{ai} and T_{mr}, 80 % RH_{in}, LowEx system. The black line presents the rate of thermal load on the body surface area ($\dot{L} = 0$, $PMV = 0$)*

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[1] Conditions T_{ai} 35 °C, T_{mr} 35 °C and RH_{in} 80 % result in the lowest exergy consumption rate for healthcare worker due to minimal rate of input exergy by metabolic thermal exergy rate, and higher rate of stored exergy than at other temperature conditions and RH_{in} 80 %. Lower metabolic thermal exergy rate appears due to smaller difference of T_{cr}–T_{ai} and smaller value of T_{cr}/T_{ai}, smaller difference of T_{sk}–T_{ai} and smaller value of T_{sk}/T_{ai}. Warm radiant exergy rate and warm convective exergy rate discharged from the whole skin and clothing surfaces are lower due to smaller difference of T_{cl} –T_{ai} and larger value of T_{ai}/T_{cl}. The exergy rate of exhalation and evaporation of sweat is lower due to smaller difference of T_{cr}–T_{ai} and smaller value of T_{cr}/T_{ai}, smaller difference of T_{cl}–T_{ai} and smaller value of T_{cl}/T_{ai}. The rate of stored exergy is higher at conditions T_{ai} 35 °C, T_{mr} 35 °C and RH_{in} 80 % than at conditions T_{ai} 15 °C, T_{mr} 15 °C and RH_{in} 80 % where exergy is released from the core and from the shell.

[2] Conditions T_{ai} 15 °C, T_{mr} 15 °C and RH_{in} 80 % result in the highest exergy consumption rate for healthcare worker due to high rate of input exergy by metabolic thermal exergy rate than at other temperature conditions and RH_{in} 80 %. High metabolic thermal exergy rate appears due to the largest difference of T_{cr}–T_{ai} and the largest value of T_{cr}/T_{ai}, largest difference of T_{sk}–T_{ai} and largest value of T_{sk}/T_{ai}. Warm radiant exergy rate and warm convective exergy rate discharged from the whole skin and clothing surfaces are higher due to the largest difference of T_{cl}–T_{ai} and the smallest value of T_{ai}/T_{cl}. The rate of exhalation and evaporation of sweat is higher due to larger difference of T_{cr}–T_{ai} and larger value of T_{cr}/T_{ai}, larger difference of T_{cl}–T_{ai} and larger value of T_{cl}/T_{ai}. The rate of stored exergy is zero because exergy is released from the core and shell.

In case of burn patient human body exergy consumption rate in a test room with LowEx system varies from 1.4 to 5.3 W/m², as can be seen from Figure 52. The results of calculations show that the exergy consumption rate is the lowest possible inside the required conditions[1]. And moreover, $\dot{L} = 0$ ($PMV = 0$) at conditions T_{ai} 27 °C, T_{mr} 29 °C with 2.0 W/m² exergy consumption rate; 31°C T_{ai} and 20 °C T_{mr} with 2.7 W/m² exergy consumption rate and also at 28 °C T_{ai} and 25 °C T_{mr} and 2.2 W/m² exergy consumption rate.

In case of visitor (Figure 53) it can be concluded that exergy consumption rate ranges between 1.6 and 5.8 W/m². $\dot{L} = 0$ ($PMV = 0$) at conditions of T_{ai} 20 °C, T_{mr} 16 °C with 5.4 W/m² of exergy consumption rate and also at 17 °C T_{ai}, 25 °C T_{mr} with 5.0 W/m² exergy consumption rate and also at 15 °C T_{ai} and 29 °C T_{mr} with 5.1 W/m² of exergy consumption rate.

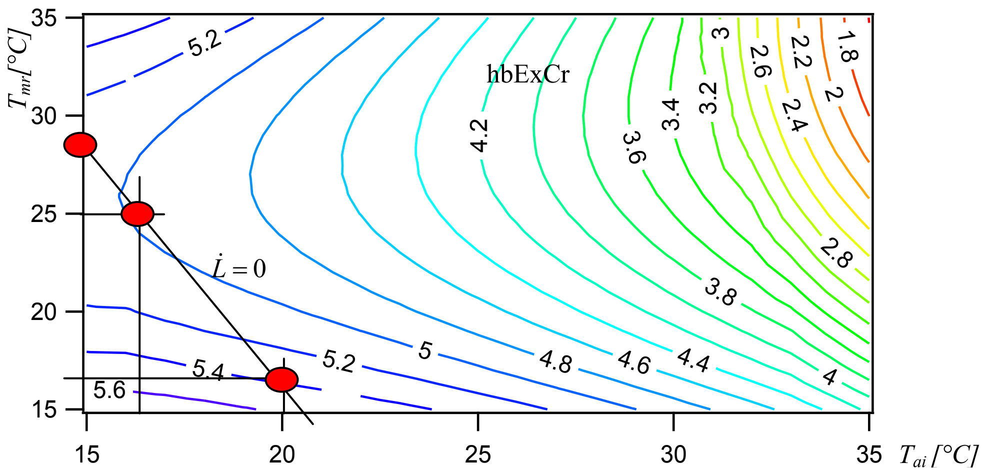

Fig 53: *Human body exergy consumption rate (hbExCr) [W/m²], visitor, as a function of T_{ai} and T_{mr}, 80 % RH_{in}, LowEx system. The black line presents the rate of thermal load on the body surface area ($\dot{L} = 0$, $PMV = 0$)*

[1] In case of a burn patient at conditions with T_{ai} 35 °C, T_{mr} 35 °C and 80 % human body exergy consumption rate is minimal due to minimal rate of input exergy by metabolic thermal exergy rate and higher rate of stored exergy than at other temperature conditions and RH_{in} 80 %. The metabolic thermal exergy rate is the lowest mainly due to the smallest difference of T_{cr}–T_{ai} and smallest value of T_{cr}/T_{ai}, smallest difference of T_{sk}–T_{ai} and smallest value of T_{sk}/T_{ai}. Warm radiant exergy rate discharged from the whole skin and clothing surfaces is lower due to lower difference of T_{cl}–T_{ai}. Warm convective exergy rate discharged from the whole skin and clothing surfaces is lower due to small difference of T_{cl}–T_{ai} and larger value of T_{ai}/T_{cl}. The exergy rate of exhalation and evaporation of sweat is lower due to smaller difference of T_{cl}–T_{ai} and smaller value of T_{cl}/T_{ai}, and smaller difference of T_{cr}–T_{ai} and smaller value of T_{cr}/T_{ai}. The rate of stored exergy is higher at conditions T_{ai} 35 °C, T_{mr} 35 °C and RH_{in} 80 % than at conditions T_{ai} 15 °C, T_{mr} 15 °C and RH_{in} 80 % where exergy is released from the core and from the shell.

For all cases, exergy consumption rate is lower if surfaces are heated to higher temperatures than that of air[1], and $\dot{L} = 0$ ($PMV = 0$). The attainment of required conditions, thermal comfort conditions, and minimal possible exergy consumption rate for visitor in test room is more pretentious than for other two subjects.

[1] In case of visitor conditions with $T_{mr} > T_{ai}$, $PMV = 0$ result in lower exergy consumption rate due to higher output exergies than at conditions with $T_{mr} < T_{ai}$, $PMV = 0$. Higher metabolic thermal exergy rate appears due to much larger difference of $T_{cr}–T_{ai}$ and larger value of T_{cr}/T_{ai}, larger difference of $T_{sk}–T_{ai}$ and larger value of T_{sk}/T_{ai}. Warm radiant exergy rate absorbed by the whole skin and clothing surfaces appears due to $T_{mr} > T_{ai}$. Warm radiant exergy rate and warm convective exergy rate discharged from the whole skin and clothing surfaces appear due to $T_{cl} > T_{ai}$. The exergy rate of exhalation and evaporation of sweat is higher due to larger difference of $T_{cr}–T_{ai}$ and larger value of T_{cr}/T_{ai}, larger difference of $T_{cl}–T_{ai}$ and larger value of T_{cl}/T_{ai}. The rate of stored exergy is lower than at conditions with $T_{mr} < T_{ai}$.

a)

b)

c)

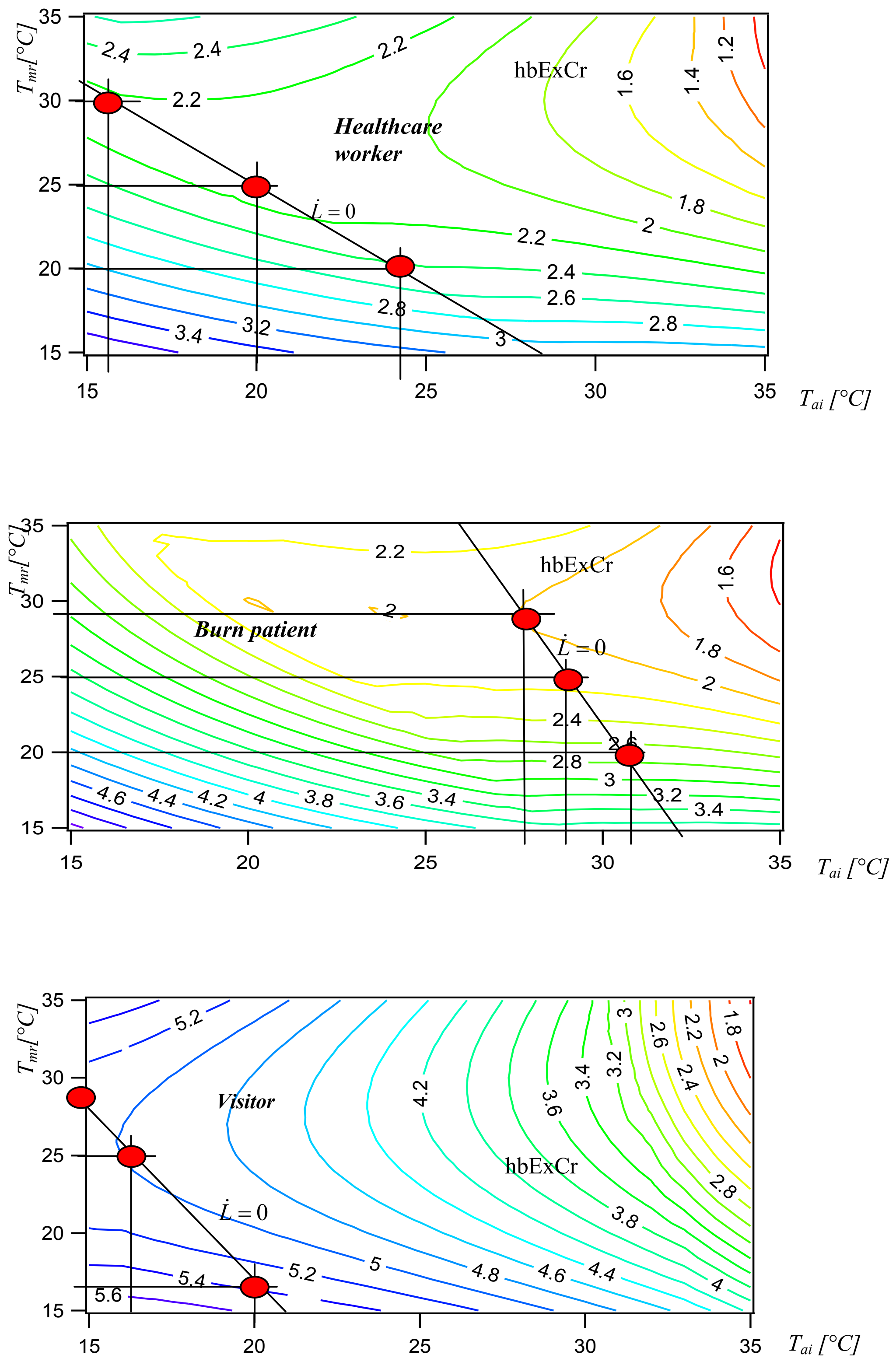

Fig 54: *Comparison between a) healthcare worker, b) burn patient, c) visitor. Human body exergy consumption rate (hbExCr) [W/m^2], as a function of T_{ai} and T_{mr}, 80 % RH_{in}, LowEx system. The black line presents the rate of thermal load on the body surface area ($\dot{L}$ = 0, PMV = 0)*

7.4.3.3 Conventional versus LowEx system

Regulation demands (ASHRAE Handbook HVAC applications ..., 2007) define that room temperature has to be up to 32 °C db and relative humidity up to 95 %. Recommendations (Herndon, 1996: 492) define values of ambient temperature and humidity that should be maintained at 30–33 °C and 80 %, in order to decrease energy demands and evaporative heat losses.

However, field regulations and recommendations define requirements for T_{ai} and RH_{in} and could be used for the room with conventional system. In case of test room with LowEx system, other requirements that include T_{mr}, T_{ai} and parameters connected with the individual user have to be found. Designed conditions in a space with LowEx system result from interactions between T_{ai} and T_{mr}. Required value for T_{ai} 32 °C is not enough to compare the real or simulated conditions inside the space with LowEx system. For this purpose, also T_o has to be taken into consideration. T_o is calculated with Eq. (1) described in Annex A.

Regarding recommendations for burn patient room (T_{ai} 32 °C, RH_{in} 80 %), the calculated T_o for burn patient is 32 °C and presents the recommended value for the conditions inside of burn patient room equipped with LowEx system. The goal is to find the optimal combination of T_{ai} and T_{mr} that results in the recommended T_o as is presented in Figure 55 with red lines. In case of conventional system, the required value is T_{ai} 32°C (ASHRAE Handbook HVAC applications …, 2007; Herndon, 1996: 492).

The two systems are compared due to the calculated human body exergy consumption rate [W/m^2] as a function of T_{ai} for conventional system and T_o for LowEx system. However, in a room with conventional system $T_{ai} = T_{mr} = T_o$ was assumed and in a room with LowEx system $T_{ai} \neq T_{mr} \neq T_o$ was assumed.

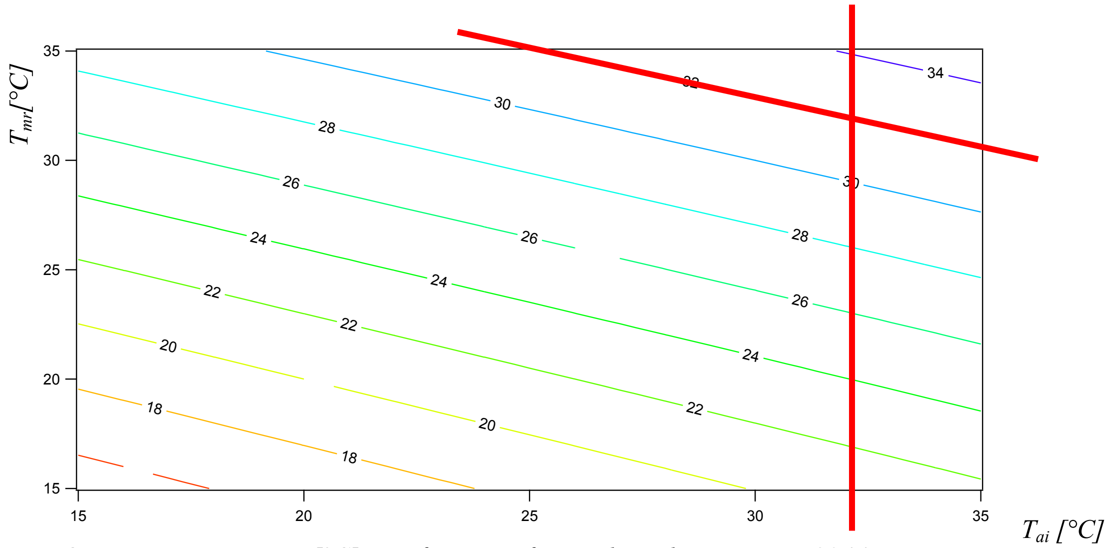

Fig 55: *Operative temperature [°C] as a function of T_{ai} and T_{mr}, burn patient, 80 % RH_{in}, LowEx system*

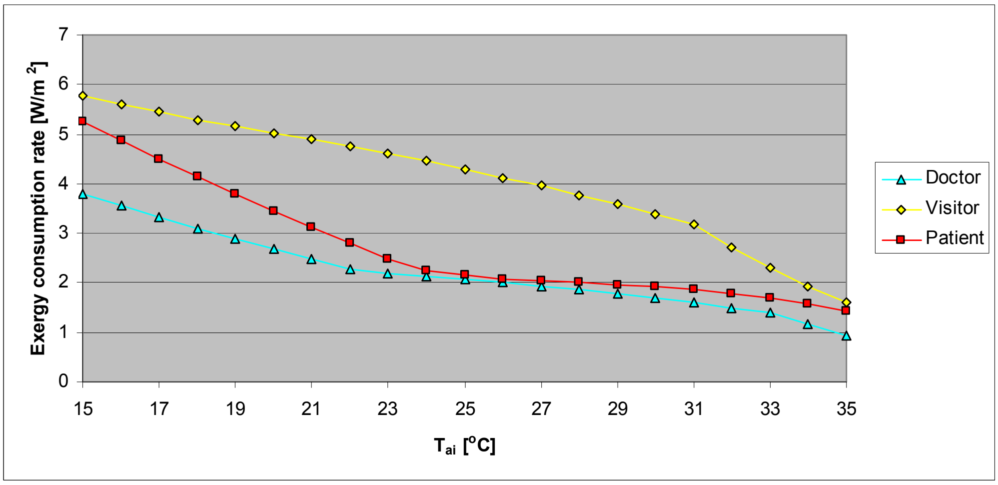

Fig 56: *Human body exergy consumption rate [W/m²] as a function of T_{ai}, 80 % RH_{in}, conventional system*

Figure 56 presents the human body exergy consumption rate as a function of T_{ai} for three subjects exposed to conditions 80 % RH_{in} in a room with conventional system. The maximal human body exergy consumption rate 5.78 W/m^2 could be found in the case of visitor exposed to T_{ai} 15 °C[1].

The minimal human body exergy consumption rate 0.93 W/m^2 results in the case of

[1] Visitor has the maximal human body exergy consumption rate among subjects at conditions T_{ai} 15 °C with 80 % RH_{in} due to high rate of input exergy with metabolic thermal exergy rate. Metabolic thermal exergy rate is high due to high metabolic rate, large difference of T_{cr}–T_{ai} and large value of T_{cr}/T_{ai}, large difference of T_{sk}–T_{ai} and large value of T_{sk}/T_{ai}. Warm radiant exergy rate discharged from the whole skin and clothing surfaces is low due to small difference of T_{cl}–T_{ai}. The exergy rate of exhalation and evaporation of sweat is low due to small difference of T_{cl}–T_{ai} and small value of T_{cl}/T_{ai}. The rate of stored exergy is high due to small value of T_{ai}/T_{cr} and small value of T_{ai}/T_{sk}.

healthcare worker at conditions of T_{ai} 35 °C[1]. Burn patient has minimal human body exergy consumption rate at conditions T_{ai} 35 °C (1.44 W/m^2) and maximal at conditions T_{ai} 15 °C (5.26 W/m^2)[2,3]. Results of previous research by Saito et al. (2000) considering T_{ao} equal to indoor T_{ai} show that human body exergy consumption rate has its minimum at T_o in a range 22–24 °C. Simone et al. (2010, 2011a, 2011b) found out that exergy consumption rates increase as the operative temperature increases above 24 °C or decreases below 22 °C. In these two studies, only the effect of air temperature was taken into consideration and not RH_{in}. Results of this study show that at conditions with 80 % RH_{in} human body exergy consumption rate has its minimum at 35 °C. Burn patient and healthcare worker have similar exergy consumption rates at temperature conditions in a range 24–28 °C; even their individual characteristics (T_{cr}, T_{sk}, metabolic rate, effective clothing insulation) are different. The average exergy consumption rate at T_{ai} 24–28 °C is 2.00 W/m^2 for healthcare worker and 2.10 W/m^2 for burn patient. However, if the whole exergy balance is taken into consideration, the rates of input, stored and output exergies between healthcare worker and burn patient differ significantly[4].

[1] Healthcare worker has the minimal human body exergy consumption rate among subjects at conditions T_{ai} 35 °C with 80 % RH_{in}, due to the lowest rate of input exergy by metabolic thermal exergy rate, and the highest rate of stored exergy. The lowest metabolic thermal exergy rate appears due to the smallest difference of T_{sk}–T_{ai} and smallest value of T_{sk}/T_{ai}. Warm radiant exergy rate and warm convective exergy rate discharged from the whole skin and clothing surfaces are the lowest due to the smallest difference of T_{cl}–T_{ai} and the largest value of T_{ai}/T_{cl}. The exergy rate of exhalation and evaporation of sweat is the lowest due to the smallest difference of T_{cl}–T_{ai} and the smallest value of T_{cl}/T_{ai}.

[2] Burn patient has the minimal human body exergy consumption rate at conditions T_{ai} 35 °C with 80 % RH_{in} due to lower input exergy rate by metabolic thermal exergy rate and higher rate of stored exergy than at other temperature conditions and 80 % RH_{in}. Metabolic thermal exergy rate is lower due to smaller difference of T_{cr}–T_{ai} and smaller value of T_{cr}/T_{ai}, smaller difference of T_{sk}–T_{ai} and smaller value of T_{sk}/T_{ai}. Warm radiant exergy rate discharged from the whole skin and clothing surfaces is lower due to lower difference of T_{cl}–T_{ai}. Warm convective exergy rate discharged from the whole skin and clothing surfaces is lower due to lower difference of T_{cl} –T_{ai} and larger value of T_{ai}/T_{cl}. The exergy rate of exhalation and evaporation of sweat is lower due to smaller difference of T_{cl}–T_{ai} and smaller value of T_{cl}/T_{ai}, and smaller difference of T_{cr}/T_{ai} and smaller value of T_{cr}/T_{ai}.

[3] Burn patient has the maximal human body exergy consumption rate at conditions T_{ai} 15 °C with 80 % RH_{in} due to higher rate of input exergy by metabolic thermal exergy rate and lower rate of stored exergy than at other temperature conditions and 80 % RH_{in}. Metabolic thermal exergy rate is higher mainly due to larger difference of T_{cr}–T_{ai} and larger value of T_{cr}/T_{ai}, large difference of T_{sk}–T_{ai} and large value of T_{sk}/T_{ai}. Warm radiant exergy rate discharged from the whole skin and clothing surfaces is higher due to larger difference of T_{cl}–T_{ai}. Warm convective exergy rate discharged from the whole skin and clothing surfaces is higher due to larger difference of T_{cl}–T_{ai} and smaller value of T_{ai}/T_{cl}. The exergy rate of exhalation and evaporation of sweat is higher due to larger difference of T_{cl}–T_{ai} and larger value of T_{cl}/T_{ai}, and larger difference of T_{cr}–T_{ai} and larger value of T_{cr}/T_{ai}. The rate of stored exergy is zero, because exergy is released from the core and shell.

[4] In the case of burn patient exposed to 24–28 °C there appear higher rates of input exergies, higher rates of output exergies and lower rate of stored exergy than in the case of healthcare worker.

The warm exergy rate generated by metabolism presents the main input exergy rate for healthcare worker and also for burn patient[1]. Lower T_{cr} and higher metabolic rate in case of burn patient (35.5 °C, 2 met for burn patient; 36.7 °C, 1.1 met for healthcare worker) result in almost doubled average value of metabolic thermal exergy rate for burn patient (4.03 W/m^2)[2] than for healthcare worker (2.54 W/m^2). Higher T_{cl} temperatures than T_{ai} result in higher average warm radiant exergy rate discharged from the whole skin and clothing surfaces for burn patient (0.47 W/m^2) than for healthcare worker (0.18 W/m^2)[3]. This causes also higher average warm convective exergy rate for burn patient (1.29 W/m^2) than for healthcare worker (0.27 W/m^2)[4]. The exergy rate of exhalation and evaporation of sweat is 0.16 W/m^2 for burn patient and 0.09 W/m^2 for healthcare worker[5]. In case of visitor the average exergy consumption rate at 24–28 °C is much higher, i.e. 4.12 W/m^2, than in case of healthcare worker or burn patient. Different average values of T_{cr} 37.1 °C, T_{sk} 35.6 °C, 2 met and 0.6 clo at conditions T_{ai} 24–28 °C affect input exergies, exergy stored and output exergies[6]: the metabolic thermal exergy rate is 4.97 W/m^2, the exergy rate by exhalation and evaporation of sweat is 0.17 W/m^2, the warm radiant exergy rate outgoing from human body is 0.17 W/m^2, the warm convective exergy rate outgoing from human body is 0.48 W/m^2 for visitor.

[1] Other rates of input exergies are zero for both subjects. Cool/warm radiant exergy rate absorbed by the whole skin and clothing surfaces is zero, because T_{ai} is equal to T_{mr}. Exergy rate of inhaled humid air is also zero, because room T_{ai} and RH_{in} are equal to outside conditions T_{ao} and RH_{out} ($p_{vr} = p_{vo}$).

[2] In the case of burn patient exposed to 24–28 °C the metabolic thermal exergy rate is higher due to higher metabolic rate, larger difference of T_{sk}–T_{ai} and larger value of T_{sk}/T_{ai} than for healthcare worker. In the case of burn patient exposed to 24–28 °C the rate of stored exergy is lower due larger value of T_{ai}/T_{cr} than for healthcare worker.

[3] Warm radiant exergy rate discharged from the whole skin and clothing surfaces is higher due to larger difference of T_{cl}–T_{ai} than for healthcare worker.

[4] In case of burn patient exposed to 24–28 °C the warm convective exergy rate discharged from the whole skin and clothing surfaces is larger due to higher difference of T_{cl}–T_{ai} and smaller value of T_{ai}/T_{cl} than for healthcare worker.

[5] In the case of burn patient exposed to 24–28 °C the exergy rate of exhalation and evaporation of sweat is higher due to larger difference of T_{cr}–T_{ai} and larger value of T_{cr}/T_{ai} than for healthcare worker.

[6] In the case of visitor there appear the highest rate of input exergy and rate of stored exergy among subjects, and higher rates of output exergies than healthcare worker and lower than burn patient. Metabolic thermal exergy rate is the highest among subjects due to high metabolic rate, the largest difference of T_{cr}–T_{ai} and largest value of T_{cr}/T_{ai}. Warm radiant exergy rate discharged from the whole skin and clothing surfaces is the lowest among subjects due to the smallest difference of T_{cl}–T_{ai}. Warm convective exergy rate discharged from the whole skin and clothing surfaces is higher than in the case of healthcare worker and lower than in case of the burn patient due to the difference of T_{cl}–T_{ai} and value of T_{ai}/T_{cl}. The exergy rate of exhalation and evaporation of sweat is the highest among subjects due to the largest difference of T_{cr}–T_{ai} and largest value of T_{cr}/T_{ai}. The rate of stored exergy is the highest among subjects due to the smaller value of T_{ai}/T_{cr} than for other subjects.

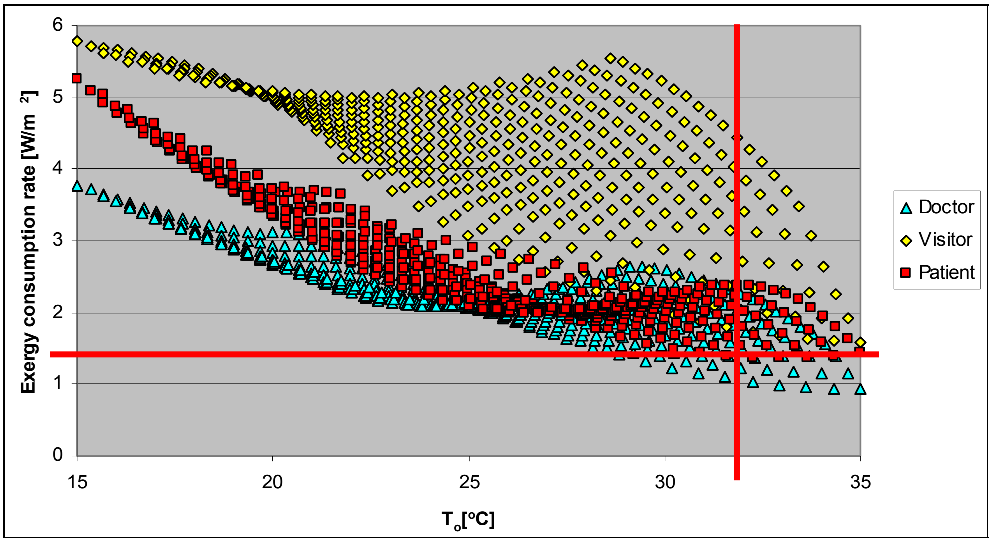

Fig 57: *Human body exergy consumption rate [W/m^2] as a function of T_o, 80 % RH_{in}, LowEx system*

Figure 57 presents the human body exergy consumption rate as a function of T_o for three subjects exposed to 80 % RH_{in} in a room with LowEx system. The maximal human body exergy consumption rate 5.78 W/m^2 could be found in the case of visitor at conditions of T_{ai} 15 °C and T_{mr} 15 °C that result in 15 °C T_o[1]. The minimal human body exergy consumption rate 0.93 W/m^2 could be found in the case of healthcare worker at conditions of T_{ai} 35 °C and T_{mr} 35 °C that result in 35 °C T_o[2].

[1] Visitor has the maximal human body exergy consumption rate among subjects at conditions T_{ai} 15 °C, T_{mr} 15 °C, T_o 15 °C with 80 % RH_{in} due to high rate of input exergy with metabolic thermal exergy rate. Metabolic thermal exergy rate is high mainly due to large difference of T_{cr}–T_{ai} and large value of T_{cr}/T_{ai}, large difference of T_{sk}–T_{ai} and large value of T_{sk}/T_{ai}. Warm radiant exergy rate discharged from the whole skin and clothing surfaces is low due to small difference of T_{cl}–$T_{ai.}$ The exergy rate of exhalation and evaporation of sweat is low due to small difference of T_{cl}–T_{ai} and small value of T_{cl}/T_{ai}. The rate of stored exergy is high due to small value of T_{ai}/T_{cr} and small value of T_{ai}/T_{sk}.

[2] Healthcare worker has the minimal human body exergy consumption rate among subjects at conditions T_{ai} 35 °C, T_{mr} 35 °C, T_o 35 °C with 80 % RH_{in}, due to the lowest rate of input exergy by metabolic thermal exergy rate, and the highest rate of stored exergy. The lowest metabolic thermal exergy rate appears due to the smallest difference of T_{sk}–T_{ai} and smallest value of $T_{sk}/T_{ai.}$ Warm radiant exergy rate and warm convective exergy rate discharged from the whole skin and clothing surfaces are the lowest due to the smallest difference of T_{cl}–T_{ai} and the largest value of T_{ai}/T_{cl}. The exergy rate of exhalation and evaporation of sweat is the lowest due to the smallest difference of T_{cl}–T_{ai} and the smallest value of T_{cl}/T_{ai}.

Burn patient has minimal human body exergy consumption rate at conditions T_o 35 °C (1.44 W/m^2) and maximal at conditions T_o 15 °C (5.26 W/m^2)[1,2]. As can be seen, the same maximal and minimal human body exergy consumption rates could be found for LowEx system as for conventional system. But the results from the Figure 57 and Table 34 show that for the LowEx system the same values of T_o result in different exergy consumption rates. It means that the same T_o could be created with different combination of T_{ai} and T_{mr}.

Table 34: *Comparison between LowEx system and conventional system for set of combinations between T_{ai} and T_{mr}. The exergy values are expressed as rates, W/m^2 (body surface)*

System	Subject	Environmental conditions[3]	PMV index	T_{cr}, T_{sk}, T_{cl} [°C]	Cool (C), Warm (W) radiant exergy in [W/m^2]	Cool (C), Warm (W) convective exergy in [W/m^2]	Breath air in [W/m^2]	From inner part [W/m^2]	Exergy consumption [W/m^2]	Stored exergy[W/m^2]	Exhalation, sweat out[W/m^2]	Cool (C),Warm (W) radiation out [W/m^2]	Cool (C),Warm (W) convection out [W/m^2]
LowEx	Burn patient	T_{ai}=35°C T_{mr}=31°C T_o=32°C RH_{in}=80%	2.6	35.5 37.0 36.2	C= 0.07 W=0	0	0	1.30	1.36	0.001	0.02	C=0 W= 0.01	C=0 W= 0.03
		T_{ai}=30°C T_{mr}=33°C T_o=32°C RH_{in}=80 %	0.8	35.5 37.0 35.7	C=0 W= 0.07	0	0	2.98	2.02	0.007	0.08	C=0 W= 0.26	C=0 W= 0.69
		T_{ai}=31°C T_{mr}=33°C T_o=32°C RH_{in}=80 %	1.1	35.5 37.0 35.8	C=0 W= 0.03	0	0	2.66	1.93	0.007	0.06	C=0 W= 0.19	C= 0 W= 0.50

to be continued…

[1] Burn patient has the minimal human body exergy consumption rate at conditions T_{ai} 35 °C, T_{mr} 35 °C, T_o 35 °C with 80 % RH_{in} due to lower input exergy rate by metabolic thermal exergy rate, and higher rate of stored exergy than at other temperature conditions with 80 % RH_{in}. Metabolic thermal exergy rate is lower mainly due to smaller difference of T_{cr}–T_{ai} and smaller value of T_{cr}/T_{ai}, smaller difference of T_{sk}–T_{ai} and smaller value of T_{sk}/T_{ai}. Warm radiant exergy rate discharged from the whole skin and clothing surfaces is lower due to lower difference of T_{cl}–T_{ai}. Warm convective exergy rate discharged from the whole skin and clothing surfaces is lower due to lower difference of T_{cl}–T_{ai} and larger value of T_{ai}/T_{cl}. The exergy rate of exhalation and evaporation of sweat is lower due to smaller difference of T_{cl}–T_{ai} and smaller value of T_{cl}/T_{ai}, and smaller difference of T_{cr}–T_{ai} and smaller value of T_{cr}/T_{ai}. The rate of stored exergy is higher at conditions T_{ai} 35 °C, T_{mr} 35 °C, T_o 35 °C with 80 % RH_{in} than at conditions T_{ai} 15 °C, T_{mr} 15 °C, T_o 15 °C with 80 % RH_{in}, where thermal exergy is released from the core and the shell.

[2] Burn patient has the maximal human body exergy consumption rate at conditions T_{ai} 15 °C, T_{mr} 15 °C, T_o 15 °C with 80 % RH_{in} due to the higher rate of input exergy by metabolic thermal exergy rate than in other temperature conditions with 80 % RH_{in}. Metabolic thermal exergy rate is the higher mainly due to larger difference of T_{cr}–T_{ai} and larger value of T_{cr}/T_{ai}, larger difference of T_{sk}–T_{ai} and larger value of T_{sk}/T_{ai}. Warm radiant exergy rate discharged from the whole skin and clothing surfaces is higher due to larger difference of T_{cl}–T_{ai}. Warm convective exergy rate discharged from the whole skin and clothing surfaces is higher due to larger difference of T_{cl}–T_{ai} and smaller value of T_{ai}/T_{cl}. The exergy rate of exhalation and evaporation of sweat is higher due to larger difference of T_{cl}–T_{ai} and larger value of T_{cl}/T_{ai}, and larger difference of T_{cr}/T_{ai} and larger value of T_{cr}/T_{ai}. The rate of stored exergy is zero, because exergy is released from the core and shell.

[3] For our calculations T_{ai} was assumed to be equal to T_{ao}. RH_{in} was assumed to be equal to RH_{out}.

…continuation

System	Subject	Environmental conditions[1]	PMV index	T_{cr}, T_{sk}, T_{cl} [°C]	Cool (C), Warm (W) radiant exergy in [W/m^2]	Cool (C), Warm (W) convective exergy in [W/m^2]	Breath air in [W/m^2]	From inner part [W/m^2]	Exergy consumption [W/m^2]	Stored exergy[W/m^2]	Exhalation, sweat out[W/m^2]	Cool (C),Warm radiation out [W/m^2]	Cool (C),Warm convection out [W/m^2]
LowEx	Burn patient	T_{ai}=26°C T_{mr}=35°C T_o=32°C RH_{in}=80 %	0.1	35.5 37.0 35.2	C=0 W= 0.64	0	0	4.36	2.37	0.006	0.16	C=0 W= 0.67	C=0 W= 1.80
		T_{ai}=28°C T_{mr}=34°C T_o=32°C RH_{in}=80 %	0.4	35.5 37.0 35.4	C=0 W= 0.29	0	0	3.67	2.21	0.007	0.12	C=0 W= 0.44	C =0 W= 1.18
Conventional	Burn patient	T_{ai}=32°C T_{mr}=32°C T_o=32°C RH_{in}=80 %	1.3	35.5 37.0 35.9	C=0 W=0	0	0	2.29	1.79	0.006	0.05	C=0 W= 0.12	C=0 W= 0.32

For example, in case of burn patient the required T_o 32 °C could be created with T_{ai} 35 °C and T_{mr} 31 °C or with T_{ai} 31 °C and T_{ai} 33 °C or with T_{ai} 26 °C and T_{mr} 35 °C, etc. But different combinations of T_{ai} and T_{mr} result in different exergy consumption rates. For example, T_{ai} 35 °C and T_{mr} 31 °C result in 1.36 W/m^2 of exergy consumption rate, T_{ai} 31 °C and T_{ai} 33 °C results in 1.93 W/m^2 and T_{ai} 26 °C T_{mr} 35 °C results in 2.37 W/m^2, etc. The other combinations between T_{ai} and T_{mr} that result in required T_o 32 °C and different exergy consumption rates are presented in Table 34. Besides effect on human body exergy consumption rate also separate parts of human body exergy balance could be affected and regulated with this aspect (with setting up optimal relation between T_{ai} and T_{mr}) (Table 34).

Table 35 shows the comparison between LowEx system and conventional system, where the whole human body exergy balance is taken into consideration. Required conditions for the test room were created separately with conventional system (T_{ai} = T_{mr} 32 °C, RH_{in} 80 %) and with LowEx system (T_o 32 °C, RH_{in} 80 %).

[1] For our calculations T_{ai} was assumed to be equal to $T_{ao.}$ RH_{in} was assumed to be equal to RH_{out}.

In case of LowEx system, exergy consumption rate in burn patient is lower, mainly due to much lower rates of input, output and stored exergies. In LowEx system the input exergies present metabolic thermal exergy rate and warm radiant exergy rate absorbed from the whole skin and clothing surfaces[1].

Table 35: *Comparison between LowEx system and conventional system for two selected conditions. The exergy values are expressed as rates, W/m² (body surface)*

System	Subject	Environmental conditions[2]	PMV index	T_{cr}, T_{sk}, T_{cl} [°C]	Cool (C), Warm (W) radiant exergy in [W/m²]	Cool (C), Warm (W) convective exergy in [W/m²]	Breath air in [W/m²]	From inner part [W/m²]	Exergy consumption [W/m²]	Stored exergy [W/m²]	Exhalation, sweat out [W/m²]	Cool (C), Warm (W) radiation out [W/m²]	Cool (C)/ Warm Convection out [W/m²]
LowEx	Burn patient	T_{ai}=35°C T_{mr}=31°C T_o=32°C RH_{in}=80 %	2.6	35.5 37.0 36.2	C=0.13 W=0	0	0	1.30	1.36	0.001	0.02	C=0 W= 0.01	C=0 W= 0.03
Conventional	Burn patient	T_{ai}=32°C T_{mr}=32°C T_o=32°C RH_{in}=80 %	1.3	35.5 37.0 35.9	C=0 W=0	0	0	2.29	1.79	0.006	0.05	C=0 W= 0.12	C=0 W= 0.32

However, in case of LowEx system also lower exergy rate of exhalation and evaporation of sweat, lower rate of warm radiant exergy absorbed from the whole skin and clothing surfaces and lower rate of warm convective exergy absorbed from the whole skin and clothing surfaces transferred from the whole skin and clothing surfaces appear[3-6].

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[1] Metabolic thermal exergy rate is lower mainly due to smaller difference of T_{cr}–T_{ai} and smaller value of T_{cr}/T_{ai}, smaller difference of T_{sk}–T_{ai} and smaller value of T_{sk}/T_{ai}. Warm radiant exergy rate absorbed from the whole skin and clothing surfaces appears due to $T_{mr} > T_{ai}$.

[2] For our calculations T_{ai} was assumed to be equal to T_{ao}. RH_{in} was assumed to be equal to RH_{out}.

[3] The exergy rate of exhalation and evaporation of sweat is lower due to smaller difference of T_{cr}–T_{ai} and smaller value of T_{cr}/T_{ai}, smaller difference of T_{cl}–T_{ai} and smaller value of T_{cl}/T_{ai}.

[4] Warm radiant exergy rate discharged from the whole skin and clothing surfaces is smaller due to lower difference of T_{cl}–T_{ai}.

[5] Warm convective exergy rate discharged from the whole skin and clothing surfaces is lower due to smaller difference of T_{cl}–T_{ai} and larger value of T_{ai}/T_{cl}.

[6] The rate of stored exergy is lower due to larger value of T_{ai}/T_{cr} and larger value of T_{ai}/T_{sk}.

PMV value is lower in case of conventional system, but regarding the mentioned facts we could conclude that for burn patient health conditions have be taken as priority and not comfort ($\dot{L}$ = 0; *PMV* = 0). Regarding Eq. (1), the burn patient exergy consumption rate is minimized only with active regulation of input and output flows, without stimulation of exergy storage with medicines. In case of LowEx system it is up to medical staff to decide what kind of combination of influential factors, such as T_{ai}, T_{mr}, RH_{in} etc., will be included into regulation of separate parts of human body exergy balance and to take into the account other specific factors concerning medical expertise side. In the case of conventional system the regulation of human body exergy balance is limited, T_o could be created only by increasing or decreasing T_{ai}. It is assumed that real conditions in a burn patient room result in much larger difference among input and output exergies between two systems. Poorly insulated building envelope and use of conventional system result at conditions where lower surface temperatures appear than in the case of LowEx system. Lower T_{surf} results in cool radiant exergy absorbed by the whole of skin and clothing surfaces and also cool exergies by radiation and convection discharged from the whole skin and clothing surfaces. These conditions often lead to discomfort. Besides that, conventional system (IR heaters) usually dries the air and high humid conditions are harder to be created (Schata et al., 1990; Lengweiler et al., 1997; Sammaljarvi, 1998).

7.5 Energy use for heating and cooling

Energy use for heating was measured for winter period (5.03.–23.03.2010) and summer period (18.06.–24.06.2010) and it presents overall 528 heating hours (hereinafter called Hh). Heating was performed also for summer period, because of high recommended T_o 32 °C for burn patient room. Energy use for cooling was measured for summer period 10.06.–24.06.2010 and 5.07.–10.07.2010 and presents overall 453 cooling hours (hereinafter called Ch).

The results were also published in our publications (Dovjak et al., 2011a, 2011c, 2011e, 2011f). Figures 58–61 present 24–hour temperature profiles for heating and cooling with conventional and LowEx system.

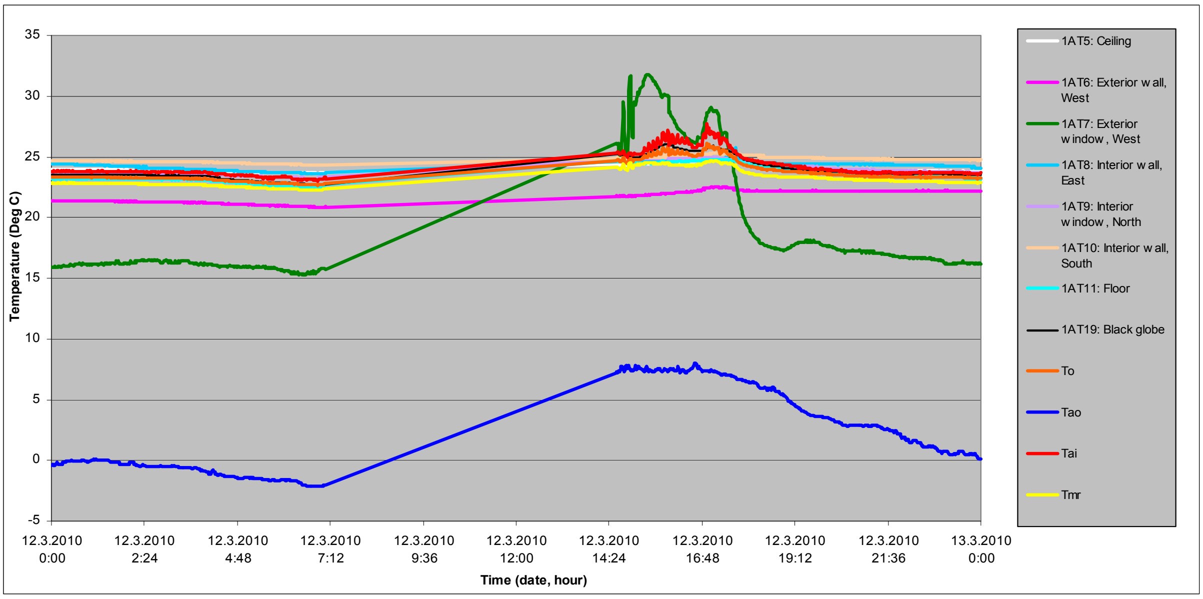

Fig 58: *Heating with conventional system, 12.3.2010 (00:00)–13.3.2010 (0:00), 20.52 MJ (5.70 kWh),* $T_{ai,avg}$ *= 24.2 °C,* $T_{o,avg}$ *= 23.7 °C,* $T_{ao,avg}$ *= 2.2 °C*

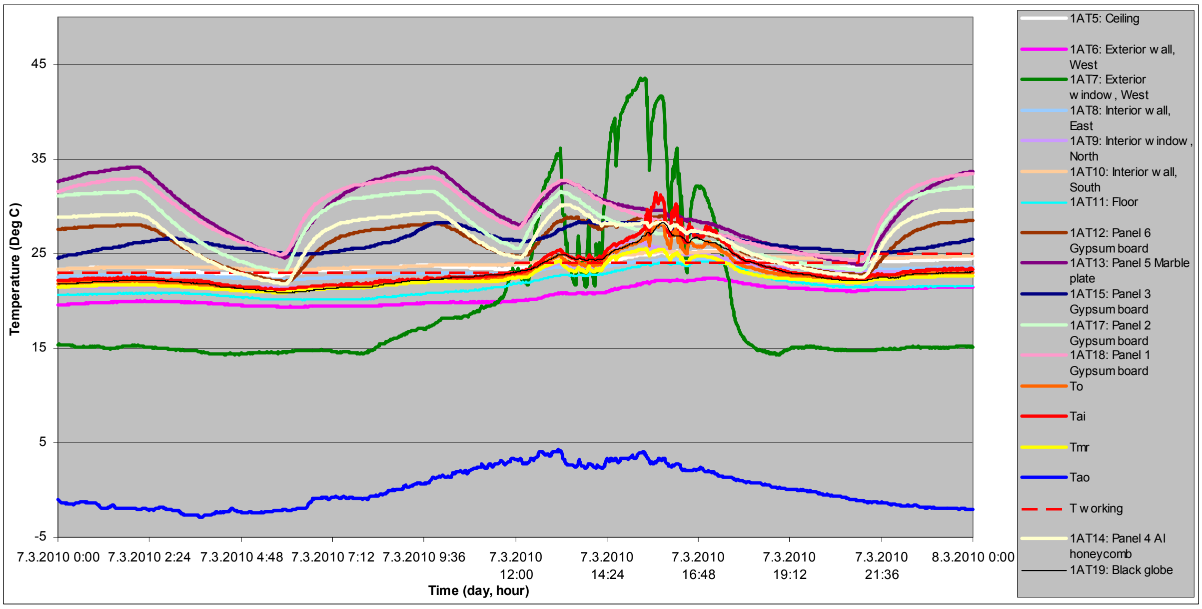

Fig 59: *Heating with LowEx system, 7.3.2010 (00:00)–8.3.2010 (0:00), 46.8 MJ (13.00 kWh),* $T_{ai,avg}$ *= 23.3 °C,* $T_{o,avg}$ *= 22.8 °C,* $T_{ao,avg}$ *= 0.05 °C*

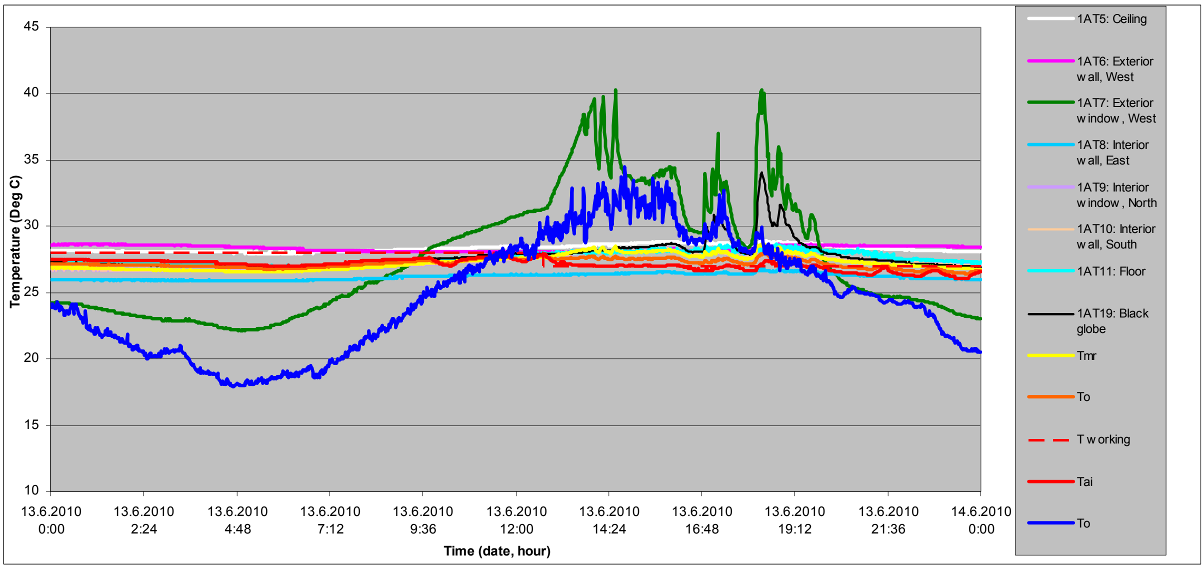

Fig 60: *Cooling with conventional system, 13.6.2010 (00:00)–14.6.2010 (0:00), 36 MJ (10.00 kWh),* $T_{ai,avg}$ *= 27.1 °C,* $T_{o,avg}$ *= 27.2 °C,* $T_{ao,avg}$ *= 24.7 °C*

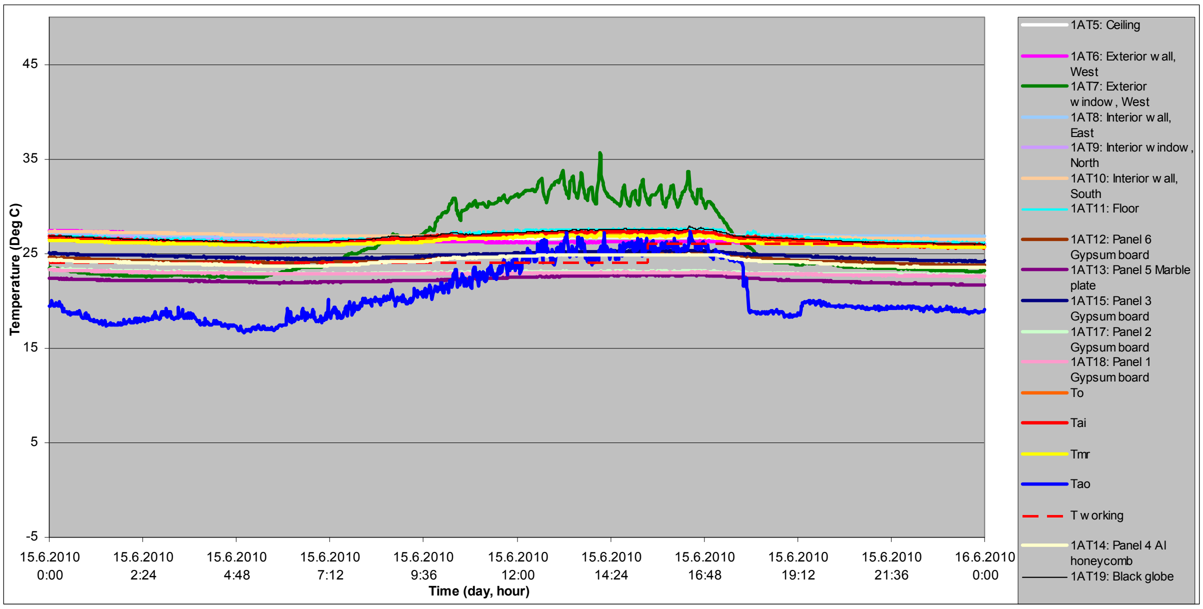

Fig 61: *Cooling with LowEx system, 15.6.2010 (00:00)–16.6.2010 (0:00), 41.76 MJ (11.60 kWh),* $T_{ai,avg}$ *= 26.5 °C,* $T_{o,avg}$ *= 26.4 °C,* $T_{ao,avg}$ *= 20.7 °C*

Since energy use was measured for the same space equipped with LowEx and conventional system in different periods, approximately the same conditions were selected for the systems` comparison (equal set–point T, time period, T_{ao} and T_{ai} vary between systems ±1.0 K; 0.8 % assumed error). The measured energy use is expressed in MJ (and kWh) for heating or cooling.

Table 36: *Results of energy use for selected conditions*

System	$\boldsymbol{T_{range}}$	***LowEx***	***Conventional***	***Reduction [%]***
Cooling summer	*24–25 °C*	T_{ai} = 24.83 °C T_{ao} = 19.45 °C C_{cool} = 1.116 MJ (0.310 kWh)	T_{ai} = 24.30 °C T_{ao} = 19.88 °C C_{cool} = 4.068 MJ (1.130 kWh)	*73*
			T_{ai} = 24.59 °C T_{ao} = 18.89 °C C_{cool} = 3.780 MJ (1.050 kWh)	*71*
			T_{ai} = 24.55 °C T_{ao} = 18.72 °C C_{cool} = 3.798 MJ (1.055 kWh)	*71*
Cooling summer	*25–26 °C*	T_{ai} = 25.49 °C T_{ao} = 18.62 °C C_{cool} = 1.296 MJ (0.360 kWh)	T_{ai} = 24.66 °C T_{ao} = 18.29 °C C_{cool} = 2.074 MJ (0.576 kWh)	*38*
			T_{ai} = 24.73 °C T_{ao} = 18.17 °C C_{cool} = 1.912 MJ (0.531 kWh)	*32*
			T_{ai} = 24.55 °C T_{ao} = 18.72 °C C_{cool} = 3.798 MJ (1.055 kWh)	*66*
			T_{ai} = 24.59 °C T_{ao} = 18.89 °C C_{cool} = 3.780 MJ (1.050 kWh)	*66*
Cooling summer	*25–26 °C*	T_{ai} = 25.50 °C T_{ao} = 21.95 °C C_{cool} =1.160 MJ (0.322 kWh)	T_{ai} = 25.39 °C T_{ao} = 20.08 °C C_{cool} = 2.448 MJ (0.680 kWh)	*53*
			T_{ai} = 25.46 °C T_{ao} = 20.54 °C C_{cool} = 2.646 MJ (0.735 kWh)	*56*
			T_{ai} = 25.43 °C T_{ao} = 21.19 °C C_{cool} = 3.956 MJ (1.099 kWh)	*71*
Cooling summer	*25–26 °C*	T_{ai} = 25.42 °C T_{ao} = 30.04 °C C_{cool} = 1.188 MJ (0.330 kWh)	T_{ai} = 25.39 °C T_{ao} = 30.03 °C C_{cool} =4.320 MJ (1.200 kWh)	*73*
Cooling summer	*25–26 °C*	T_{ai} = 25.32 °C T_{ao} = 22.18 °C C_{cool} = 1.188 MJ (0.330 kWh) T_{ai} = 25.65 °C T_{ao} = 21.59 °C C_{cool} = 1.188MJ (0.330 kWh)	T_{in} = 25.39°C T_{ao} = 20.08 °C C_{cool} = 2.448 MJ (0.680 kWh)	*52*
			T_{ai} = 25.46 °C T_{ao} = 20.54 °C C_{cool} = 2.646 MJ(0.735 kWh)	*55*
			T_{ai} = 25.43 °C T_{ao} = 21.19 °C C_{cool} = 3.956 MJ (1.099 kWh)	*70*
			T_{ai} = 24.82 °C T_{ao} = 20.14 °C C_{cool} = 4.104 MJ (1.140 kWh)	*71*

to be continued...

...continuation

System	T_{range}	***LowEx***	***Conventional***	***Reduction [%]***
Cooling summer	*25–26 °C*	T_{ai} = 25.39 °C T_{ao} = 23.87 °C C_{cool} = 1.260 MJ (0.350 kWh)	T_{ai} = 25.40 °C T_{ao} = 23.38 °C C_{cool} = 4.032 MJ (1.120 kWh)	*69*
Cooling summer	*25–26 °C*	T_{ai} = 25.65 °C T_{ao} = 21.59 °C C_{cool} = 1.188 MJ (0.330 kWh)	T_{ai} = 25.43 °C T_{ao} = 21.19 °C C_{cool} = 3.956 MJ (1.099 kWh)	*70*
Cooling summer	*25–26 °C*	T_{ai} = 25.65 °C T_{ao} = 20.21 °C C_{cool} = 1.440 MJ (0.400 kWh)	T_{ai} = 25.36 °C T_{ao} = 20.30 °C C_{cool} = 2.880 MJ (0.800 kWh)	*50*
		T_{ai} = 25.68 °C T_{ao} = 19.28 °C C_{cool} = 1.199 MJ (0.333 kWh)	T_{ai} = 25.32 °C T_{ao} = 20.0 °C C_{cool} = 3.240 MJ (0.900 kWh)	*63*
Cooling summer	*26–27 °C*	T_{ai} = 26.17 °C T_{ao} = 29.23 °C C_{cool} = 1.188 MJ (0.330 kWh)	T_{ai} = 26.48 °C T_{ao} = 29.86 °C C_{cool} = 3.550 MJ (0.986 kWh)	*67*
Cooling summer	*26–27 °C*	T_{ai} = 26.10 °C T_{ao} = 29.70 °C C_{cool} = 1.368 MJ (0.380 kWh)	T_{ai} = 26.48 °C T_{ao} = 29.86 °C C_{cool} = 3.550 MJ (0.986 kWh)	*62*
Cooling summer	*26–27 °C*	T_{ai} = 26.53 °C T_{ao} = 20.66 °C C_{cool} = 2.002 MJ (0.556 kWh)	T_{ai} = 26.39 °C T_{ao} = 21.60 °C C_{cool} = 3.406 MJ (0.946 kWh)	*41*
		T_{ai} = 26.47°C T_{ao} = 19.78°C C_{cool} = 1.386 MJ (0.385 kWh)	T_{ai} = 26.37 °C T_{ao} = 20.86 °C C_{cool} = 3.409 MJ (0.947 kWh)	*59*
		T_{ai} = 26.55 °C T_{ao} = 20.79 °C C_{cool} = 1.501 MJ (0.417 kWh)	T_{ai} = 26.37 °C T_{ao} = 20.86 °C C_{cool} = 3.409 MJ (0.947 kWh)	*56*

to be continued...

continuation...

System	***T_{range}***	***LowEx***	***Conventional***	***Reduction [%]***
Cooling summer	*27–28 °C*	T_{ai} = 27.35 °C T_{ao} = 24.21 °C C_{cool} = 1.800 MJ (0.500 kWh)	T_{ai} = 26.54 °C T_{ao} = 24.71 °C C_{cool} = 2.567 MJ(0.713 kWh)	*30*
Cooling summer	*27–28 °C*	T_{ai} = 27.57 °C T_{ao} = 27.32 °C C_{cool} = 0.601 MJ (0.167 kWh)	T_{ai} = 27.62 °C T_{ao} = 25.34 °C C_{cool} = 1.238 MJ (0.344 kWh)	*52*
Heating winter	*23–24 °C*	T_{ai} = 23.3 °C T_{ao} =–2.60 °C C_{heat} = 2.63 MJ (0.73 kWh)	T_{ai} = 23.7 °C T_{ao} =–2.60 °C C_{heat} = 2.95 MJ (0.82 kWh)	*11*
Heating summer	*25–26 °C*	T_{ai} = 25.50 °C T_{ao} = 13.50 °C C_{heat} = 0.972 MJ (0.270 kWh)	T_{ai} = 25.40 °C T_{ao} = 13.20 °C C_{heat} = 1.332 MJ (0.370 kWh)	*27*

Table 36 presents the measured energy use for heating and cooling for both systems inside the selected conditions. The measured energy use for heating was by 11–27 % lower for LowEx system than for conventional system. The energy use for cooling was by 30–73 % lower for LowEx system. The overall energy use for the whole measured cooling period (453 Ch) was 518.4 MJ (144 kWh) (1.145 MJ or 0.318 kWh average energy use per Ch) for LowEx system, and 1.314 GJ (365 kWh) (2.902 MJ or 0.806 kWh) for conventional system (61 % reduction).

The reason for relatively low efficiency of LowEx system during the experiment was small ceiling surface area that was covered with panels. The calculated energy use for heating in case of four times larger surface area of panels is 40 % lower energy use than for the conventional system.

Let us consider the temperature measurements between systems for selected conditions where measured energy use for heating was by 11 % and 27 % lower for LowEx system and where for cooling these data were 41 % and 63 %, respectively.

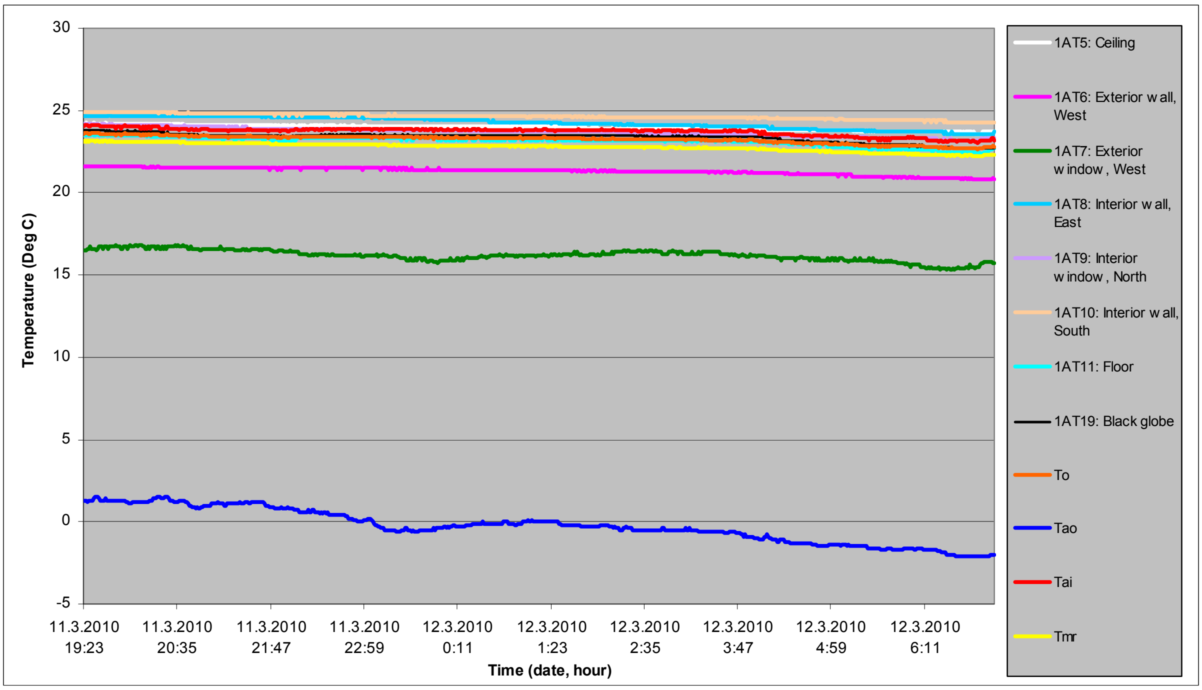

***Fig 62:** Heating with conventional system, 11.3.2010 (19:23)–12.3.2010 (7:05), 2.95 MJ (0.82 kWh),* T_{ai} *= 23–24 °C,* $T_{ai,avg}$ *= 23.7 °C,* $T_{o,avg}$ *= 23.2 °C,* $T_{ao,avg}$ *= –0.3 °C*

Figure 62 presents temperature measurements in the room heated with conventional system for selected conditions (Table 36, turquoise colour) during winter period. For this period, measured energy use for heating was by 11 % higher for conventional system than for LowEx system. However, to compare of energy use between the two systems, ΔT (T_{ai}–T_{ao}) was considered, while in Figure 62 real T_{ao} is presented. For the conditions $T_{ai,avg}$ = 23.7 °C, $T_{mr,\mathrm{avg}}$ = 22.8 °C, $T_{o,avg}$ = 23.2 °C in the room with conventional system, temperature measurements show that interior constructional complexes have the highest surface temperatures, such as interior south wall (T_{avg} = 24.6 °C), interior east wall (T_{avg} = 24.3 °C), ceiling (T_{avg} = 24.0 °C), interior north window (T_{avg} = 23.7 °C) and floor (T_{avg} = 23.1 °C). Lower heat flow rate occurs through interior building complexes due to lower ΔT (1 K) and lower *U–value* ($U_{interior\ wall}$ = 1.17 W/(m^{2}K), $U_{ceiling,\ floor}$ = 0.83 W/(m^{2}K)) than through exterior building complexes (ΔT = 24.0 K, $U_{exterior\ wall}$ = 1.29 W/(m^{2}K)). Exterior building complexes have the lowest surface temperatures, such as exterior west window (T_{avg} = 16.2 °C) and exterior west wall (T_{avg} = 21.3 °C). The reason is higher *U–value* ($U_{exterior\ wall}$ = 1.29 W/(m^{2}K), U_{window} = 2.90 W/(m^{2}K)) that results in larger transmission heat losses and presents a weak part of building system. The same conclusions were made with calculation of surface temperature and heat flow rate for Case 1, U_{cu} in Ljubljana that presents the exterior wall of our test room (sections 7.2.3, 7.2.4). The calculated surface temperature for Case 1, U_{cu} was the lowest (17.6 °C) among cases, and heat flow rate the highest (18.71 W/m^2). The changes of surface temperatures for exterior window and exterior wall were similar as changes of T_{ao}. Moreover, from 3:45 am to 7:05 am a large decrease of T_{ao} could be detected (from –0.7 °C to –2 °C) and that results in decrease of surface temperatures of exterior west wall (from 21.2 °C to 20.8 °C), exterior west window (from 16.2 °C to 15.7 °C) and room conditions T_{ai} (from 23.7 °C to 23.2 °C), T_{mr} (from 22.7 °C to 22.4 °C), $T_{black\ globe}$ (from 23.4 °C to 22.7 °C) and T_o (from 23.2 °C to 22.8 °C).

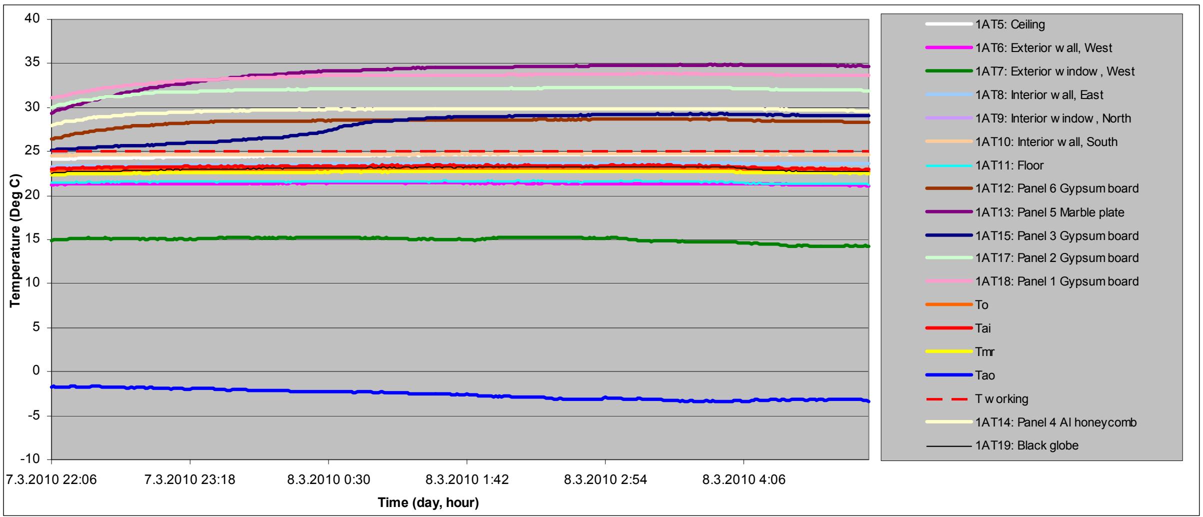

Fig 63: *Heating with LowEx system, 11 % lower energy use, 7.3.2010 (22:06)–8.3.2010 (5:11), 2.63 MJ (0.73 kWh),* T_{ai} *= 23–24 °C,* $T_{ai,avg}$ *= 23.3 °C,* $T_{o,avg}$ *= 22.9 °C,* $T_{ao,avg}$ *= –2.6 °C*

Figure 63 presents temperature measurements in the room heated with LowEx system for selected conditions (Table 36, turquoise colour), where measured energy use for heating was by 11 % higher than in the room with conventional system. For creating room conditions ($T_{ai,avg}$ = 23.3 °C, $T_{mr,avg}$ = 22.7 °C, $T_{o,avg}$ = 22.9 °C) panels were set up to 25 °C. Panel–3 gypsum board had the lowest surface temperature (T_{avg} = 28.0 °C) and panel 5–marble plate had the highest surface temperature (T_{avg} = 33.9 °C). The reason is that marble has higher thermal conductivity (2.330 W/(mK)) and specific heat (880 J/(kgK)) that gypsum plate (0.210 W/(mK), 840 J/(kgK)). In the room with LowEx system, similar conclusion about surface temperatures of building envelope could be made as in a room with conventional system. The highest surface temperatures appear for interior south wall (T_{avg} = 24.7 °C), ceiling (T_{avg} = 24.6 °C), interior east wall (T_{avg} = 23.6 °C), interior east window (T_{avg} = 23.0 °C) and floor (T_{avg} = 21.6 °C). Average floor temperature in a room with LowEx system is by 1.5°C lower than in a room with conventional system, mainly because of the position of radiators and radiative and convective heat flux on floor surfaces. Lower T_{ao} causes lower surface temperatures on exterior west window (T_{avg} = 15.0 °C) than in a room with conventional system (T_{avg} = 16.2 °C). Other surface temperatures and room conditions ($T_{exterior\ west\ wall,avg}$ = 21.4 °C, $T_{o,avg}$ = 22.9 °C, $T_{mr,avg}$ = 22.7 °C, $T_{black\ globe}$ = 23.0 °C) are almost the same as in the room with conventional system. Because in a room with LowEx system additional surface area is heated to higher temperatures (28.0–33.9 °C), much higher T_{mr} could be expected. The reason is the surface area of panels that affects T_{mr} (9 m^2). T_{mr} is affected by all room surfaces, not just panels. However, for both cases, almost the same room conditions are created, but in the case of LowEx system energy use is by 11 % lower. More precise look at T_{ai}, T_{mr}, T_o, $T_{black\ globe}$ (Figures 64 and 65) shows that temperatures are much more constant in a room with LowEx system than in a room with conventional system. Temperature measurements with black globe thermometer are close to temperature T_o, which was calculated (Annex A).

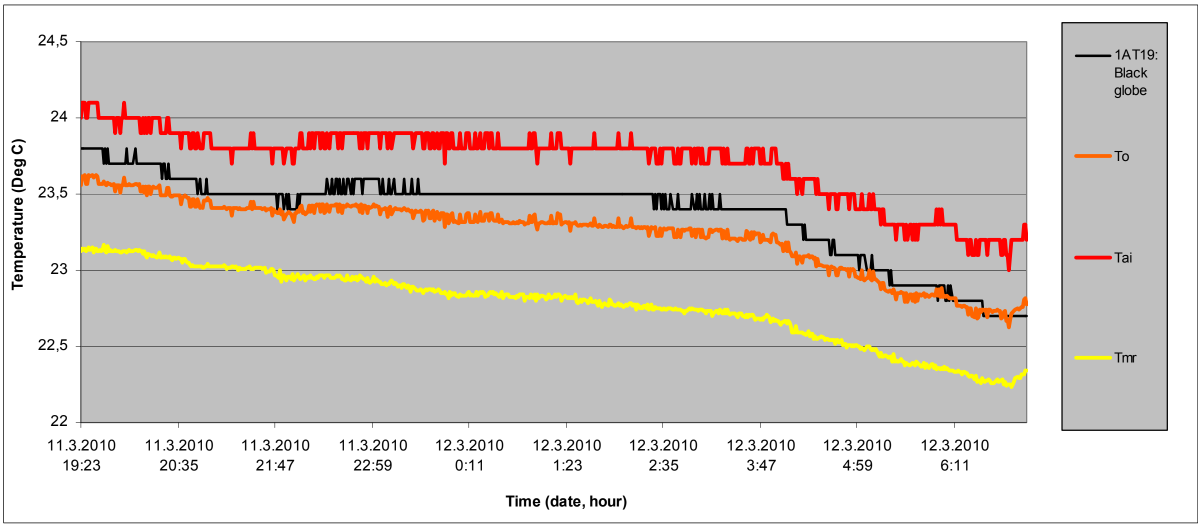

Fig 64: *Temperature conditions in the room heated with conventional system, 11.3.2010 (19:23)–12.3.2010 (7:05), 2.95 MJ (0.82 kWh),* T_{ai} *= 23–24 °C,* $T_{ai,avg}$ *= 23.7 °C,* $T_{o,\ avg}$ *= 23.2 °C,* $T_{black\ globe,avg}$ *= 23.4 °C,* $T_{ao,avg}$ *= –0.3 °C*

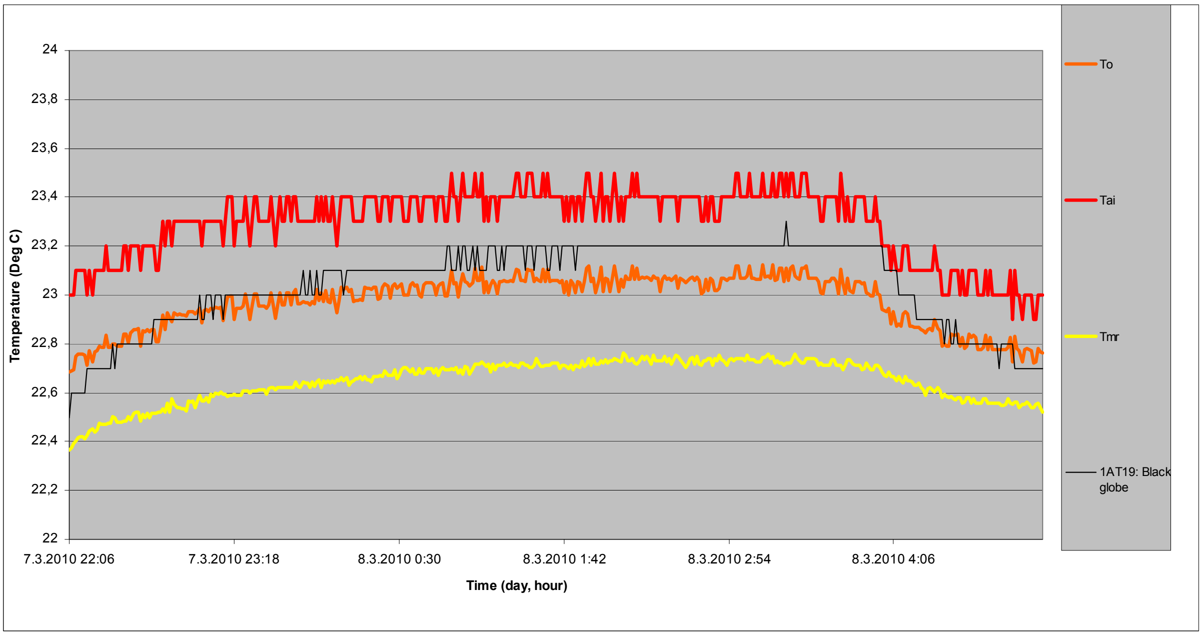

Fig 65: *Temperature conditions in the room heated with LowEx system, 11 % lower energy use, 7.3.2010 (22:06)–8.3.2010 (5:11), 2.63 MJ (0.73 kWh),* $T_{ai,avg}$ *= 23.3 °C,* $T_{o,avg}$ *= 22.9 °C,* $T_{black\ globe,\ avg}$ *= 23.0 °C,* $T_{ao,avg}$ *= –2.6 °C*

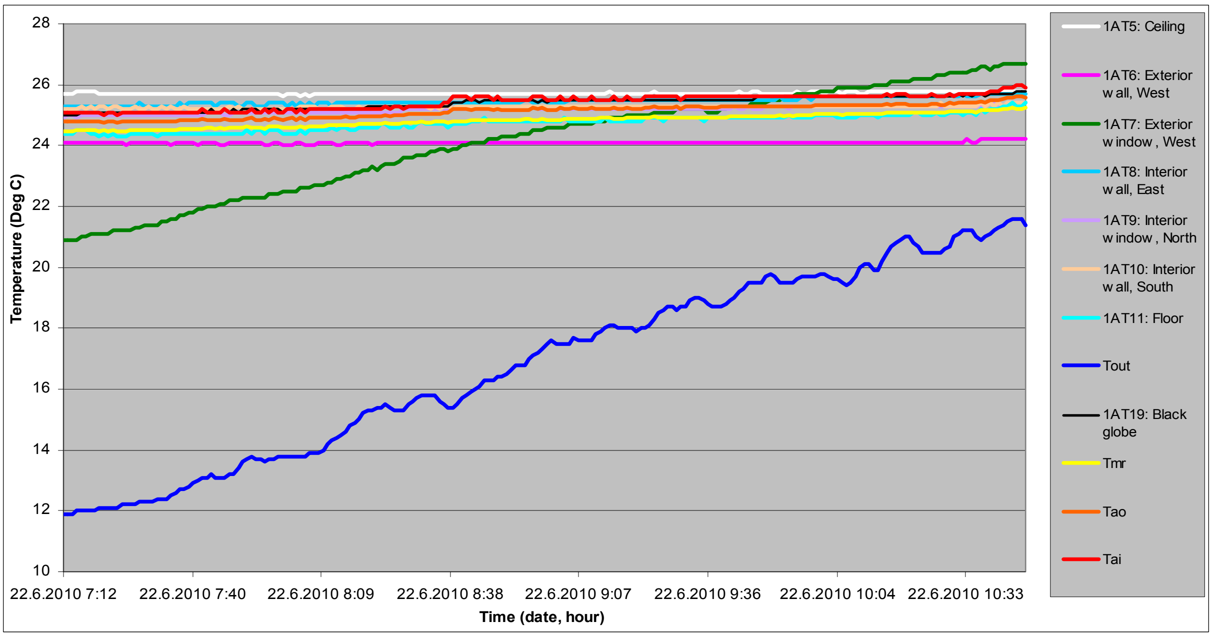

Fig 66: *Heating with conventional system, 22.6.2010 (7:12)–22.6.2010 (10:47), 1.33 MJ (0.37 kWh),* T_{ai} *= 25–26 °C,* $T_{ai,avg}$ *= 25.4 °C,* $T_{o,avg}$ *= 25.1 °C,* $T_{ao,avg}$ *= 16.8 °C*

Figure 66 presents temperature measurements in a room with conventional system for selected conditions (Table 36, turquoise colour) during summer period where energy use for heating was by 27 % higher for conventional system than for LowEx system. Heating during summer was performed because the recommended T_o 32 °C has to be created. If comparing the same heating system with the one during winter period (with 11 % less efficiency, Figure 62), it can be concluded that higher T_{ai}, T_{mr}, T_o and $T_{black\ globe}$ ($T_{ai,avg}$ = 25.4 °C, $T_{mr,avg}$ = 24.8 °C, $T_{o,avg}$ = 25.1 °C, $T_{black\ globe}$ = 25.4 °C) and much higher T_{ao} in summer ($T_{ao,avg}$ = 16.8 °C) result in higher surface temperatures of interior constructional complexes than in the case of conventional system during winter period. For example, 1.7 °C in case of ceiling and floor, 1.5 °C for interior north window, 0.6 °C for interior south wall and 1.2 °C for interior east wall. Due to higher T_{ao} (16.8 °C), higher surface temperatures appear also on non–transparent exterior complexes appears (by 2.8 °C for exterior west wall). Sun radiation causes particularly high surface temperatures on exterior west window (T_{avg} = 24.1 °C).

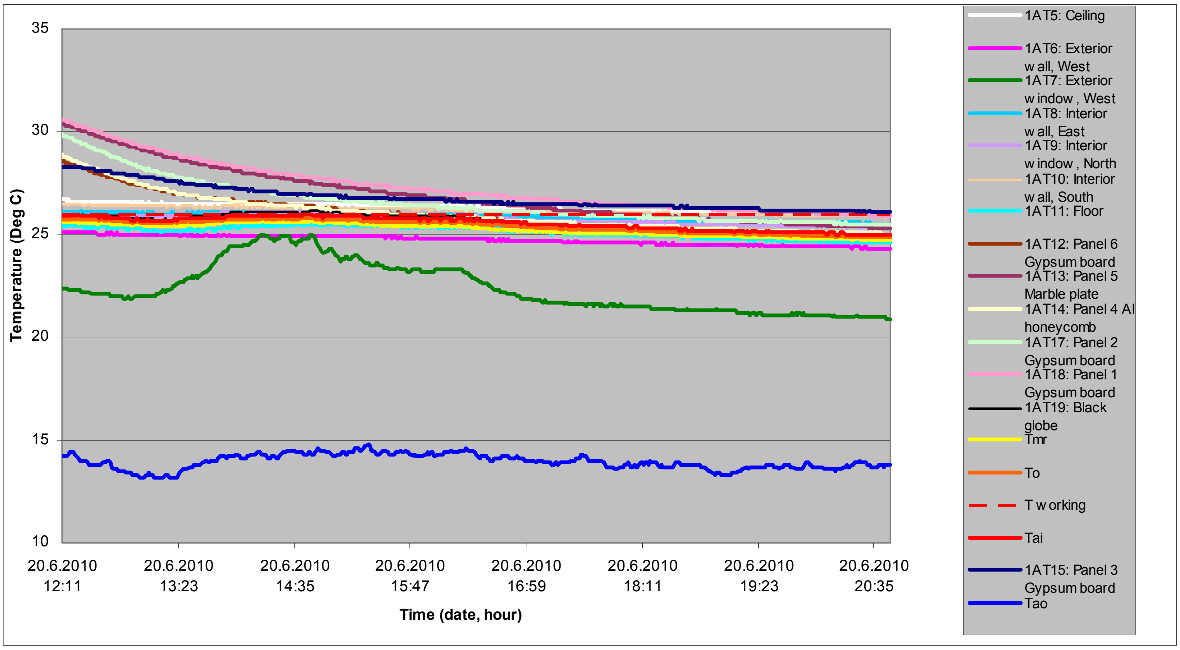

Fig 67: *Heating with LowEx system, 27 % lower energy use, 20.6.2010 (12:11)–20.6.2010 (20:45), 0.97 MJ (0.27 kWh),* T_{ai} *= 25–26 °C,* $T_{ai,avg}$ *= 25.5 °C,* $T_{o,avg}$ *= 25.4 °C,* $T_{ao,avg}$ *= 13.5 °C*

In case of heating with LowEx system, panels were set up to 26 °C and room conditions with $T_{ai,avg}$ 25.5 °C, $T_{mr,avg}$ 25.2 °C, $T_{o,avg}$ 25.4 °C were created (Figure 67). Measured energy use for heating was by 27 % lower than in the room with conventional system. Panel 1–marble plate had the highest surface temperature (T_{avg} = 27.0 °C) and panel 6–gypsum board had the lowest (T_{avg} = 26.0 °C). In the room with LowEx system, similar conclusion about surface temperatures of building envelope can be made as in the room with conventional system. Also it shows 27 %

less efficiency. The highest surface temperatures appear for ceiling (T_{avg} = 26.3 °C), interior south wall (T_{avg} = 26.2 °C), interior east wall (T_{avg} = 25.9 °C), interior north window (T_{avg} = 25.8 °C) and floor (T_{avg} = 25.1 °C). Surface temperatures and room conditions ($T_{exterior\ west\ wall,avg}$ = 24.7 °C, $T_{o,avg}$ = 25.4 °C, $T_{mr,avg}$ = 25.2 °C, $T_{black\ globe}$ = 25.6 °C) are almost the same as in the room with conventional system. Because of different T_{ao} (13.5 °C), surface temperature of exterior west window differs by 1.7 °C. If we compare the same system for different period, the main conclusion is that during summer period with higher outdoor temperatures (T_{avg} = 13.5 °C) LowEx system is more effective. 1.66 MJ (0.46 kWh) less energy is used for heating, and higher temperatures are created ($T_{ai,avg}$ = 25.5 °C, $T_{mr,\mathrm{avg}}$ = 25.2 °C, $T_{o,avg}$ = 25.4 °C) than in winter case with 2.63 MJ (0.73 kWh), $T_{ao,\mathrm{avg}}$ = –2.6 °C and created conditions $T_{ai,avg}$ = 23.3 °C, $T_{mr,\mathrm{avg}}$ = 22.7 °C, $T_{o,avg}$ = 22.9 °C.

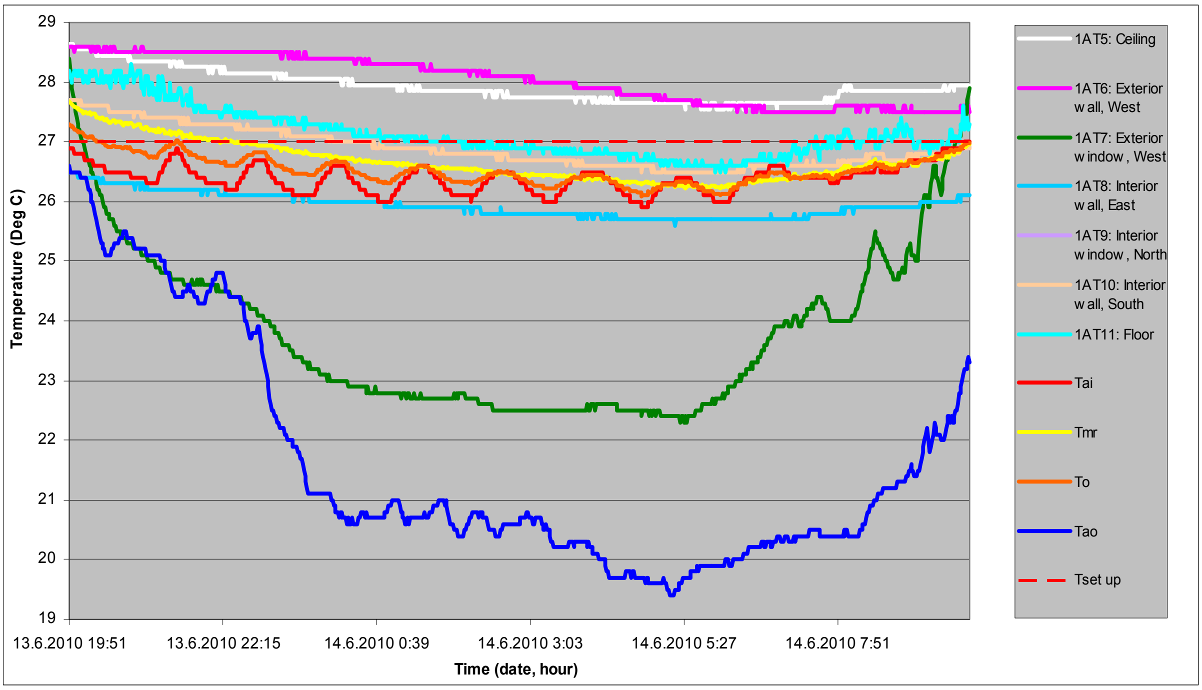

Fig 68: *Cooling with conventional system, 13.6.2010 (19:51)–14.6.2010 (9:55),* T_{ai} *= 26–27 °C, 3.406 MJ (0.946 kWh),* $T_{ai,avg}$ *= 26.4 °C,* $T_{o,avg}$ *= 26.5 °C,* $T_{ao,avg}$ *= 21.6 °C*

Figure 68 presents temperature measurements in a room cooled with conventional system for selected conditions for cooling (Table 36, orange colour). During this period, the measured energy use for cooling was by 41 % higher for conventional system than for LowEx system. For room conditions ($T_{ai,avg}$ = 26.4 °C, $T_{mr,avg}$ = 26.7 °C, $T_{o,avg}$ = 26.5 °C) in the room with conventional system exterior west wall (T_{avg} =

28.0 °C) has the highest surface temperatures and exterior window has the lowest (T_{avg} = 23.8 °C), because they are highly affected by outdoor temperature variations. The surface temperatures are similar as T_{ao}. The curve of T_{ai} and T_o changes approximately for 0.5 °C/0.5 h.

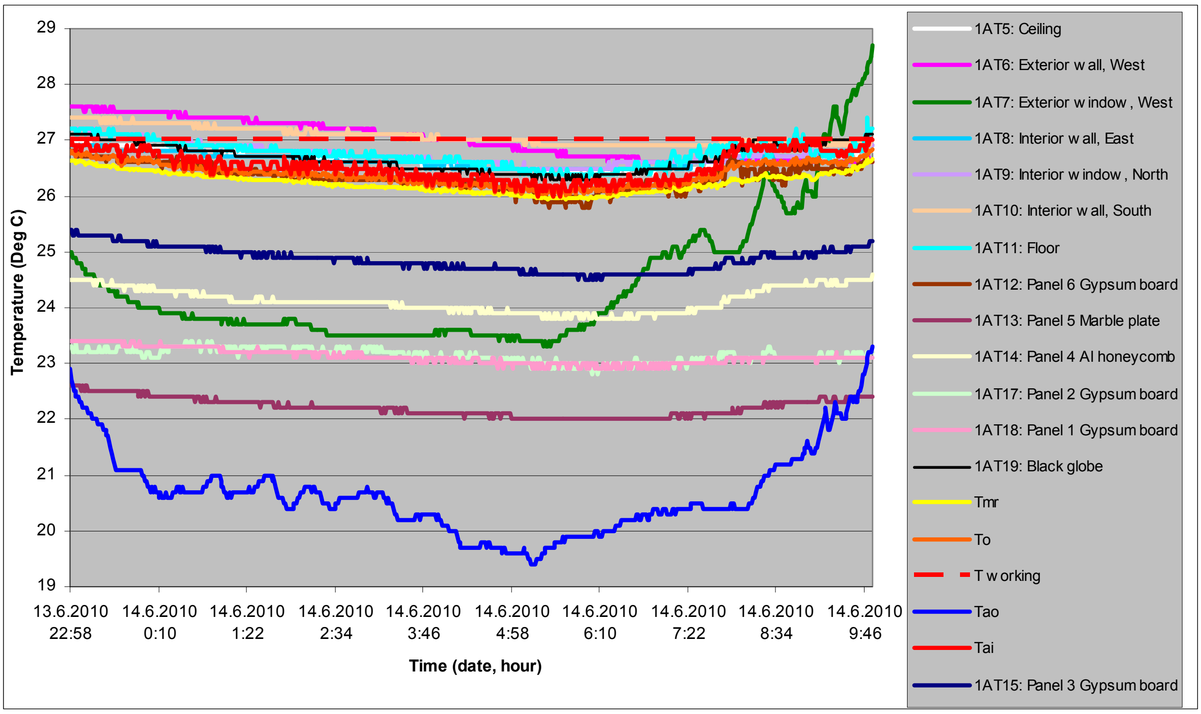

Fig 69: *Cooling with LowEx system, 41 % lower energy use, 13.6.2010 (22:58)–14.6.2010 (9:53),* T_{ai} *= 26–27 °C, 2.002 MJ (0.556 kWh),* $T_{ai,avg}$ *= 26.5 °C,* $T_{o,avg}$ *= 26.4 °C,* $T_{ao,avg}$ *= 20.7 °C*

The comparison with LowEx system shows that interior south wall has the highest surface temperatures (T_{avg} = 27.1 °C) and exterior west window the lowest (T_{avg} = 24.4 °C) (Figure 69). For room conditions ($T_{ai,avg}$ = 26.5 °C, $T_{mr,avg}$ = 26.2 °C, $T_{o,avg}$ = 26.4 °C) panels were set up to 27 °C. Panel 5–marble plate had the lowest surface temperature (T_{avg} = 22.2 °C) and panel–3 gypsum board the highest (T_{avg} = 24.9 °C). Figures 70 and 71 show that T_o, T_{mr}, $T_{black\ globe}$ are more constant in case of LowEx system than conventional.

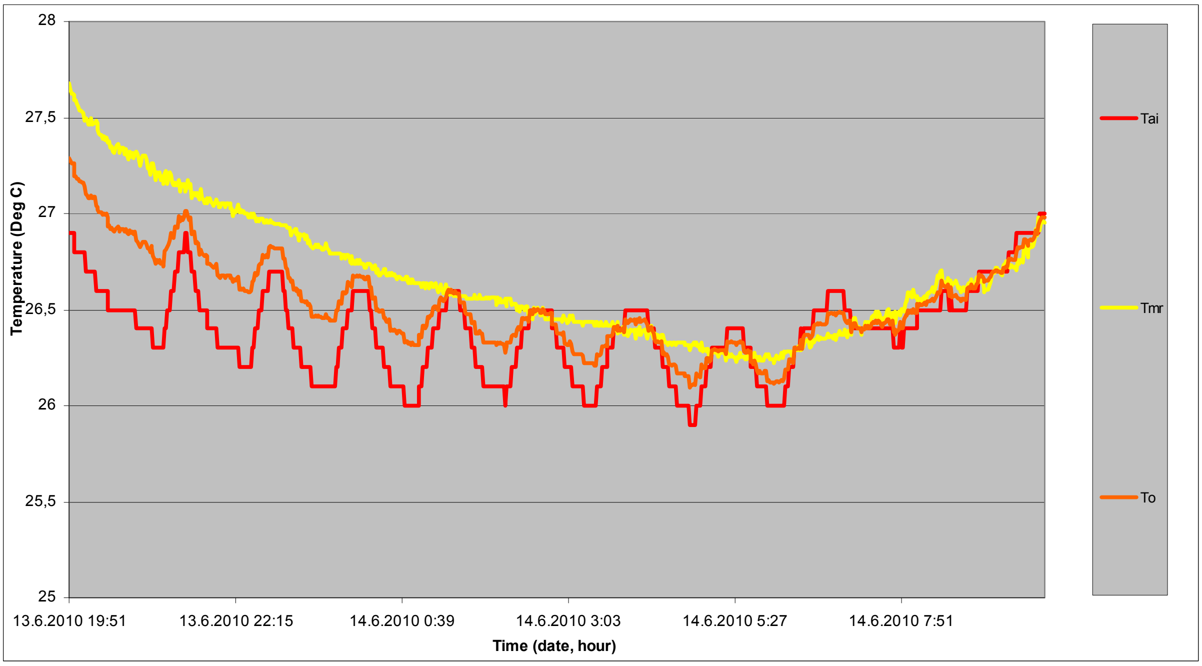

Fig 70: *Temperature conditions in the room cooling with conventional system, 13.6.2010 (19:51)–14.6.2010 (9:55),* T_{ai} *= 26–27 °C, 3.406 MJ (0.946 kWh),* $T_{ai,avg}$ *= 26.4 °C,* $T_{o,avg}$ *= 26.5 °C,* $T_{ao,avg}$ *= 21.6 °C*

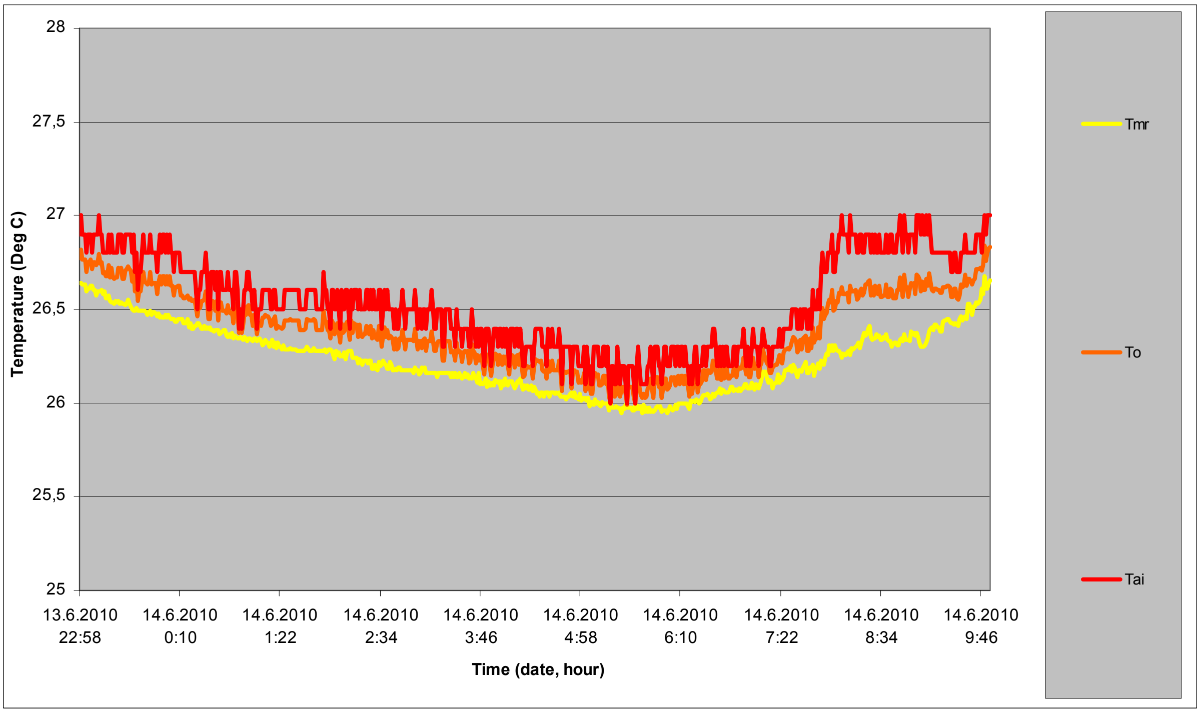

Fig 71: *Temperature conditions in the room cooling with LowEx system, 41 % lower energy use, 13.6.2010 (22:58)–14.6.2010 (9:53),* T_{ai} *= 26–27 °C, 2.002 MJ (0.556 kWh),* $T_{ai,avg}$ *= 26.5 °C,* $T_{o,avg}$ *= 26.4 °C,* $T_{ao,avg}$ *= 20.7 °C*

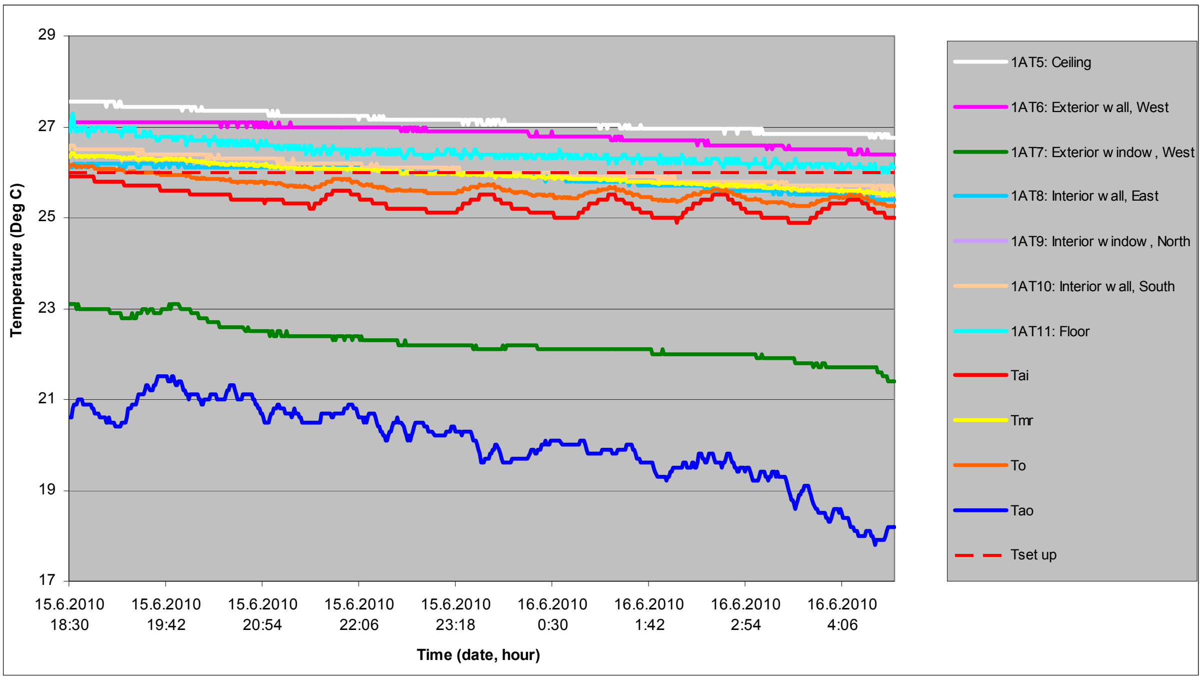

Fig 72: *Cooling with conventional system, 15.6.2010 (18:30)–16.6.2010 (4:45), 3.240 MJ (0.900 kWh),* T_{ai} *= 25–26 °C,* $T_{ai,avg}$ *= 25.3 °C,* $T_{o,avg}$ *= 25.6 °C,* $T_{ao,avg}$ *= 20.0 °C*

Figure 72 presents temperature measurements in the room with conventional system for selected conditions (Table 36, green colour) for cooling period. At that time period, measured energy use for cooling was by 63 % higher for conventional system than for LowEx system. Lower T_{ai}, T_{mr} and T_o ($T_{ai,avg}$ = 25.3 °C, $T_{mr,avg}$ = 26.0 °C, $T_{o,avg}$ = 25.6 °C) and higher T_{ao} ($T_{ao,avg}$ = 20.0 °C) than in previous case with conventional system and 41 % less efficiency result in lower surface temperatures of interior constructional complexes, i.e. 0.8 °C in case of ceiling, interior north window, interior south wall and 0.1 °C for interior east wall. Due to lower T_{ao} ($T_{ao,\text{avg}}$ = 20.0 °C), higher surface temperatures appear also on non–transparent exterior complexes (1.2 °C for exterior west wall, 1.5 °C for exterior west window).

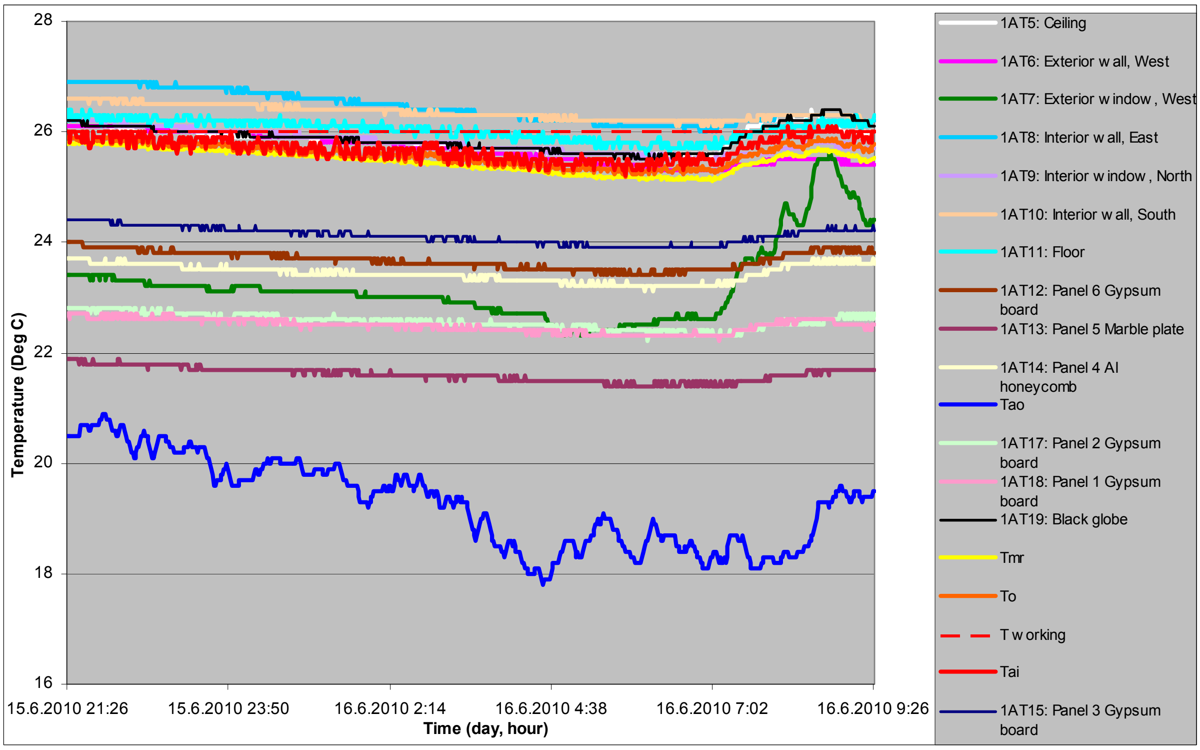

Fig 73: *Cooling with LowEx system, 63 % lower energy use, 15.6.2010 (21:26)–16.6.2010 (9:27),* T_{ai} *= 26–27 °C, 1.199 MJ (0.333 kWh),* $T_{ai,avg}$ *= 25.7 °C,* $T_{o,avg}$ *= 25.6 °C,* $T_{ao,avg}$ *= 19.3 °C*

In the room cooled with LowEx system and 63 % lower energy use, panels were set up to 26 °C and room conditions with $T_{ai,avg}$ = 25.7 °C, $T_{mr,avg}$ = 25.5 °C, $T_{o,avg}$ = 25.6 °C were created. Figure 73 presents temperature measurements in the room with LowEx system for selected conditions (Table 36, green colour), where measured energy use for cooling was by 63 % lower than in the room with conventional system. Panel 5–marble plate had the lowest surface temperature (T_{avg} = 21.6 °C) and panel 3–gypsum board the highest (T_{avg} = 24.1 °C). It is interesting that at almost the same indoor and outdoor conditions, surface temperatures differ, for example: ceiling by 1.3 °C, exterior west wall by 1.2 °C, exterior west window by –0.9 °C, interior east wall by –0.5 °C, interior north window by 0.3 °C, interior south wall by –0.3 °C and floor by 0.4 °C. If comparing the same system for different period, the main conclusion is that lower outdoor temperatures (T_{avg} = 19.3 °C) cause higher efficiency of LowEx system. 0.803 MJ (0.223 kWh) less energy is used for cooling, and lower temperatures are created ($T_{ai,avg}$ = 25.7 °C, $T_{mr,avg}$ = 25.5 °C, $T_{o,avg}$ = 25.6 °C) than in previous case with $T_{ao,avg}$ = 20.7 °C and created conditions $T_{ai,avg}$ = 26.5 °C, $T_{mr,avg}$ = 26.2 °C, $T_{o,avg}$ = 26.4 °C.

7.6 Active regulation

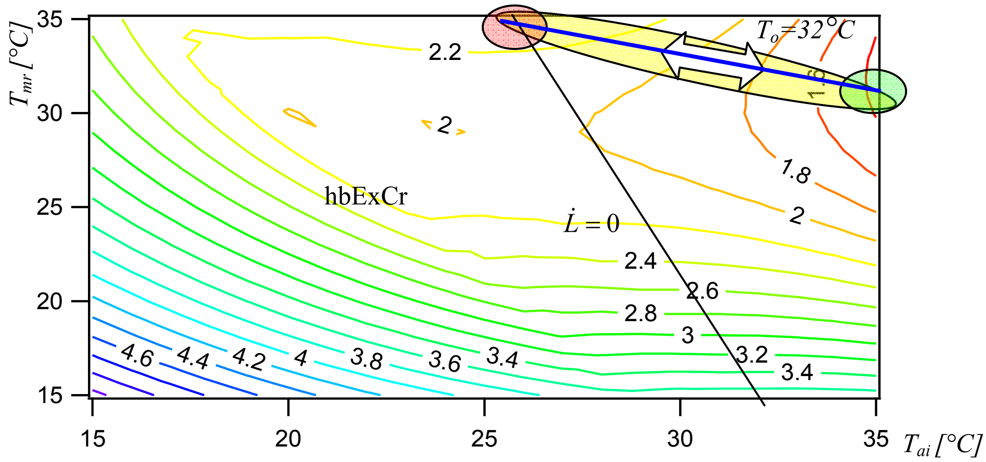

Fig 74: *Human body exergy consumption rate (hbExCr) [W/m²], burn patient, as a function of T_{ai} and T_{mr}, 80 % RH_{in}, LowEx system. The black line presents the rate of thermal load on the body surface area ($\dot{L}$ = 0, PMV = 0)*

LowEx system enables the regulation of human body exergy consumption rate, health and comfort conditions at the same time. What has to be taken as priority depends considerably on regulated demands and individual needs (effective clothing insulation, metabolic rate, health status, etc.). For example, in hospital environment, patient's health has to be taken as priority. In hospital room for burn patient the recommended value T_o 32 °C (blue line) has to be fulfilled with combination of T_{mr} in a range 31–35 °C, and T_{ai} in a range 26–35 °C (yellow ellipse). Within the required conditions, only a small zone (red cycle) presents conditions where internal heat production is equal to the sum of heat losses from the body (i.e. comfort conditions). To create conditions where internal heat production is equal to the sum of heat losses from the body, attained minimal possible human body exergy consumption rate and fulfillment of required conditions, LowEx system has to be set up to T_{ai} 26 °C and T_{mr} 35 °C (marked with red circle on Figure 74). If T_o is 32 °C, internal heat production is equal to the sum of heat losses from the body and minimal possible human body exergy consumption rate (2.37 W/m²) is attained. LowEx system gives us new possibilities to control and regulate every part of the human body exergy balance (e.g. sweat regulation, increase or decrease of human body exergy consumption rate, regulation of warm radiation, etc.). For example, if we want to increase exhalation and evaporation of sweat losses, LowEx system enables us to set the temperatures and create conditions where human body will release the exergy mainly with evaporation losses (e.g. for detoxification therapy)[1]. In conditions T_{ai} 15 °C and T_{mr} 35 °C with 80 % RH_{in} the exergy rate of exhalation of sweat is 0.51 W/m^2 and at conditions 35 °C T_{ai} and 15 °C T_{mr} with 80 % RH_{in} the exergy rate of exhalation of sweat is much lower, i.e. 0.02 W/m^2. This aspect is very important and gives us totally new possibilities in curative or preventive medicine. If the health status of the burn patient requires different conditions inside of active space or even higher or lower human body exergy consumption rate, LowEx system enables this[2]. For example, human body exergy consumption rate in burn patient could be lowered to minimal possible value, with setting up T_{mr} to 31 °C and T_{ai} to 35 °C (1.36 W/m^2). In such way, active zone for burn patient is designed and regulated. Within the designed active zone (green cycle in Figure 74) optimal conditions (recommended T_o 32 °C and healthy conditions) were created with minimal human body exergy consumption rate (1.36 W/m^2) and minimal exergy rate of exhalation and evaporation of sweat (0.02 W/m^2), warm radiant exergy rate (0.01 W/m^2) and warm convective exergy rate transferred from the whole skin and clothing surfaces (0.03 W/m^2). However, in previous conditions T_{ai} 26 °C and T_{mr} 35 °C with exergy consumption rate 2.37 W/m^2 (marked with red circle), loss by exergy rate of exhalation and evaporation of sweat is 0.16 W/m^2, warm radiant exergy rate 0.67 W/m^2 and warm convective exergy rate transferred from the whole skin and clothing surfaces 1.80 W/m^2 (required conditions T_o 32 °C and comfort conditions $\dot{L} = 0$)[3].

[1] The exergy rate of exhalation and evaporation of sweat is higher at conditions T_{ai} 15 °C and T_{mr} 35 °C with 80 % RH_{in} than at conditions T_{ai} 35 °C and T_{mr} 15 °C with 80 % RH_{in} due to larger difference of T_{cr}–T_{ai} and larger value of T_{cr}/T_{ai}, larger difference of T_{cl}–T_{ai} and larger value of T_{cl}/T_{ai}. At conditions T_{ai} 15 °C and T_{mr} 35 °C with 80 % RH_{in} (T_{cr} is 35.5 °C, T_{cl} is 31.7 °C, T_{sk} is 37 °C), the exergy rate of exhalation of sweat is 0.51 W/m^2 and the difference of T_{cr}–T_{ai} is 20.5 and value of T_{cr}/T_{ai} is 2.4, the difference of T_{sk}–T_{ai} is 22.0 and value of T_{sk}/T_{ai} is 2.5. At conditions T_{ai} 35 °C and T_{mr} 15 °C with 80 % RH_{in} (T_{cr} is 35.5 °C, T_{cl} is 34.6 °C, T_{sk} 37 °C), the exergy rate of exhalation of sweat is 0.02 W/m^2 and the difference of T_{cr}–T_{ai} is 0.5 and value of T_{cr}/T_{ai} is 1.0, the difference of T_{sk}–T_{ai} is 2.0 and value of T_{sk}/T_{ai} is 1.1.

[2] Human body exergy consumption rate is regulated with influential factors–variables that affect the rates of input exergies, output exergies and stored exergy.

[3] Human body exergy consumption rate for burn patient at conditions T_{mr} 31 °C and T_{ai} 35 °C is much lower than for example at conditions with T_{mr} 35 °C and T_{ai} 26 °C and due to lower rates of input exergies. Metabolic thermal exergy rate is lower due to smaller difference of T_{cr}–T_{ai} and smaller value of T_{cr}/T_{ai}, smaller difference of T_{sk}–T_{ai} and smaller value of T_{sk}/T_{ai}. Cool radiant exergy rate absorbed from the whole skin and clothing surfaces appears, because $T_{mr} < T_{ai}$. Warm radiant exergy rate discharged from the whole skin and clothing surfaces is smaller due to lower difference of T_{cl}–T_{ai}. Warm convective exergy rate discharged from the whole skin and clothing surfaces is lower due to smaller difference of T_{cl}–T_{ai} and larger value of T_{ai}/T_{cl}. The exergy rate of exhalation and evaporation of sweat is lower due to smaller difference of T_{cr}–T_{ai} and smaller value of T_{cr}/T_{ai}, smaller difference of T_{cl}–T_{ai} and smaller value of T_{cl}/T_{ai}. Stored exergy is lower due to larger value of T_{ai}/T_{cr} and larger value of T_{ai}/T_{sk}.

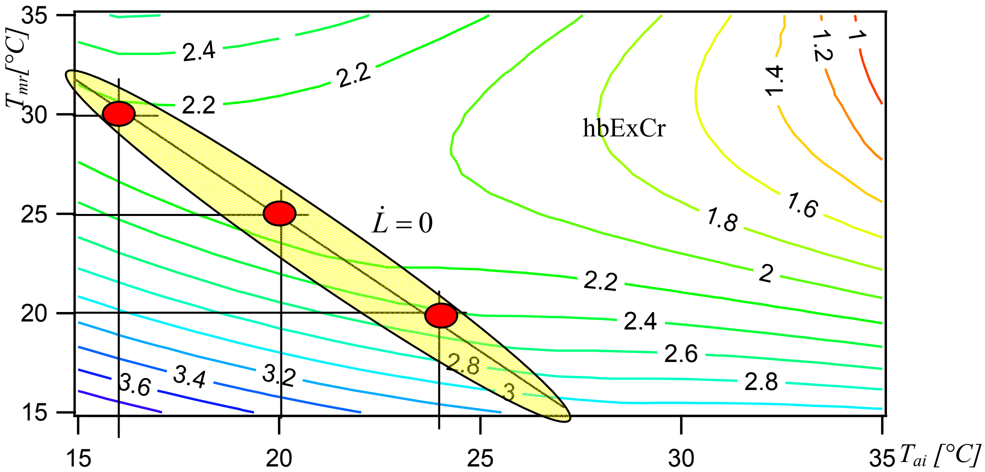

Fig 75: *Human body exergy consumption rate (hbExCr) [W/m²], healthcare worker, as a function of T_{ai} and T_{mr}, 80 % RH_{in}, LowEx system. The black line presents the rate of thermal load on the body surface area ($\dot{L}$ = 0, PMV = 0)*

In the case of healthcare worker who is present in the patient room only for a short period, the idea is to design an active zone inside of active space (i.e. zoning of space), without deterioration of required conditions for burn patient. Zoning is presented in Figure 76, panels could be movable per position, per high. Regarding that, LowEx system for healthcare worker could be set up to the conditions T_{ai} 16 °C and T_{mr} 30 °C (T_o 25.6 °C) with exergy consumption rate 2.20 W/m², or T_{ai} 24 °C and T_{mr} 20 °C (T_o 21.3 °C) with exergy consumption rate 2.43 W/m² and T_{ai} 20 °C and T_{mr} 25°C (T_o 23.4 °C) with exergy consumption rate 2.10 W/m². All these conditions result in $\dot{L}$ = 0 (*PMV* = 0) (Figure 75). If minimal possible exergy consumption rate (2.10 W/m²) and comfort conditions ($\dot{L}$ = 0; *PMV* = 0) have to be attained at the same time, T_{ai} and T_{mr} have to be 20 °C and 25 °C (T_o = 23.4 °C), respectively.[1]

[1] Human body exergy consumption rate for healthcare worker at conditions with T_{ai} 20 °C and T_{mr} 25 °C is lower than for example at conditions with T_{ai} 24 °C and T_{mr} 20 °C due to higher rates of output exergies and higher rate of stored exergy. Metabolic thermal exergy rate is higher due to larger difference of T_{cr}–T_{ai} and larger value of T_{cr}/T_{ai}, higher difference of T_{sk}–T_{ai} and larger value of T_{sk}/T_{ai}. Warm radiant exergy rate absorbed from the whole skin and clothing surfaces appears, because $T_{mr} > T_{ai}$. Warm radiant exergy rate discharged from the whole skin and clothing surfaces is higher due to larger difference of T_{cl}–T_{ai}. Warm convective exergy rate discharged from the whole skin and clothing surfaces is higher due to larger difference of T_{cl}–T_{ai} and smaller value of T_{ai}/T_{cl}. The exergy rate of exhalation and evaporation of sweat is higher due to larger difference of T_{cr}–T_{ai} and larger value of T_{cr}/T_{ai}, higher difference of T_{cl}–T_{ai} and larger value of T_{cl}/T_{ai}. The rate of stored exergy is higher due to smaller value of T_{ai}/T_{cr} and smaller value of T_{ai}/T_{sk}.

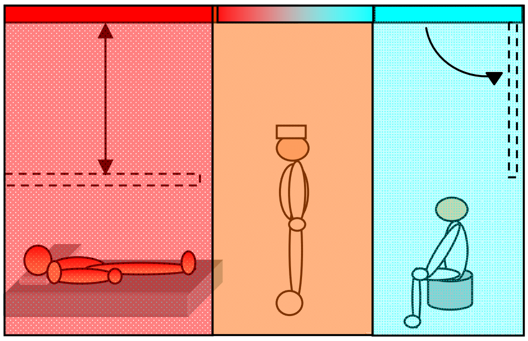

Fig 76: *Created active zones (red, orange, blue) inside of active space*

LowEx system above the seated visitor could be set up to T_{ai} 17.0 °C, T_{mr} 25 °C (T_o = 22.3 °C) with 4.95 W/m^2 of human body exergy consumption rate[1]. Thus, optimal conditions for visitor's active zone are created, with minimal possible human body exergy consumption rate and attained comfort conditions ($\dot{L} = \dot{M}$; PMV = 0), while the required and recommended conditions for burn patient inside his active space stay unchanged at the same time. And even more, if the surfaces are heated to higher temperatures and not the air, the number of air changes could be increased and better indoor air quality and thermally comfortable conditions are attained. Lower temperatures lead to lower building energy use, as it was proven in section 7.5 (Energy use for heating and cooling).

[1] Human body exergy consumption rate for visitor at conditions with T_{ai} 17 °C and T_{mr} 25 °C is lower than for example at conditions with T_{ai} 20 °C and T_{mr} 16 °C due to higher rates of output exergies and higher rate of stored exergy. Metabolic thermal exergy rate is higher due to larger difference of T_{cr}–T_{ai} and larger value of T_{cr}/T_{ai}, higher difference of T_{sk}–T_{ai} and larger value of T_{sk}/T_{ai}. Warm radiant exergy rate absorbed from the whole skin and clothing surfaces appears, because $T_{mr} > T_{ai}$. Warm radiant exergy rate discharged from the whole skin and clothing surfaces is higher due to larger difference of T_{cl}–T_{ai}. Warm convective exergy rate discharged from the whole skin and clothing surfaces is higher due to larger difference of T_{cl}–T_{ai} and smaller value of T_{ai}/T_{cl}. The exergy rate of exhalation and evaporation of sweat is higher due to larger difference of T_{cr}–T_{ai} and larger value of T_{cr}/T_{ai}, larger difference of T_{cl}–T_{ai} and larger value of T_{cl}/T_{ai}. The rate of stored exergy is higher due to smaller value of T_{ai}/T_{cr} and smaller value of T_{ai}/T_{sk}.

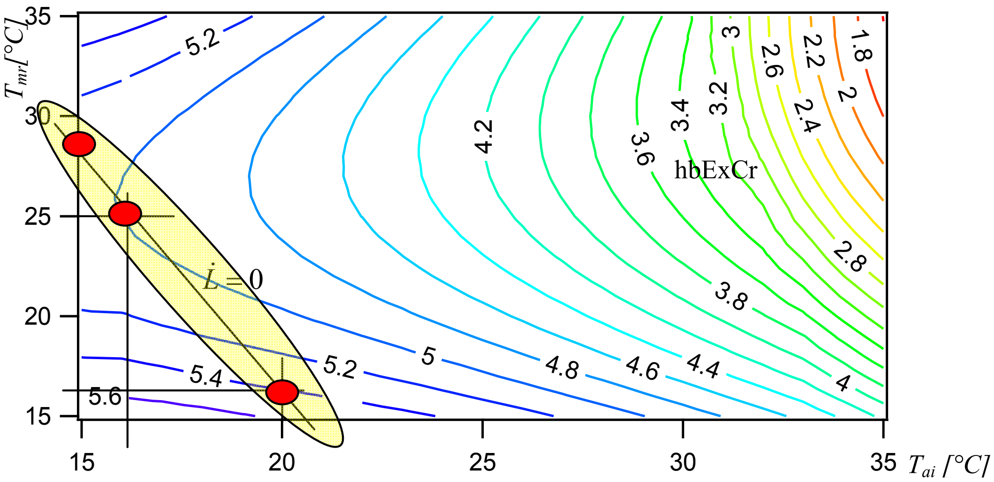

Fig 77: *Human body exergy consumption rate (hbExCr) [W/m²], visitor, as a function of T_{ai} and T_{mr}, LowEx system. The black line presents the rate of the rate of thermal load on the body surface area surface area ($\dot{L}$ = 0, PMV = 0)*

A new possibility of active regulation of human body exergy consumption rate and creation of thermal comfort conditions for all users is proposed. The designed LowEx system enables active regulation of thermal comfort zone by changing the settings of T_{mr} and T_{ai} (from zone A to zone B, C, D) for every individual separately without deterioration of one's comfort or even required conditions (Figure 78). In such way, an individualized climate is designed. And even more, lower temperatures lead to lower energy use. H/C system has to enable human body to dissipate warm exergy at an appropriate and manageable rate.

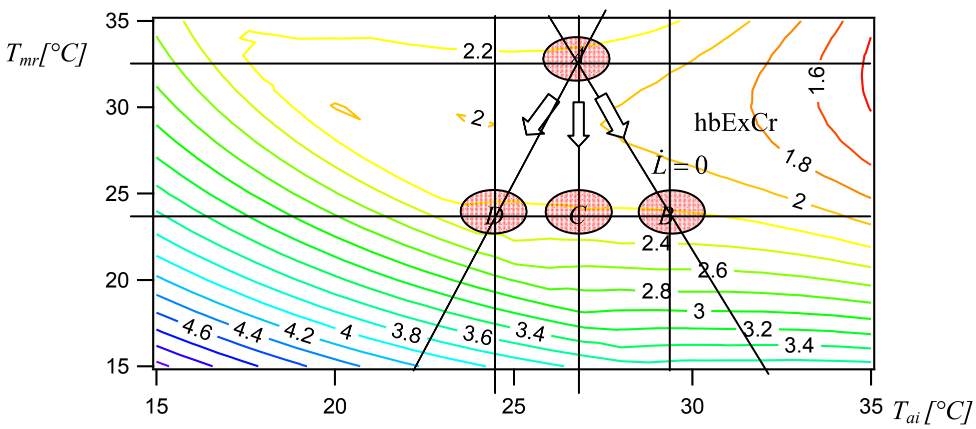

Fig 78: *Active regulation of human body exergy consumption rate [W/m²], health and comfort conditions with LowEx system. Red cycles are active zones with desired conditions*

Regarding desired conditions, we actively regulate thermal comfort zone and exergy consumption rate by setting T_{ai} and T_{mr} inside of LowEx system, e.g.:

- If the desired conditions are lower T_{mr}, besides unchanged comfort and exergy consumption rate, we have to set up the change of LowEx system from T_{ai} 27 °C, T_{mr} 33 °C (zone A) to T_{ai} 29 °C, T_{mr} 24 °C (zone B);
- If the desired conditions are lower T_{mr}, besides unchanged T_{ai} and human body exergy consumption rate we also have to set up the change of LowEx system from T_{mr} 33 °C (zone A) to T_{mr} 24 °C (zone C);
- If the desired conditions are lower T_{mr} and lower T_{ai}, lower building exergy consumption, beside unchanged human body exergy consumption rate we also have to set up the change of LowEx system from T_{ai} 27 °C, T_{mr} 33 °C (zone A) to T_{ai} 24 °C, T_{mr} 24 °C (zone D) (Figure 78).

The presented tool for active regulation of human body exergy consumption rate, health and comfort conditions could be upgraded to ICSIE and connected with LowEx system. As a starting point, the formulas for average burn patients could be modified according to the individual response. According to the required conditions and individual needs, a healthcare worker could set up input data in ICSIE system (T_{ai}, T_{mr}, RH_{in}, etc.) and regulate separate parts of human body exergy balance. In case of different users, separate panels could be regulated separately and zones could be created–comfort zones for visitor/healthcare worker, heath zones for burn patients (Figure 79).

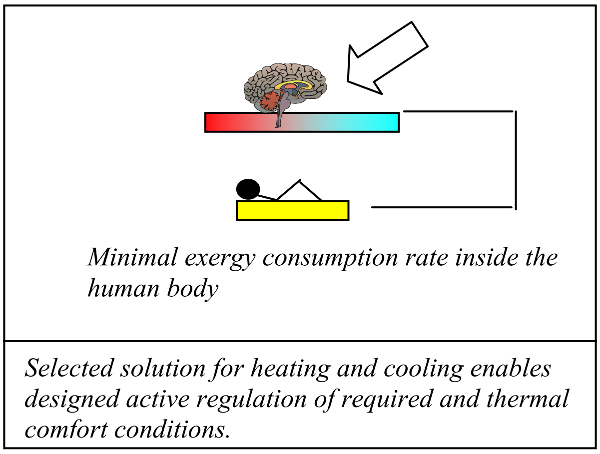

Fig 79: *Active regulation of human body exergy consumption rate [W/m²], health and comfort conditions with LowEx system*

The system could be easily applied also for other users. For example, newborns, especially premature ones, have four times greater heat loss to the same unit of body weight than adults (Bruck, 1961) and require much higher operative temperatures. The heat loss can occur by different mechanisms, of which, radiation and evaporation are the most important (Oldenburg Neto and Amorim, 2006). Hey and Katz (1970) found out that operative environmental temperature needed to provide thermal neutrality for healthy baby nursed naked in draft–free surroundings of uniform temperature and moderate humidity after birth is in the range from 35 °C (for 1 kg birth weight) to 32 °C (more than 2.5 kg of birth weight). As a standard in the care of premature newborns to prevent evaporative losses. Oldenburg Neto and Amorim (2006) proposed the use of incubators with humidity control. However, for complete control of heat loss during all nursing procedures LowEx systems are more convenient.

The presented approach also presents a good strategy for solving the problem of building energy use. At the end the overall harmonization of individual demands and needs for users is attained, required regulated demands for hospital environment are fulfilled and at the same time building energy use is minimized.

8 CONCLUSIONS

On the path towards sustainability of our future buildings with attained 20–20–20 targets of EPBD (Directive 2010/31/EU), a new approach to solving the problem of high energy use has to be taken. With this respect, a method that enables holistic manipulation of the problems connected to the design of living and working environment and searching of alternative solutions is developed. The problem of high energy use is treated hierarchically, from the analysis of building envelope characteristics at different locations, to evaluation of indoor conditions and thermal comfort of their users. The developed method enables transparency of energy flows between different hierarchy levels. Moreover, exergy concept jointly treats processes inside the human body and processes in a building. In this respect, two models are jointly used: human body exergy balance model and model of building energy use.

Preliminary study of the whole heating system is made with exergy and energy analysis at selected locations. Three cases of our test room are placed in three typical Slovenian climate zones. The results are compared and discussed. The exergy concept shows explicitly how much exergy is consumed during separate stages of space heating. The results of exergy analysis and the results of energy analysis are the same, if only the input amount of supplied energy is observed. For example, the space heating exergy demand that presents part of total exergy consumption rate is 1705 W, while the average heating energy demand calculated with TOST is 1898 W for Case 1 in the Continental part, 1776 W and 1968 W in the Alps, and 1364 W and 1516 W in the Mediterranean. The small difference appears, because of different calculation approaches; calculation of the average heating energy demand is based on ISO 13790, and the exergy calculation followed the procedure described by Shukuya (1994). However, the exergy analysis enables a more precise insight into how, not only how much exergy is consumed, more or less at every step in the process of space heating. In this way, exergy analyses enable better understanding of the problem, show possible solutions and enable the search for an effective and practical solution to minimize the building exergy consumption. In this respect, both analyses have to be combined. Results show that insulation has much bigger effect than boiler efficiency. However, the most effective solution is to improve building envelope together with boiler efficiency. In the framework of selected individual steps of analysis it is more beneficial to reduce the heating exergy load by installing thermally well–insulated glazing and exterior walls (988 W reduction in the Continental, 1029 W in the Alps and 790 W in the Mediterranean part) than to develop a boiler with an extremely high thermal efficiency (160 W reduction in the Continental, 167 W in the Alps and 128 W in the Mediterranean part) in order to decrease the rate of total exergy consumption. The most effective solution is holistic approach, i.e. improvement of boiler efficiency together with thermally well insulated building envelope (1148 W reduction in the Continental, 1196 W in the Alps and 918 W in the Mediterranean part) (Table 37).

Table 37: *Total exergy consumption rate for the whole process of space heating and building exergy saving potential because of interventions [W]*

Climate zone	***Continental***	***Alps***	***Mediterranean***
Total exergy consumption rate of original test room	*2003*	*2086*	*1603*
Intervention on building envelope	*988*	*1029*	*790*
Intervention on boiler efficiency	*160*	*167*	*128*
Holistic approach	***1148***	***1196***	***918***

By comparing heat flow rate and exergy flow rate with respect to different thermally insulated walls, it can be established that significant difference appears (78 % less heat flow rate and exergy flow rate) with improved thermal insulation between Case 1 and Case 2, but more importantly, the location should be taken into consideration (83 % less heat flow rate and exergy flow rate) with improved thermal insulation between Case 1 and Case 2. Exergy analyses present the basis for bioclimatic design that allows taking into consideration also regional characteristics. Important aspect on the level of building design and on the level of thermal comfort is warm radiant exergy rate in relation to space heating. As can be seen, the differences between surface temperatures or warm radiant exergy flow rates are not so significant, when focusing only on three cases at one particular location (0.4–2.3 °C, 68–558 mW/m^2) in the Continental part). However, when the changes in surface temperatures are taken into consideration together with warm radiant exergy flow rates and also location characteristics, the differences become important too (2.3 °C and 300 mW/m^2 difference between Case 1 in the Alps and Case 3 in the Mediterranean part). It is important to recognize that the walls with good thermal insulation result in higher inside surface temperature, which works as a kind of system that emits warm radiant exergy, i.e. a LowEx system. In the future design of building envelope systems such calculation of radiant exergy should be applied.

The proposed methodology could be applied with any other system configurations, from building as a whole, to any of sub–systems, such as a cooling or ventilation systems. Holistic approach is useful for identifying and analyzing the problems of building energy use and as a starting point for relevant interventions. Further exergy analysis should include internal and solar heat gains. For sustainable future buildings, energy and exergy analyses have to be combined and also software for dynamical simulations should be developed. First priority action for our test room that is obligatory for the future is improvement of building envelope together with effective H/C system.

Heating and cooling of spaces could be done with conventional or LowEx systems. However, heating–cooling panels have proved to have positive impact on thermal comfort conditions (Zöllner, 1985; Dongen, 1985; Skov and Valbjorn, 1990; Cox et al., 1993; Fort, 1995; Olesen, 1997; 1998; Dijk et al., 1998; Museums energy efficiency …, 1999; Sammaljarvi, 1998; Krainer et al., 2007, 2011; Dovjak et al., 2010c, 2011a, 2011c, 2011f; Košir et al., 2010; Dovjak and Shukuya, 2011d) and decreased building energy use (Olesen, 1997, 1998; Dijk et al., 1998, Museums energy efficiency …, 1999; Krainer et al., 2007, 2011; Sakulpipatsin, 2008; Dovjak et al., 2010c, 2011a, 2011c, 2011e, 2011f; Košir et al., 2010; Dovjak and Shukuya, 2011d). Moreover, in future buildings, the possibility of the design of H/C system that is based on individual user needs besides minimal energy use for heating and cooling has to be well considered.

Systems are compared on the basis of measured energy use for heating and cooling, calculated *PMV* index and human body exergy consumption rate. Calculations of human body exergy consumption rate enable us to see more precisely how a subject produces and gives away heat depending on different environmental conditions. Past studies (Isawa et al., 2003; Shukuya et al., 2003; Shukuya, 2006a; Simone et al., 2010, 2011a, 2011b; Tokunaga and Shukuya, 2011) on average test subjects showed that in the framework of thermal comfort conditions only the effect of temperature was taken into consideration, when exergy consumption rate was minimal. The present study on average test subjects shows that relative humidity in combination with air temperature has an important effect on thermal comfort conditions and human body exergy rate and is further on analyzed also on individual test subjects.

Results of the study on individual test subjects in normal room conditions show that the human body exergy rates and *PMV* index vary between individuals for both systems, even if they are exposed to the same environmental conditions. Better comfort conditions (*PMV* closer to 0) are created in the room with LowEx system for all subjects. Results of individual human body exergy consumption rate and *PMV* index in extreme conditions show that at hot and dry conditions the human body exergy rate is the largest while at hot and humid conditions is the minimal. The same maximal and minimal values could be found for both systems. The difference appears if the whole human body exergy balance is taken into consideration.

Room conditions that differ from normal ones defined in ANSI/ASHRAE Standard 55, result in uncomfortable conditions for all healthy individuals. For specific subject, such as burn patient, much higher T_{ai} and RH_{in} are required. ASHRAE Handbook HVAC applications … (2007) defines that air temperature has to be 32 ºC and RH_{in} 95 % for burn patient room. These required conditions are healing oriented conditions for patient and cause the lowest possible human body exergy consumption rate valid for thermoregulation, lower metabolic thermal exergy rate and also lower exergy rates of exhalation and evaporation of sweat, radiation and convection, as it was mentioned by Wilmore et al. (1975), Caldwell (1976, 1991), Herndon et al. (1987a, 1988), Cone at al. (1988), Caldwell et al. (1992), Herndon and Parks (1992), Marin et al. (1992), Wallace et al. (1994), Kelemen et al. (1996), and Herndon and Tompkins (2004). If we expose a burn patient to different conditions than the required ones, the exergy consumption rate will become higher, due to higher metabolic thermal exergy rate, and also higher exergy rates of evaporation and exhalation of sweat, radiation and convection. That will results at conditions which do not contribute to the medical treatment and have to be avoided. However, with creating optimal microclimate conditions for burn patient, we do not heal the patients (clinically), but we contribute to their thermal comfort and health conditions and indirectly to quicker recovery (thermal discomfort and unhealthy conditions could synergistically affect users` health).

If the required conditions for test room are created with conventional system (T_{ai} = T_{mr} 32 °C, RH_{in} 80 %) and with LowEx system (T_o 32 °C, RH_{in} 80 %) separately, the results of human body exergy consumption rates for burn patient are almost the same. However, in the case of LowEx system, the required T_o can be created with different combinations of T_{ai} and T_{mr}. The significant difference between systems appears, if the whole chain of human body exergy consumption inside of burn patient is taken into consideration. In the case of LowEx system, exergy consumption rate in burn patient is lower, due to lower rate of metabolic thermal exergy, lower exergy rate of exhalation and evaporation of sweat, and also lower rates of warm radiative and warm convective exergies transferred by the whole skin and clothing surfaces. *PMV* value is lower in the case of conventional system, but regarding the mentioned facts we could conclude that for burn patient conditions which support medical treatment have to be taken as priority and not comfort ($\dot{L}$ = 0; *PMV* = 0). Regarding Eq. (1) (section 6.1.4.3), exergy consumption rate inside the burn patient is minimized only with active regulation of input and output flows, without regulating stimulation of exergy storage with medicaments. In future standards for thermal environment, T_o has to be required and not just $T_{ai.}$

The measured energy use for space heating was by 11–27 % lower when using LowEx system than for conventional system. The energy use for space cooling was by 30–73 % lower with for LowEx system. The reason for relatively low efficiency of LowEx system during the experiment was small ceiling surface area that was heated, i.e. 25 % of ceiling surface. For application in real environment, the larger part of the ceiling has to be covered with panels. The calculated energy use for heating in case of four times larger surface area of panels is 40 % lower energy use. The results of temperature measurements show for both systems that surface temperatures on outdoor walls and windows are considerably affected by outdoor temperature variations. Poorly insulated building envelope causes larger transmission heat losses and presents a weak part of building system, as it was proven in the analysis of exergy consumption patterns. Moreover, temperatures are much more constant in a room with LowEx system than in a room with conventional H/C system. One of the most important advantages of LowEx systems is the possibility to actively control the surface temperatures, while in the conventional system surfaces are heated or cooled indirectly with air. The most effective was marble plate and the least gypsum board, especially due to higher thermal conductivity of a marble (λ_{marble} = 2.330 W/(mK), λ_{gypsum} = 0.210 W/(mK)). For future application, also movable panels have to be considered. They should be made of lighter materials with good thermal characteristics.

LowEx system enables to regulate human body exergy consumption rate, health and comfort conditions at the same time. What has to be taken as priority depends considerably on regulated demands and individual needs. For example, in hospital environment, medical treatment requirements for burn patient have to be taken as priority. In hospital room for burn patient recommended T_o has to be fulfilled with setting up optimal relation between T_{mr} and T_{ai}. In such a way, human body exergy consumption rate in burn patient could be lowered to minimal possible and thermal comfort zone is actively regulated. Inside of designed active zone, we create optimal conditions (medical treatment demand) with minimal human body exergy consumption rate and minimal loss by evaporation, radiation and convection (health conditions). The expected consequences are quicker recovery and shorter hospitalization. For other users LowEx system enables us to create active space (i.e. zoning) where comfort conditions for healthy individuals are attained without deterioration of required conditions for burn patients.

8.1 Future work

The obtained results present a basis for further phases of design (Phase II. Preliminary design). Attainment of overall individualization in hospital environment that allows individual control and generation of all influential parameters (for example microclimate, illuminance, noise) requires an extensive work from the conceptual to the research and development point of view. Inside such space, it is possible to introduce many different technologies and active systems that are harmonically adjusted with PSA of the building system. The developed LowEx system represents a theoretical background for application in hospital environment. All selected parameters and their validated interactions will also present the basis for upgrading the existing central control system for the panels and other smart systems. Dynamic analysis also presents a challenge.

Why hospital environment? The complexity of the treated environment enables huge possibilities to acquire all data about patients (patient reception, part of health documentation) for efficient medical treatment. The data could be input into a central control system and in such a way enable individual settings for the functioning of panels for individual user (Figure 80).

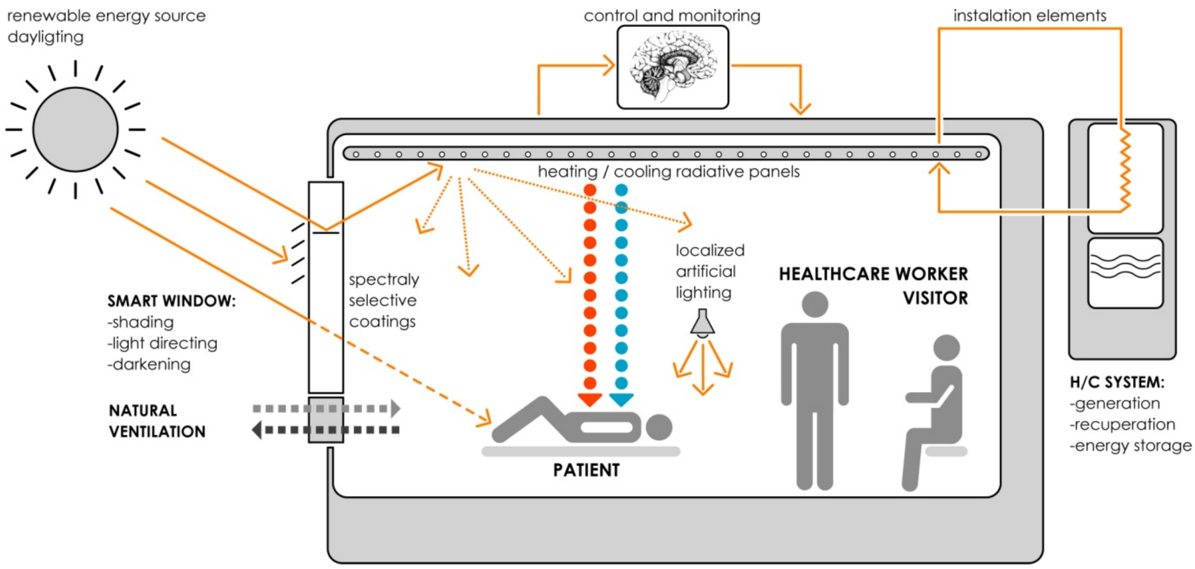

Fig 80: *Individualization of personal space inside hospital environment*

The main results of my Ph.D. work were published within ten scientific papers in peer–reviewed international journals and proceedings:

1. Dovjak M., Shukuya M., Olesen B.W., Krainer A. 2010a. Analysis on exergy consumption patterns for space heating in Slovenian buildings. Energy policy, 38, 6: 2998–3007

2. Dovjak M., Shukuya M., Olesen B.W., Krainer A. 2010b. Towards sustainable buildings with holistic consideration of high building exergy consumption in Slovenia. In: Clima 2010. 10th REHVA World Congress. Sustainable energy use in buildings 2010, Antalya, 9–12 May 2010. Arisoy A. (ed.). Brussels, Federation of European Heating and Air–Conditioning Associations: 1–8

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ANNEX A.

CALCULATION OF OPERATIVE TEMPERATURE

Calculation of the operative temperature [K] is based on air temperature and mean radiant temperature and is defined in ISO 7726. The exact equation for the operative temperature is:

$$T_o = \frac{h_c T_{ai} + \overline{h}_r \overline{T}_{mr}}{h_c + h_r}, \tag{1}$$

where T_{ai} is the air temperature, T_{mr} is the mean radiant temperature, h_c is the heat–transfer coefficient by convection [W/(m^{2}K)], h_r is the heat–transfer coefficient by radiation [W/(m^{2}K)].

Calculation of the heat–transfer coefficient by convection, h_c [W/(m^{2}K)] (from Kerslake, 1972)

$$h_c = 8.3\, v_a^{0.6}, \tag{2}$$

where v_a is air velocity [m/s].

Calculation of heat–transfer coefficient by radiation, h_r [W/(m^{2}K)] (from Parsons, 1993)

$$h_r = 4\varepsilon_{sk} \sigma A_r A_d (273.2 + (T_{mb} + T_{mr})/2)^3 \tag{3}$$

where ε_{sk} is emissivity of the skin surface (0.98: Chang et al., 1988), σ is Stefan–Boltzmann constant [5.67 10^{-8} W/(m^{2}K^4)], $A_r A_d$ is ratio of the area of the body exposed to radiation versus the total body surface area (0.70 for seated postures, 0.73 for standing postures), T_{mb} is mean surface temperature of the body [^{o}C], and T_{mr} is mean radiant temperature [^{o}C].

In most particular cases where the relative velocity is small (< 0.2 m/s) or where the difference between mean radiant and air temperature is small (< 4 $^\circ$C), the operative temperature can be calculated with sufficient approximation as the mean value of air and mean radiant temperature.

ANNEX B.

CALCULATION OF THE HUMAN–BODY EXERGY CONSUMPTION RATE, *PMV* INDEX

Individual human body exergy consumption rate can be calculated with human body exergy balance model developed by Shukuya et al. (2010). The model deals with human body as thermodynamic system based on exergy–entropy processes, i.e. on feeding and consuming of exergy stores, entropy generation and return of entropy back to environment. Or with other words, human body obtains exergy from food, consumes it for work, which results in entropy generation and entropy disposal into surrounding environment. Human body is treated as system consisting of core and shell and placed in active space with its environmental temperature. Energy balance equation includes net of thermal energy emerged by metabolism and stored by the core and by the shell and net of thermal energy released from the body by respiration, evaporation, radiation and convection. Entropy balance equation includes the net of thermal energy stored by human body and the net of thermal energy released from the body by respiration, evaporation, radiation and convection. The model of human body exergy balance also includes the amount of wet/dry exergy and the amount of warm/cool exergy. We talk about "warm" exergy, if the environmental temperature is lower than body temperature. Vice versa, the term "cool" exergy is used, if the environmental temperature is higher than body temperature. "Wet" exergy is the ability of the volume of air with water vapour to disperse to the surrounding environment that has lower humidity, and, vice versa, "dry" exergy is the ability of volume of air with water vapour to disperse to the surrounding environment that has higher humidity (Shukuya et al., 2010).

The six most relevant and independent variables, necessary for defining also the indoor thermal comfort conditions, are the input data in the human–body exergy balance calculation tool: T_{ao}, RH_{out}, T_{ai}, RH_{in}, v_a, room dimensions, metabolic rate, effective clothing insulation. The outputs of the calculation tools are:

- skin and clothing surface temperatures;
- input–output exergy flow rates from human–body;
- exergy consumption rate within human body.

The thermal exergy balance of human body consisting of two subsystems, the core and the shell, is derived as follows by combining the energy and entropy balance equations together with the environmental temperature for exergy calculation. The calculated exergy values from Eqs. (1)−(12) are expressed as rates, W/m^2 (body surface).

[The rate of warm exergy generated by metabolism]
+[The rate of warm/cool and wet/dry exergies of inhaled humid air]
+[The rate of warm and wet exergies of liquid water generated in the core by metabolism]
+[The rate of warm/cool and wet/dry exergies of the sum of liquid water generated in the shell by metabolism and dry air to let the liquid water disperse]
+[The rate of warm/cool radiant exergy absorbed by the whole of skin and clothing surfaces]
−[Exergy consumption rate valid for thermoregulation]
= [The rate of warm exergy stored in the core and the shell]
+[The rate of warm and wet exergies of exhaled humid air]
+[The rate of warm/cool radiant exergy discharged from the whole of skin and clothing surfaces]
+[The rate of warm/cool exergy transferred by convection from the whole of skin and clothing surfaces into the surrounding air] (1)

The rate of warm exergy generated by metabolism

$$\dot{M}\left(1-\frac{T_{ao}}{T_{cr}}\right)dt \tag{2}$$

The rate of warm/cool[1] and wet/dry[2] exergies of inhaled humid air

$$\dot{V}_{in}\left[\begin{array}{l}\left\{c_{pa}\left(\frac{m_a}{RT_{ai}}\right)(P-p_{vr})+c_{pv}\left(\frac{m_w}{RT_{ai}}\right)p_{vr}\right\}\left\{(T_{ai}-T_{ao})-T_{ao}\ln\frac{T_{ai}}{T_{ao}}\right\}+\\ \frac{T_{ao}}{T_{ai}}\left\{(P-p_{vr})\ln\frac{P-p_{vr}}{P-p_{vo}}+p_{vr}\ln\frac{p_{vr}}{p_{vo}}\right\}\end{array}\right]dt \tag{3}$$

[1] Whether exergy transfer by convection or radiation turns out to be "warm" or "cool" is determined by the sign of Carnot factor, and whether it becomes outgoing or incoming is determined by the sign of the product of temperature difference and Carnot factor.

[2] The moist air system contains wet exergy, if vapour pressure of moist air system p_{vr} is higher than vapour pressure of outdoor air p_{vo}. The moist air system contains dry exergy, if vapour pressure of moist air system p_{vr} is lower than vapour pressure of outdoor air p_{vo}.

The rate of warm and wet exergies of liquid water generated in the core by metabolism

$$\dot{V}_{w-core}\,\rho_w\left[c_{pw}\left\{\left(T_{cr}-T_{ao}\right)-T_{ao}\ln\frac{T_{cr}}{T_{ao}}\right\}+\frac{R}{m_w}T_{ao}\ln\frac{p_{vs}\left(T_{ao}\right)}{p_{vo}}\right]dt \tag{4}$$

The rate of warm/cool[1] and wet/dry[2] exergies of the sum of liquid water generated in the shell by metabolism and dry air to let the liquid water disperse

$$\dot{V}_{w-shell}\,\rho_w\left[c_{pw}\left\{\left(T_{sk}-T_{ao}\right)-T_{ao}\ln\frac{T_{sk}}{T_{ao}}\right\}+\frac{R}{m_w}T_{ao}\left\{\ln\frac{p_{vs}\left(T_{ao}\right)}{p_{vo}}+\frac{P-p_{vr}}{p_{vr}}\ln\frac{P-p_{vr}}{P-p_{vo}}\right\}\right]dt \tag{5}$$

The rate of warm/cool[1] radiant exergy absorbed by the whole of skin and clothing surfaces

$$f_{eff}\,f_{cl}\sum_{j=1}^{N}\alpha_{pj}\varepsilon_{cl}h_{rb,hb}\frac{\left(T_{surf}-T_{ao}\right)^2}{T_{surf}+T_{ao}}dt \tag{6}$$

Exergy consumption rate, which is only for thermoregulation

$$\delta S_{gen}\;T_0 \tag{7}$$

The rate of warm exergy stored in the core and the shell

$$Q_{core}\left(1-\frac{T_{ao}}{T_{cr}}\right)dT_{cr}+Q_{shell}\left(1-\frac{T_{ao}}{T_{sk}}\right)dT_{sk} \tag{8}$$

The rate of warm and wet exergies of exhaled humid air

$$\dot{V}_{out}\left[\begin{array}{l}\left\{c_{pa}\left(\dfrac{m_a}{RT_{cr}}\right)\left(P-p_{vs}\left(T_{cr}\right)\right)+c_{pv}\left(\dfrac{m_w}{RT_{cr}}\right)p_{vs}\left(T_{cr}\right)\right\}\left\{\left(T_{cr}-T_{ao}\right)-T_{ao}\ln\dfrac{T_{cr}}{T_{ao}}\right\}\\+\dfrac{T_{ao}}{T_{cr}}\left\{\left(P-p_{vs}\left(T_{cr}\right)\right)\ln\dfrac{P-p_{vs}\left(T_{cr}\right)}{P-p_{vo}}+p_{vs}\left(T_{cr}\right)\ln\dfrac{p_{vs}\left(T_{cr}\right)}{p_{vo}}\right\}\end{array}\right]dt \tag{9}$$

The rate of warm/cool[1] exergy of the water vapour originating from the sweat and wet/dry[2] exergy of the humid air containing the evaporated sweat

$$\dot{V}_{w-shell}\,\rho_w\left[c_{pw}\left\{\left(T_{cl}-T_{ao}\right)-T_{ao}\ln\frac{T_{cl}}{T_{ao}}\right\}+\frac{R}{m_w}T_{ao}\left\{\ln\frac{p_{vr}}{p_{vo}}+\frac{P-p_{vr}}{p_{vr}}\ln\frac{P-p_{vr}}{P-p_{vo}}\right\}\right]dt \tag{10}$$

---

[1] Whether exergy transfer by convection or radiation turns out to be "warm" or "cool" is determined by the sign of Carnot factor, and whether it becomes outgoing or incoming is determined by the sign of the product of temperature difference and Carnot factor.

[2] The moist air system contains wet exergy, if vapour pressure of moist air system p_{vr} is higher than vapour presure of outdoor air p_{vo}. The moist air system contains dry exergy, if vapour pressure of moist air system p_{vr} is lower than vapour pressure of outdoor air p_{vo}.

The rate of warm/cool[1] radiant exergy discharged from the whole of skin and clothing surfaces

$$f_{eff}\, f_{cl}\, \varepsilon_{cl}\, h_{rb,hb} \frac{(T_{cl} - T_{ao})^2}{T_{cl} + T_{ao}} dt \tag{11}$$

The rate of warm/cool[1] exergy transferred by convection from the whole of skin and clothing surfaces into the surrounding air

$$f_{cl} h_{ccl} (T_{cl} - T_{ai}) \left(1 - \frac{T_{ao}}{T_{cl}}\right) dt \tag{12}$$

1. We assume that $T_{ai} = T_{ao}$, $p_{vo} = p_{vr}$, $T_{surf} = T_{mr}$.

2. The value of $\dot{V}_{in}$ can be determined from the empirical formula given for human–body energy balance calculations as a function of metabolic generation rate.

$$\dot{V}_{in} = 1.2\ 10^{-6}\ \dot{M} \tag{13}$$

3. The value of $\dot{V}_{w-core}$ can be determined from the empirical formula as a function of metabolic energy generation rate and water–vapour pressure in the room space.

$$\dot{V}_{w-core} = 1.2\ 10^{-6}\ \dot{M}\ (0.029 – 0.049\ 10^{-4} p_{vr}) \tag{14}$$

4. The value of $\dot{V}_{w-shell}\ \rho_w$ is given as the product of the skin wettedness, w [dimensionless], and the rate of the maximum evaporative potential from the skin surface to the surrounding room space, $\dot{E}_{\max}$ [W/m^2], divided by the latent–heat value of evaporation of the liquid water at 30 °C (= 2430 J/g).

$$\dot{V}_{w-shell}\ \rho_w = w \dot{E}_{\max} / 2430 \tag{15}$$

The value of w is determined by the calculation procedure given for effective temperature based on human body energy balance by Gagge et al. (1969, 1971, 1972, 1986). The value of $\dot{E}_{\max}$ can be determined as the product of evaporative heat transfer coefficient, which is proportional to convective heat–transfer coefficient via a Lewis–relation constant, and the difference water–vapour pressure between liquid water at skin–surface temperature and room air.

[1] Whether exergy transfer by convection or radiation turns out to be "warm" or "cool" is determined by the sign of Carnot factor, and whether it becomes outgoing or incoming is determined by the sign of the product of temperature difference and Carnot factor.

5. The values of Q_{core} and Q_{shell} are given by the following formulae.

$$Q_{core} = (1 - \alpha_{sk})(m_{body}/A_{du})C_{body} \text{ and } Q_{shell} = \alpha_{sk}(m_{body}/A_{du})C_{body}, \quad (16)$$

where α_{sk} is the fractional skin mass depending on the blood flow rate to the body shell/skin; m_{body}/A_{du} is the ratio of body mass to body surface area [kg/m^2]; and C_{body} is the specific heat capacity of human body that is 3490 J/(kgK).

6. The value of $\dot{V}_{out}$ is assumed to be equal to that of $\dot{V}_{in}$.

7. We assume that the boundary–surface temperature of human–body system is represented by the average clothing temperature. Therefore, the water vapour pressure for the calculation of wet/dry exergy flow rate of humid air coming out of the human body system should be based on the value at the clothing surface. However, in reality, much dispersion of water vapour takes places directly at the skin surface, such as forehead, neck, arms, etc. For this reason, together with the avoidance of unnecessary complicated calculation, we use water vapour pressure in the room space for the calculation of wet/dry exergy flow rate of the humid air containing the evaporated sweat.

8. The value of h_{ccl} can be determined by one of the empirical formulae of convective heat–transfer coefficient of the human body as a whole, which is given for human–body energy balance calculation by Gagge et al. (1986).

Input exergies, output exergies, stored exergy and exergy consumption in human body exergy balance are presented in Table 38. The calculated exergy values are expressed as rates, W/m^2 (body surface).

__Table 38:__ Input exergies, output exergies, stored exergy and exergy consumption in human body exergy balance

Input exergies	
Metabolic thermal exergy (inner part)	*= Warm exergy generated by metabolism* *+Warm and wet exergies of liquid water generated in the core by metabolism* *+Warm/cool and wet/dry exergies of the sum of liquid water generated in the shell by metabolism and dry air to let the liquid water disperse* *+Warm exergy stored in the core and the shell* *(17)*
Warm/cool radiant exergy in	*= Warm/cool radiant exergy absorbed by the whole of skin and clothing surfaces (18)*
Breath air in	*= Warm/cool and wet/dry exergies of inhaled humid air (19)*
Warm/cool convective exergies in	*= Warm/cool convective exergy by absorbed from the whole of skin and clothing surfaces into the surrounding air (20)*
Exergy consumption, which is only for thermoregulation	
Stored exergy	*= Warm exergy stored in the core and the shell* *(21)*
Output exergies	
Exhalation and evaporation of sweat out	*= Warm and wet exergies of exhaled humid air* *+Warm/cool exergy of the water vapour originating from the sweat and wet/dry exergy of the humid air containing the evaporated sweat* *(22)*
Warm/cool radiant exergy out	*= Warm/cool radiant exergy discharged from the whole of skin and clothing surfaces (23)*
Warm/cool convective exergy out	*= Warm/cool exergy transferred by convection from the whole of skin and clothing surfaces into the surrounding air (24)*

Predicted mean vote (*PMV*) is an index that predicts the mean value of the votes of a large group of persons on the 7–point thermal sensation scale (+3 hot,+2 warm,+1 slightly warm, 0 neutral,–1 slightly cool,–2 cool,–3 cold), based on the heat balance of the human body. Thermal balance is obtained when the internal heat production in the body is equal to the loss of heat to the environment (ISO 7730).

PMV was calculated with software and it followed the procedure developed by Fanger (1970):

$$PMV = (0.303\, e^{-0.036\dot{M}} + 0.028)\, \dot{L}, \tag{25}$$

where $\dot{L}$ is the rate of thermal load on the body surface area [W/m^2], $\dot{M}$ is metabolic generation rate [W/m^2]. $\dot{L}$ was calculated as follows:

$$\dot{L} = \dot{H} - \dot{E}_d - \dot{E}_{sw} - \dot{E}_{re} - \dot{R}_{rad} - \dot{C}, \tag{26}$$

where $\dot{H}$ = rate of internal heat production on the body surface area [W/m^2], $\dot{E}_d$ = rate of evaporative heat transfer via skin diffusion due to water vapour diffusion through the skin surface [W/m^2], $\dot{E}_{sw}$ = rate of heat transfer by sweat evaporation from the skin surface [W/m^2], $\dot{E}_{re}$ = rate of latent heat loss due to respiration through the skin surface [W/m^2], $\dot{R}_{rad}$ = rate of heat transfer via radiation from the surface of a clothed body [W/m^2], $\dot{C}$ = rate of heat transfer via convection from the surface of a clothed body [W/m^2].

ANNEX C.

ENERGY ANALYSIS VS. EXERGY ANALYSIS OF THERMAL COMFORT

Traditional methods of human thermal comfort are based on the first law of thermodynamics. These methods use an energy balance of the human body to determine heat transfer between the body and its environment. By contrast, the second law of thermodynamics introduces the concept of exergy. It enables the determination of exergy consumption within the human body dependent on personal and environmental factors. For comparison between exergy and energy analysis of thermal comfort we assume a human body as a biological system of a core and a shell (two−node model) inside of steady–state conditions. Thermal exergy balance of human body (Shukuya et al., 2010) were derived by combining the water balance equation, energy balance equation, the entropy balance equation and environmental temperature under steady–state condition. All of them are the resultant equations of the mathematical operations described in Shukuya et al. (2010). Water balance equation, energy, entropy and exergy balance equations (Shukuya et al., 2010) are present in Tables 39−41. The calculated exergy values are expressed as rates, W/m^2 (body surface).

Table 39: *Energy, entropy and exergy balance equations*

Energy balance eq.:	$\sum E_{in,hb} = \sum E_{out,hb}$ *(1)*
Entropy balance eq.:	*entropy generation* $\sum S_{in,hb} + S_{gen,hb} = \sum S_{out,hb}$ *(2)*
Exergy balance eq.:	*energy balance eq., entropy balance eq., environmental temperature, exergy consumption (3)*

Table 40: *Water balance equation*

Core:	***Shell:***
∑Water in core = ∑Water out core (4)	*∑Water in shell = ∑Water out shell (5)*
Inhaled humid air+liquid water by met in the core+blood flow into the core = exhaled humid air+blood flow out of the core (6)	*Liquid water by met in the shell+blood flow into the shell = sweat+blood flow out of the shell (7)*
Water balance eq.	
∑Water in	*= ∑Water out in the core and shell (8)*
[The inhaled humid air]+[The liquid water generated by metabolism in the core]+[The liquid water generated by metabolism in the shell] (9)	*= [The exhaled humid air]+[The liquid water secreted as sweat at the skin surface] (10)*

Table 41: Energy balance equations

$\sum E_{in,hb}$	$= \sum E_{out,hb}$ (11)
[Thermal energy emerged by metabolism] +[Enthalpy of the inhaled humid air] +[Enthalpy of the liquid water generated in the core by metabolism] +[Enthalpy of the liquid water generated in the shell by metabolism] +[Radiant energy absorbed by the whole of skin and clothing surface] (12)	= [Thermal energy stored in the core and the shell] +[Enthalpy of the exhaled humid air] +[Enthalpy of the water vapour originated from the sweat secreted] +[Radiant energy emitted by the whole of skin and clothing surfaces] +[Thermal energy transferred by convection from the whole of skin and clothing surfaces into the surrounding air] (13)

Table 42: Entropy balance equations

$\sum S_{in,hb}+S_{gen,hb}$	$= \sum S_{out,hb}$ (14)
[Thermal entropy given to the body by metabolism] +[Entropy of the inhaled humid air] +[Entropy of the liquid water generated in the core by metabolism] +[Entropy of the liquid water generated in the shell by metabolism] +[Radiant entropy absorbed by the whole of skin and clothing surfaces] +[Entropy generation] (15)	= [Thermal entropy stored in the core and the shell] +[Entropy of the exhaled humid air] +[Entropy of the water vapour originated from the sweat secreted and dispersing into the surrounding space] +[Radiant entropy discharged from the whole of skin and clothing surfaces] +[Thermal entropy given off by convection from the whole of skin and clothing surfaces] (16)

Table 43: Exergy balance equations

Energy+entropy, environmental temperature	
[Warm exergy generated by metabolism] +[Warm/cool and wet/dry exergies of the inhaled humid air] +[Warm and wet exergies of the liquid water generated in the core by metabolism] +[Warm/cool and wet/dry exergies of the sum of liquid water generated in the shell by metabolism and dry air to let the liquid water disperse] +[Warm/cool radiant exergy absorbed by the whole of skin and clothing surfaces] –[Exergy consumption] (17)	= [Warm exergy stored in the core and the shell] +[Warm and wet exergies of the exhaled humid air] +[Warm/cool exergy of the water vapour originating from the sweat and wet/dry exergy of the humid air containing the evaporated water from the sweat] +[Warm/cool radiant exergy discharged from the whole of skin and clothing] +[Warm/cool exergy transferred by convection from the whole of skin and clothing surfaces into the surrounding air] (18)

Energy balance model (Table 44) is based on the modified energy balance two– node model of the human body by Gagge at al. (1971, 1986). Modifications include the change of the net thermal energy transfer due to the humid−air transport by breathing and the evaporation of sweat into five explicit forms of the enthalpy values: those of inhaled and exhaled humid air, those of liquid water produced by metabolism in the body–core and in the body shell, and that of water vapour discharged from the skin surface by evaporation; the net radiant energy transfer between the human body and its surrounding was made into the explicit forms of radiant energy: one absorbed by the whole of skin and clothing surfaces and the other emitted from the whole of skin and clothing surfaces. The equations of the energy balance model are in the rate form (Table 44).

Table 44: *Energy balance model in the rate form*

$(dE_{hb}/dt) = \dot{E}_{in,hb} - \dot{E}_{out,hb}$	*(19)*
The rate of thermal energy emerged by metabolism $\dot{M}$	*(20)*
The rate of enthalpy of the inhaled humid air $\dot{V}_{in}\left[\left\{c_{pa}\left(\frac{m_a}{RT_{ai}}\right)(P - p_{vr}) + c_{pv}\left(\frac{m_w}{RT_{ai}}\right)p_{vr}\right\}\{(T_{ai} - T_{ao})\}\right]$	*(21)*
The rate of enthalpy of the liquid water generated in the core by metabolism $\dot{V}_{w-core}\,\rho_w\left[c_{pw}\left\{(T_{cr} - T_{ao}) - T_{ao}\ln\frac{T_{cr}}{T_{ao}}\right\}\right]$	*(22)*
The rate of enthalpy of the liquid water generated in the shell by metabolism $\dot{V}_{w-shell}\,\rho_w\left[c_{pw}\left\{(T_{sk} - T_{ao}) - T_{ao}\ln\frac{T_{sk}}{T_{ao}}\right\}\right]$	*(23)*
The rate of radiant energy absorbed by the whole of skin and clothing surface $f_{eff}f_{cl}\sum_{j=1}^{N}\alpha_{pj}\varepsilon_{cl}h_{rb,hb}(T_{surf} - T_{ao})$	*(24)*
The rate of thermal energy stored in the core and the shell $Q_{core} + Q_{shell}$	*(25)*
The rate of enthalpy of the exhaled humid air $\dot{V}_{out}\left[\left\{c_{pa}\left(\frac{m_a}{RT_{cr}}\right)(P - p_{vs}(T_{cr})) + c_{pv}\left(\frac{m_w}{RT_{cr}}\right)p_{vs}(T_{cr})\right\}\left\{(T_{cr} - T_{ao}) - T_{ao}\ln\frac{T_{cr}}{T_{ao}}\right\}\right]$	*(26)*
The rate of enthalpy of the water vapour originated from the sweat secreted $\dot{V}_{w-shell}\,\rho_w\left[c_{pw}\left\{(T_{cl} - T_{ao}) - T_{ao}\ln\frac{T_{cl}}{T_{ao}}\right\}\right]$	*(27)*

to be continued…

…continuation

The rate of radiant energy emitted by the whole of skin and clothing surfaces $f_{eff} f_{cl} \varepsilon_{cl} h_{rb,hb} (T_{cl} - T_{ao})$	*(28)*
The rate of thermal energy transferred by convection from the whole of skin and clothing surfaces into the surrounding air $f_{cl} h_{ccl} \left(T_{cl} - T_{ai}\right)$	*(29)*

Exergy balance model by Shukuya et al. (2010) was derived by combining the water balance equation, energy balance equation, the entropy balance equation and environmental temperature under steady–state condition. The equations of the exergy balance model are in the rate form (Table 45).

Table 45: *Exergy balance model in the rate form*

$\dot{X}_{in,hb} - \dot{X}_{consumed,hb} = \dot{X}_{stored,hb} + \dot{X}_{out,hb}$	*(30)*
The rate of warm exergy generated by metabolism $\dot{M}\left(1 - \frac{T_{ao}}{T_{cr}}\right) dt$	*(31)*
The rate of warm/cool and wet/dry exergies of inhaled humid air $\dot{V}_{in} \left[\begin{array}{l} \left\{ c_{pa} \left(\frac{m_a}{RT_{ai}} \right) \left(P - p_{vr}\right) + c_{pv} \left(\frac{m_w}{RT_{ai}} \right) p_{vr} \right\} \left\{ \left(T_{ai} - T_{ao}\right) - T_{ao} \ln \frac{T_{ai}}{T_{ao}} \right\} + \\ \frac{T_{ao}}{T_{ai}} \left\{ \left(P - p_{vr}\right) \ln \frac{P - p_{vr}}{P - p_{vo}} + p_{vr} \ln \frac{p_{vr}}{p_{vo}} \right\} \end{array} \right] dt$	*(32)*
The rate of warm and wet exergies of liquid water generated in the core by metabolism $\dot{V}_{w-core}\, \rho_w \left[c_{pw} \left\{ \left(T_{cr} - T_{ao}\right) - T_{ao} \ln \frac{T_{cr}}{T_{ao}} \right\} + \frac{R}{m_w} T_{ao} \ln \frac{p_{vs}\left(T_{ao}\right)}{p_{vo}} \right] dt$	*(33)*
The rate of warm/cool and wet/dry exergies of the sum of liquid water generated in the shell by metabolism and dry air to let the liquid water disperse $\dot{V}_{w-shell}\, \rho_w \left[c_{pw} \left\{ \left(T_{sk} - T_{ao}\right) - T_{ao} \ln \frac{T_{sk}}{T_{ao}} \right\} + \frac{R}{m_w} T_{ao} \left\{ \ln \frac{p_{vs}\left(T_{ao}\right)}{p_{vo}} + \frac{P - p_{vr}}{p_{vr}} \ln \frac{P - p_{vr}}{P - p_{vo}} \right\} \right] dt$	*(34)*
The rate of warm/cool radiant exergy absorbed by the whole of skin and clothing surfaces $f_{eff} f_{cl} \sum_{j=1}^{N} \alpha_{pj} \varepsilon_{cl} h_{rb,hb} \frac{\left(T_{surf} - T_{ao}\right)^2}{T_{surf} + T_{ao}} dt$	*(35)*
Exergy consumption rate, which is only for thermoregulation $\delta S_{gen} T_{ao}$	*(36)*
The rate of warm exergy stored in the core and the shell $Q_{core}\left(1 - \frac{T_{ao}}{T_{cr}}\right) dT_{cr} + Q_{shell}\left(1 - \frac{T_{ao}}{T_{sk}}\right) dT_{sk}$	*(37)*

to be continued…

…continuation

The rate of warm and wet exergies of exhaled humid air

$$\dot{V}_{out}\left[\begin{array}{l}\left\{c_{pa}\left(\frac{m_a}{RT_{cr}}\right)\left(P-p_{vs}\left(T_{cr}\right)\right)+c_{pv}\left(\frac{m_w}{RT_{cr}}\right)p_{vs}\left(T_{cr}\right)\right\}\left\{\left(T_{cr}-T_{ao}\right)-T_{ao}\ln\frac{T_{cr}}{T_{ao}}\right\}\\ +\frac{T_{ao}}{T_{cr}}\left\{\left(P-p_{vs}\left(T_{cr}\right)\right)\ln\frac{P-p_{vs}\left(T_{cr}\right)}{P-p_{vo}}+p_{vs}\left(T_{cr}\right)\ln\frac{p_{vs}\left(T_{cr}\right)}{p_{vo}}\right\}\end{array}\right]dt \tag{38}$$

The rate of warm/cool exergy of the water vapour originating from the sweat and wet/dry exergy of the humid air containing the evaporated sweat

$$\dot{V}_{w-shell}\,\rho_w\left[c_{pw}\left\{\left(T_{cl}-T_{ao}\right)-T_{ao}\ln\frac{T_{cl}}{T_{ao}}\right\}+\frac{R}{m_w}T_{ao}\left\{\ln\frac{p_{vr}}{p_{vo}}+\frac{P-p_{vr}}{p_{vr}}\ln\frac{P-p_{vr}}{P-p_{vo}}\right\}\right]dt \tag{39}$$

The rate of warm/cool radiant exergy discharged from the whole of skin and clothing surfaces

$$f_{eff}\,f_{cl}\,\varepsilon_{cl}\,h_{rb,hb}\,\frac{\left(T_{cl}-T_{ao}\right)^2}{T_{cl}+T_{ao}}\,dt \tag{40}$$

The rate of warm/cool exergy transferred by convection from the whole of skin and clothing surfaces into the surrounding air

$$f_{cl}h_{ccl}\left(T_{cl}-T_{ai}\right)\left(1-\frac{T_{ao}}{T_{cl}}\right)dt \tag{41}$$

ANNEX D.

CALCULATION OF AVERAGE BUILDING HEATING ENERGY DEMAND

Average building energy demand for heating is calculated for one active space. The methodology for the calculation of annual energy use for heating is defined by ISO 13790 and presents the basis for the calculation of average building energy demand. For the purpose our study, the average energy demand for the heating of our test room is expressed in W.

Heat losses, Q_l and heat gains, Q_g are calculated for each calculation period. The space heating for each calculation period is obtained from:

$$Q_h = Q_l - \eta Q_g , \tag{1}$$

where Q_h is energy to be delivered to the heating system to satisfy the heat use [MJ], Q_l is heat loss by transmission and by ventilation [MJ], Q_g is heat gain by internal heat gains and solar heat gains [MJ] and η is utilization factor for the heat gain [dimensionless].

Step–by–step calculation procedure for the room space is listed below (1–11):

1) Define the boundaries of heated space

Heated space 163.4 m^3 has one exterior wall with 15 m^2 glazed window, west orientated, without shades. Other walls, floor and ceiling are adjacent to another heated space.

2) Input data

The input data required for the calculation are building type, calculation period, occupancy period, location data, climatic data, geometry, zoning, characteristics of building envelope and constructional complexes, special elements (sunspace, opaque elements with transparent insulation, ventilated solar walls–Trombe walls, ventilated envelope elements, opaque envelope elements) and characteristics of mechanical systems. Energy use for heating could be calculated for residential or non–residential building type. The most used calculation periods are week, month or heating season. Calculation could be done for the whole heating season or per month and for occupied or non–occupied period of time. On–line climatic data that downloaded from RS–MOP (Climate data ..., 2009) for selected location are projective air temperature, degree days, heating season and global sun radiation. After defining basic parameters related to building type and location characteristics, input parameters for zones have to be defined. First, zone type has to be selected (heated or non–heated zone), then geometry (internal volume of heated space, heated floor area) and thermal characteristics of a zone (effective thermal capacity, set–point interior temperature, average internal heat gains, thermal bridges). In every zone also mechanical systems has to be considered (characteristics of ventilation system and efficiency of heating system, characteristics of sun collectors). Important part presents characteristics of transparent and non−transparent constructional complexes that are related with the area of the building element surface, *U–value*, total solar energy transmittance of a building element, frame factor, shading correction factors, curtain factor, area of opening, linear heat bridges. In the program we could use special elements.

3) Calculate the heat loss coefficient of heated space

Heat loss coefficients are calculated for every zone separately. They present the sum of transmission heat loss coefficient, ventilation heat loss coefficient and heat loss with special elements. For the calculation of transmissive heat losses, direct heat transfer coefficient by transmission to the external environment, heat transfer coefficient by transmission to the ground, heat transfer coefficient by transmission through unconditioned spaces, heat transfer coefficient by transmission to adjacent buildings are considered. Ventilation heat loss coefficient depends upon ventilation air volume and heat capacity of the air per volume. If we have heat exchangers, losses are diminished regarding efficiency factor.

4) Calculate the equivalent indoor temperature

The equivalent internal indoor temperature is equal to the normal set point temperature (room air temperature).

5) Calculate the heat losses, Q_l

Heat losses present the sum of all losses to exterior environment and losses to the other zones. They are calculated from internal temperature, external temperature, calculation period and heat loss coefficient of the building. Heat transmission losses are expressed as rates [W].

6) Calculate the internal heat gains, Q_i

Internal heat gains include metabolic heat from occupants and dissipated heat from appliances, heat dissipated from lighting devices, heat dissipated from, or absorbed by, hot and mains water and sewage systems, heat dissipated from, or absorbed by, heating, cooling and ventilation systems, heat from or to processes and goods.

7) Calculate the solar heat gains, Q_s

Solar heat gains result from the solar radiation normally available in the locality concerned, the orientation of the collecting areas, the permanent and movable shading, the solar transmittance and thermal heat transfer characteristics of collecting areas.

8) Calculate utilization factor for the total heat gains

The heat gains are multiplied by utilization factor. Utilization factor is a function of the heat–balance ratio and a numerical parameter that depends on the building inertia.

9) Calculate energy use for heating or delivered energy to heating system to satisfy the heat use, Q_h

The heat losses, Q_l and heat gains, Q_g are calculated for each calculation period. The space heating for each calculation period is obtained from Eq. (1).

10) Calculate annual energy use for heating

The annual energy use of heating is the sum over all months with positive heat requirements, i.e. months for which the average external temperature is lower than the set–point temperature (per heating season).

11) Calculate average energy demand for the heating of our test room

The average energy demand for the heating of our test room is calculated from annual energy use for heating [MJ/a] and presented in W.

ANNEX E.

ANALYSIS OF REAL–TIME CONDITIONS

Table 46: *List of references for physical factors*

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ANNEX F.

CALCULATION OF EXERGY CONSUMPTION RATE FOR SPACE HEATING

The procedure of exergy calculation for space heating was first described by Shukuya and Hammache (2002). The procedure will be presented in this annex on an example of three cases in the Continental part. The main contribution of this calculation was that also location characteristics, different *U–values* and the number of air changes taken from the test room characteristics, where Slovenian regulation demands were well considered.

A 163.4 m^3 room with one exterior wall having a 15 m^2 glazed window is analyzed. Other walls are supposed to be interior. For the exterior window and exterior wall *U–values* are 1.29 and 2.9 W/(m^2K) for Case 1; 0.28 and 1.14 for Cases 2 and 0.28 and 1.14 for Case 3. The number of air changes due to infiltration is 1 h^{-1} for Case 1; and 0.7 h^{-1} for Cases 2, 3, 2/1 and 3/1. The room air temperature is ideally controlled and kept constant at 293 K (20 °C) in all cases, while the outdoor air temperature is assumed to be constant at 281.6 (8.4 °C) for Koper, 278.65 K (5.5 °C) for Ljubljana and 278.0 K (4.9 °C) for Rateče. Outlet air temperature, inlet and outlet water temperatures of the heat exchanger are assumed to be 303 K (30 °C), 343 K (70 °C), and 333 K (60 °C), respectively, for all cases. The electric power supplied to a fan and a pump is 30 W and 23 W in Case 1; 16 W and 12 W in Cases 2 and 3. The ratio of the chemical exergy to the higher heating value of liquidified natural gas (LNG_{ch}) is 0.94. The thermal efficiency of the power plant, i.e. the ratio of produced electricity to the higher heating value of liquidified natural gas (LNG_{el}) supplied, is 0.35.

Regarding step–by–step procedure described by Shukuya and Hammache (2002), the space heating exergy demand using the window and wall areas, their corresponding *U–values*, the number of air changes, and the exterior–interior temperature difference shown in Table 22 were calculated. For Case 1 in the Continental part, the space heating exergy demand, q [W], turns out to be as follows:

$$q = \{\ (15\,m^2\,2.9\,W/(m^2K) + 15m^2\,1.29\,W/(m^2K))\ (293.15\,K - 278.65\,K)\ \} +$$
$$+ \left\{ 1005\,J/(kgK)1.2\,kg/m^3\,\frac{1.0}{3600s}\,163.4\,m^3\,(293.15\,K - 278.65\,K) \right\} = 1705\,J/s = 1705\,W$$

(1)

In the above equation, 1005 is the specific heat of air in the unit of J/(kgK), 1.2 is the density of air in kg/m^3, and 1.0/3600 is the number of air changes in s^{-1}. For Case 2, q = 865 W, Case 3, q = 836 W in the Continental part.

Space heating exergy load for the room space in question is given by

$$X_{rm} = 1 - \frac{T_{ao}}{T_{ai}} q\,, \tag{2}$$

Since T_{ao} is assumed to be in the Continental part 279 K (5.5 °C) and T_{ai} 293 K (20 °C), X_{rm} turns out to be 84 W for Case 1, 43 W for Case 2 and 41 W for Case 3 in the Continental part. In Figure 36, these two values are indicated as the input to "building envelope", which is actually the output from the "room air".

Next, the mass flow rate of air transmitting through the water–to–air heat exchanger was determined from the following equation.

$$\dot{m}_{ar} = \frac{q}{c_{air}(T_h - T_{ai})}\,, \tag{3}$$

where $\dot{m}_{ar}$ is mass flow rate of air [kg/s], c_{air} is specific heat of air [J/(kgK)] (=1005), T_h is outlet air temperature of the heat exchanger and T_{ai} is room air temperature. Assuming that T_h is 303 K (30 °C), the amount of $\dot{m}_{ar}$ turns out to be 0.172 kg/s for Case 1, 0.087 kg/s for Case 2 and 0.084 kg/s for Case 3 in the Continental part. The next step is to calculate the rate of net "warm" exergy delivered by the air circulated through the water–to–air heat exchanger, which is the difference between the "warm" exergies calculated from Eq.(4) substituting $\dot{m}_{ar}$ for m_{ra}, T_h for T_h in the case of inlet air and T_{ai} for T_h in the case of outlet air.

$$X_{ra} = c_{air} m_{ra} \left(\left(T_h - T_{ao}\right) - T_{ao} \ln \frac{T_h}{T_{ao}} \right), \tag{4}$$

where X_{ra} is thermal exergy contained by a volume of room air [kJ].

The rate of net "warm" exergy delivered by air, $\dot{X}_{ra}$ is 111 W for Case 1, 56 W for Case 2 and 54 W for Case 3 in the Continental part. These three values are the input thermal exergy to "room air".

The fan equipped with the water–to–air heat exchanger delivers these thermal exergies. The rate of exergy requirement of the fan is assumed to be 30 W for Case 1 and 16 W for Case 2 and 3 in the Continental part; this corresponds to the condition that the fan efficiency is 0.6 and the pressure decrease between the air inlet and outlet of the heat exchanger is about 100 Pa. Then, the rate of chemical exergy to produce the electricity for the fan becomes 81 W (= 30 (0.94/0.35)) for Case 1 and 43 W (= 16 (0.94/0.35)) for Cases 2 and 3. Here, the number 0.94 is the ratio of chemical exergy to the higher heating value of liquidified natural gas (LNG_{ch}), and the number 0.35 is thermal efficiency of the LNG–fired power plant. The rate of chemical exergy of 81 W or 43 W required by the fan is comparable to the amounts of space heating exergy load, 84 W, 43 W or 41 W. This suggests that it is also very important to design a system with as short duct length and low mass flow rate of air as possible.

The next step is to do exergy calculation in terms of water circulation. Here, the inlet and outlet water temperatures of the water–to–air heat exchanger were assumed to be 343 K (70 °C) and 333 K (60 °C), respectively. The mass flow rate of water is determined as it was done for the air flow rate using Eq. (3), substituting the specific heat of water, 4186 J/(kgK), for c_{air} and the inlet water temperature, 343 K (70 °C) for T_h and the outlet water temperature, 333 K (60 °C) for T_{ai}. The values of mass flow rate obtained are 0.041 kg/s for Case 1, 0.021 kg/s for Case 2 and 0.020 kg/s for Case 3. Then the warm exergy contained by water can be calculated from Eq. (4), substituting 4186 J/(kgK) for c_{air}, the mass flow rate of water for m_{ra}, and the inlet water temperature of 343 K for T_h in case of inlet water and the outlet water temperature of 333 K for T_h in case of outlet water. The rate of net "warm" exergy delivered by water circulation is the difference in thermal exergy contained by water between the inlet and the outlet; it becomes 299 W for Case 1, 152 W for Case 2 and

147 W for Case 3. In Figure 36, these are indicated as the input exergy to "heat exchanger".

A pump circulates the water as a working fluid to deliver thermal exergy from the boiler to the water–to–air heat exchanger. The rate of exergy requirement for the pump is assumed to be 23 W for Case 1 and 12 W for Cases 2 and 3; this corresponds to the condition that the pump efficiency is 0.7 and the pressure decrease within the pipe is about 300 kPa.

Then, the rate of chemical exergy to produce the electricity for the pump becomes 62 W (= 23 (0.94/0.35)) for Case 1 and 32 W (= 12 (0.94/0.35)) for Case 2. The rate of chemical exergy for the pump is smaller than that for the fan, but it is still comparable to the space heating exergy load, especially of Case 2 and Case 3. It is also important to design a system with as short pipe length and low mass flow rate of water as possible.

The rate of chemical exergy supplied to the boiler is determined by the following equation.

$$\dot{X}_{LNB} = \frac{\alpha_q \, q}{\eta_b}, \tag{5}$$

where α_q is the ratio of chemical exergy to the higher heating value of LNG_{ch} (= 0.94), and η_b is the thermal efficiency of the boiler (= 0.8). The value of $\dot{X}_{LNB}$ turns out to be 2003 W for Case 1, 1016 W for Case 2 and 983 W for Case 3 in the Continental part. In Figure 36, these are indicated as the rate of input exergy to "boiler". If the boiler efficiency is improved from 0.8 to 0.95, $\dot{X}_{LNB}$ turns out to be 855 W for Case 2/1; this is indicated, in Figure 36, as the starting point of the crossed line below the line of Case 2. The value of $\dot{X}_{LNB}$ with the boiler efficiency of 0.95 together with the improved thermal insulation of building envelope becomes 827 W; this is indicated as the exergy input to the boiler for Case 3/1.

ANNEX G.

HEAT FLOW RATE AND EXERGY FLOW RATE

Calculation of heat flow rate and exergy flow rate through building wall and calculation of radiant exergy flow rate available from interior wall surfaces followed the detailed calculation procedure described by Shukuya (1994), Shukuya and Hammache (2002), published in IEA ECBCS Annex 37 (Guidebook to IEA ECBCS Annex 37 …, 2006) and IEA ECBCS Annex 49 (Guidebook to IEA ECBCS Annex 49 …, 2011).

Under steady–state condition there is no thermal storage within the wall. Energy balance in the rate form can be described as follows.

$$\dot{q}_{in} = \dot{q}_{out} , \tag{1}$$

$\dot{q}_{in}$ is heat flow rate into the wall [W/m^2] across the boundary surface, where the temperature is T_{ai} [K]. $\dot{q}_{out}$ is heat flow rate out of the wall into the outdoor space [W/m^2]. Since the steady–state condition was assumed here, $\dot{q}_{in}$ can be calculated as follows.

$$\dot{q}_{in} = U\left(T_{ai} - T_{ao}\right), \tag{2}$$

where U is overall heat transmission coefficient of the wall [W/(m^{2}K)], T_{ai} is room air temperature [K], T_{ao} is outdoor air temperature [K].

The entropy balance equation in the rate form corresponding to Eq. (1) is as follows.

$$\frac{\dot{q}_{in}}{T_{ai}} + \dot{S}_{gen,bw} = \frac{\dot{q}_{out}}{T_{ao}} , \tag{3}$$

$\frac{\dot{q}_{in}}{T_{ai}}$ is the entropy flow rate into the wall. It is important to use the boundary–surface temperature for setting up the entropy flow rate across the boundary surface. $\frac{\dot{q}_{out}}{T_{ao}}$ is the entropy flow rate out of the wall. $\dot{S}_{gen,bw}$ is entropy generation rate for the building wall [W/(m^{2}K)].

Heat flow is thermal energy transfer by diffusion, which is quantified explicitly by entropy generation. Since the entropy has the unit of energy divided by the absolute temperature, we may rewrite Eq. (3) by multiplying the outdoor temperature, as follows.

$$\frac{T_{ao}}{T_{ai}}\dot{q}_{in} + \dot{S}_{gen,bw}\,T_{ao} = \dot{q}_{out}\,, \tag{4}$$

This is energy balance equation in the rate form. Anergy is energy already dispersed. Subtraction of Eq. (4) from Eq. (1) results in the exergy balance equation in the rate form, which is exactly the energy not yet dispersed. This is exergy.

$$\left(1-\frac{T_{ao}}{T_{ai}}\right)q_{in} - \dot{S}_{gen,bw}\,T_{ao} = 0\,, \tag{5}$$

We read this equation as follows: $\left(1-\frac{T_{ao}}{T_{ai}}\right)\dot{q}_{in}$ is exergy flow rate into the wall surface and it is totally consumed until the exergy flow rate reaches the outside boundary surface. This is why the right side of Eq. (5) is 0.

Substituting Eq. (6) into Eq. (5),

$$\dot{S}_{gen,bw}\,T_{ao} = \left(1-\frac{T_{ao}}{T_{ai}}\right)\dot{q}_{in} = \frac{T_{ai}-T_{ao}}{T_{ai}}U\left(T_{ai}-T_{ao}\right) = \frac{U\left(T_{ai}-T_{ao}\right)^2}{T_{ai}} > 0\,, \tag{6}$$

It is important to recognize that exergy flow rate and also exergy consumption rate are always positive as indicated by Eq. (6).

In a winter case, i.e. $T_{ao} < T_{ai}$, $\left(1-\frac{T_{ao}}{T_{ai}}\right)\dot{q}_{in}$ is warm exergy flow rate.

ANNEX H.

THERMAL RADIANT EXERGY FLOW RATE

Thermal radiant exergy flow rate from a unit area of building wall surfaces $\dot{X}_r$, were unit is W/m^2, can be calculated from the following Eq. (1) by Shukuya and Hammache (2002).

$$\dot{X}_r = \varepsilon h_{rb} \frac{\left(T_{surf} - T_{ao}\right)^2}{\left(T_{surf} + T_{ao}\right)}, \tag{1}$$

where ε is the overall emittance of the surface, which is usually higher than 0.9 in the case of building envelopes, h_{rb} is the radiative heat transfer coefficient of black–body surface [W/(m^{2}K)], which is given as a constant for built–environmental calculation. T_{surf} and T_{ao} are the surface temperature and environmental temperature in Kelvin, respectively.

When T_{surf} is higher than T_{ao}, the value of $\dot{X}_r$ is the rate of "warm" radiant exergy; on the other hand, when T_{surf} is lower than T_{ao}, it is the rate of "cool" radiant exergy.

Surface temperature was calculated from Eq. (2).

$$T_{surf} = T_{ai} - \left[\frac{\frac{1}{\alpha_{in}}}{R_{wall}} \left(T_{ai} - T_{ao}\right) \right], \tag{2}$$

where T_{ai} is room temperature in Kelvin, T_{ao} is environmental temperature in Kelvin, α_{in} is heat transfer coefficient over the interior surface of the wall in W/(m^{2}K), and R_{wall} is overall thermal resistance of the wall in m^{2}K/W.

ANNEX I.

INDIVIDUAL CHARACTERISTICS OF SUBJECTS WITH REFERENCES

Table 50: *Individual characteristics with references*

Parameter	***Burn patient***	***Visitor, healthcare worker***
T_{sk}	*"…the body tries to raise the skin and core temperature by 2 °C secondary to a hypothalamic reset…"(Herndon, 1996: 422)* *"…burn patient strives for temperatures of about 38 °C…" (Herndon, 1996: 492)*	*"…after 3 hours in a hot room (50 °C), skin temperature differentials amounted to only 2.5 °C (= 35 °C to 37.5 °C), with an average core/surface gradient of ~1 °C. With normal clothing in a room at 15 – 20 °C, mean skin temperature is 32 – 35 °C…" (LeDuc et al., 2002)*
T_{cr}	*"…in patients their core body temperature declines below 35.5 °C…"(Corralo et al., 2007)* *"…in the general surgical population, approximately one–half of patients in routine peri–operative thermal care develop a core body temperature of less than 36 °C during the peri–operative period, and a further one–third exhibit core temperatures of less than 35 °C…" (Ramos et al., 2002)* *"…burn patients are by far the most susceptible to intra– and post–operative hypothermia, since the damaged skin is no longer able to prevent the loss of body heat…" (Ramos et al., 2002)* *"…core temperature is generally expected to be 0.5 °C higher than body surface temperatures…" (Herndon, 1996: 530)*	*"… the normal range for body temperature is 36.1 to 37.8 °C … " (Simmers, 1988:150; Sund–Levander et al., 2002)*

to be continued…

continuation…

Parameter	***Burn patient***	***Visitor, healthcare worker***
T_{cr}	*"...in normal individuals the threshold range is generally near 36.5 °C–37.5 °C. In patient the threshold set point is higher and the increase is proportional to the size of the burn, 0.003 °C / % total body size area regarding the size of the burns…" (Herndon, 1996:205)* *"... hypothermia of less than 35 °C occurs in 89 % of the total operations performed in extensively burned patients…" (Shiozaki et al., 1993)* *"...hypothermia is a particular hazard in children, with their relatively larger surface area, and in all patients with extensive burns (Herndon, 1996: 94)*	
Metabolic rate	*"…numerous recent reports using indirect calorimetry document metabolic rates, which are 120–150 % of normal…"(Herndon, 1996: 399)* *"…increased metabolic rate takes place after thermal injury. Within the range of 70–80 % of TBSA burn injury the hypermetabolism tends to be proportional to the size of burn wound…" (Herndon, 1996: 205)* *"…using indirect calorimetry in acute patient with major burn injuries that are treated according to current standards, resting energy expenditures 110–150 % above predicted values are frequently measured…" (Herndon ,1996: 205)* *"REE in adults might be 200–300 % greater than predicted basal values" (Wilmore et al., 1974b)*	*"standing relaxed 1 met, standing under stress 2 met"* *(ISO 7730; ISO 8996; Fanger, 1970; ASHRAE Handbook & product …, 1977; Skoog et al., 2005)*

to be continued…

continuation…

Parameter	***Burn patient***	***Visitor, healthcare worker***
Metabolism	*"Metabolic rate is increased after burn injury up to about 150 % of normal levels when burn size is greater than 20–30 % TBSA" (Wilmore et al., 1974b)* *"The increase in metabolic rate approaches twice the normal" (Herndon et al., 1987b)* *"...metabolic rate was increased by a factor of 1.5 times basal metabolic rate..." (Cope et al., 1953)* *"...measured energy expenditure reached 2.7±0.9 times the basal energy expenditure in extensively burned patients with hypothermia of less than 35 °C…" (Shiozaki et al., 1993)*	
Effective clothing insulation	*"…naked 0 clo…" (Fanger, 1970)* *"…artificial skin on very large burns covered over 80 % of TBSA… " (Herndon, 1996: 6)*	*"… the insulation of different sets varies within the range of 0.54 ±0.01 clo to 0.95 ±0.01 clo…" (Sudoł–Szopińska and Tarnowski, 2007: 40–46; ASHRAE Handbook & product …, 1977; Skoog et al., 2005)*
T_{ai} RH_{in}	*"…ambient temperature and humidity should be maintained at 30–33 °C and 80 %, respectively, in order to decrease energy demands and evaporative heat losses…" (Herndon, 1996: 492)*	*ANSI/ASHRAE Standard 55 recommends that the relative humidity in occupied spaces is controlled in the ranges from 30 % up to 60 % and at air temperatures between 20 °C and 25 °C.*

to be continued…

continuation…

Parameter	***Burn patient***	***Visitor, healthcare worker***
T_{ai} RH_{in}	*"…the hypermetabolic response may be reduced by warming the ambient temperature to thermal neutrality (33 °C), at which point the heat for evaporation is derived from the environment, taking the burden away from the patient…" (Herndon, 1996: 425)* *"… patients need a hot environment and high relative humidity. A ward for severe burn victims should have temperature controls that permit adjusting the room temperature up to 32 °C db and relative humidity up to 95 %..." (ASHRAE Handbook HVAC applications …, 2007)* *"Patients can be treated at ambient temperatures of 32–35 °C in the intensive care room with a specially designed airflow system..." (Martin et al., 1992)*	

ANNEX J. ADDITIONAL RESULTS OF HUMAN BODY EXERGY CONSUMPTION RATE

Table 51: *Input and output data for the analysis of human body exergy balance for average test subjects. The exergy values are expressed as rates, W/m²*

Influence factor	Subject	Case	Clo material	Moisture permeability	v_a [m/s]	Effective clothing insulation [clo]	T_{ai} [1] [°C]	T_{mr} [1] [°C]	RH_{in} [1] [%]	Metabolic rate [met]	Room [m^3]	PMV index	T_{cr}, T_{sk}, T_{cl} [°C]	RH_{sk} [ ]	Cool (C), Warm (W) radiant exergy in [W/m²]	Cool (C),Warm (W) convective exergy in [W/m²]	Breath air in [W/m²]	From inner part [W/m²]	Exergy consumption [W/m²]	Stored exergy [W/m²]	Exhalation, sweat out [W/m²]	Cool (C), Warm (W) radiation out [W/m²]	Warm convection out [W/m²]	Cool convection out [W/m²]
Effectice clothing insulation, RH_{in}, T_{ai} (Toftum et al., 1998), calculated T_{cr}, T_{sk}, T_{cl}	*N=20 men+20 women 22.8±2.6 years, 1.89±0.14 m² surface body area; 70 kg, 180 cm*	*1 average*	*Cotton+Gore −Tex*	*0.40*	*0.1*	*0.63*	*25.5*	*25.5*	*50*	*1*	*Length: 4.8 m; Width: 3.6 m; High: 2.6 m*	*0.5*	*36.8 34.1 30.3*	*0.32*	*C=0 W=0*	*C=0 W=0*	*0*	*2.862*	*2.261*	*0.001*	*0.176*	*C=0 W= 0.177*	*0.247*	*0*
		2 average	*Cotton+Micro fiber*	*0.32*		*0.90*	*25.0*	*25.0*	*80*			*1.4*	*36.8 34.3 29.3*	*0.59*	*C=0 W=0*	*C=0 W=0*	*0*	*2.499*	*2.062*	*0.001*	*0.094*	*C=0 W= 0.142*	*0.199*	*0*
		3 average	*Cotton+PU nylon*	*0.12*		*0.89*	*25.0*	*25.0*	*50*			*0.9*	*36.8 34.2 29.3*	*0.56*	*C=0 W=0*	*C=0 W=0*	*0*	*3.073*	*2.548*	*0.001*	*0.184*	*C=0 W= 0.141*	*0.198*	*0*
		4 average	*Cotton+PU nylon*	*0.12*		*0.89*	*25.0*	*25.0*	*80*			*1.4*	*36.8 34.2 29.3*	*0.56*	*C=0 W=0*	*C=0 W=0*	*0*	*2.497*	*2.056*	*0.001*	*0.094*	*C=0 W= 0.143*	*0.202*	*0*
		5 average	*Cotton+PU nylon*	*0.12*		*0.89*	*25.5*	*25.5*	*80*			*1.4*	*36.8 34.2 29.3*	*0.69*	*C=0 W=0*	*C=0 W=0*	*0*	*2.424*	*2.022*	*0.001*	*0.088*	*C=0 W= 0.131*	*0.183*	*0*

[1] For our calculations T_{ai} was assumed to be equal to T_{ao}, T_{ai} was assumed to be equal to T_{mr}, RH_{in} was assumed to be equal to RH_{out}.

Table 52: *Input and output data for the analysis of human body exergy consumption rate and thermal comfort conditions for average test subjects. The exergy values are expressed as rates, W/m²*

Influence factor	**Subject**	**Case**	**Clo material**	**Moisture permeability**	**v_a [m/s]**	**Effective clothing insulation [clo]**	**T_{ai} [1] [°C]**	**T_{mr} [1] [°C]**	**RH_{in} [1] [%]**	**Metabolic rate [met]**	**Room [m³]**	**PMV index**	**T_{cr}, T_{sk}, T_{cl} [°C]**	**Cool (C), Warm (W) radiant exergy in [W/m²]**	**Cool (C), Warm (W) convective exergy in [W/m²]**	**Breath air in [W/m²]**	**From inner part [W/m²]**	**Exergy consumption r [W/m²]**	**Stored exergy [W/m²]**	**Exhalation, sweat out [W/m²]**	**Cool (C), Warm (W) radiation out [W/m²]**	**Warm convection out [W/m²]**	**Cool convection out [W/m²]**
T_{ai}=T_{mr}: 15–35 °C, RH_{in}: 30–96 %, calculated T_{cr}, T_{sk}, T_{cl}	N=20 men+20 women 22.8±2.6 years, 1.89±0.14 m² surface body area; 70 kg, 180 cm	1 average	Cotton+Gore–Tex	0.40	0.1	0.63	15.0	15.0	30.0	1	Length: 4.8 m ; Width: 3.6 m; High: 2.6 m	–1.4	36.8 29.9 23.5	W=0 C=0	W=0 C=0	0	6.070	4.230	0	0.518	C=0 W= 0.524	0.800	0
		2 average					35.0	35.0	80.0			3.0	37.1 36.6 35.9	W=0 C=0	W=0 C=0	0	0.928	0.832	0.072	0.009	C=0 W= 0.0065	0.0084	0
		3 average					35.0	35.0	96.0			3.0	37.5 37.2 36.2	W=0 C=0	W=0 C=0	0	0.521	0.288	0.204	0.002	C=0 W= 0.012	0.015	0
		4 average					35.0	35.0	30.0			3.0	36.9 35.5 35.3	W=0 C=0	W=0 C=0	0	4.295	4.137	0.001	0.155	C=0 W= 0.000732	0.001	0
		5 average					15.0	15.0	80.0			–1.3	36.8 30.0 23.5	W=0 C=0	W=0 C=0	0	5.244	3.634	0	0.261	C=0 W= 0.536	0.812	0
		6 average					15.0	15.0	96.0			–1.3	36.8 30.1 23.6	W=0 C=0	W=0 C=0	0	5.103	3.526	0	0.222	C=0 W= 0.539	0.816	0

[1] For our calculations T_{ai} was assumed to be equal to T_{ao}, T_{ai} was assumed to be equal to T_{mr}, RH_{in} was assumed to be equal to RH_{out}.

Table 53: *Input and output data for the analysis of individual human body exergy consumption rate and PMV index at normal room conditions, healthcare worker. The exergy values are expressed as rates, W/m²*

Subject	***T_{ai} [1] [°C]***	***T_{mr} [1] [°C]***	***RH_{in} [1] [%]***	***v_a [m/s]***	***Effective clothing insulation [clo]***	***Metabolic rate [met]***	***PMV index***	***T_{cr} [°C]***	***T_{sk} [°C]***	***T_{cl} [°C]***	***Cool (C), Warm (W) radiant exergy in [W/m²]***	***Cool (C), Warm (W) convective exergy in [W/m²]***	***Breath air in [W/m²]***	***From inner part [W/m²]***	***Exergy consumption [W/m²]***	***Stored exergy [W/m²]***	***Exhalation, sweat out [W/m²]***	***Cool (C), Warm (W) radiation out [W/m²]***	***Cool (C), Warm convection out [W/m²]***
Health. worker	*22.5*	*22.5*	*60*	*0.1*	*0.6*	*1.1*	*− 0.2*	*36.8*	*33.4*	*28.6*	*C=0 W=0*	*C=0 W=0*	*0*	*3.319*	*2.380*	*0.00004*	*0.204*	*C=0 W=0.287*	*C=0 W=0.449*
	18.0	*27.0*	*60*	*0.1*	*0.6*	*1.1*	*− 0.1*	*36.8*	*33.5*	*29.0*	*C=0 W=0.608*	*C=0 W=0*	*0*	*4.280*	*2.234*	*0.004*	*0.295*	*C=0 W=0.910*	*C=0 W=1.445*
	27.0	*18.0*	*60*	*0.1*	*0.6*	*1.1*	*− 0.2*	*36.8*	*33.3*	*28.4*	*C=0.608 W=0*	*C=0 W=0*	*0*	*2.342*	*2.784*	*0.00003*	*0.129*	*C=0 W=0.015*	*C=0 W=0.023*

[1] For our calculations T_{ai} was assumed to be equal to T_{ao}, T_{ai} was assumed to be equal to T_{mr}, RH_{in} was assumed to be equal to RH_{out}.

Table 54: *Input and output data for the analysis of individual human body exergy consumption rate and PMV index at normal room conditions, burn patient. The exergy values are expressed as rates, W/m²*

Subject	T_{ai} [1] [°C]	T_{mr} [1] [°C]	RH_{in} [1] [%]	v_a [m/s]	*Effective clothing insulation [clo]*	*Metabolic rate [met]*	*PMV index*	T_{cr} [°C]	T_{sk} [°C]	T_{cl} [°C]	*Cool (C), Warm (W) radiant exergy in [W/m²]*	*Cool (C), Warm (W) convective exergy in [W/m²]*	*Breath air in [W/m²]*	*From inner part [W/m²]*	*Exergy consumption [W/m²]*	*Stored exergy [W/m²]*	*Exhalation, sweat out [W/m²]*	*Cool (C), Warm (W) radiation out [W/m²]*	*Cool (C), Warm convection out [W/m²]*
Burn patient	*22.5*	*22.5*	*60*	*0.1*	*0*	*2*	*— 1.0*	*35.5*	*37.0*	*30.2*	*C=0 W=0*	*C=0 W=0*	*0*	*5.704*	*3.182*	*0*	*0.359*	*C=0 W=0.569*	*C=0 W=1.592*
	18.0	*27.0*	*60*	*0.1*	*0*	*2*	*— 1.3*	*35.5*	*37.0*	*30.5*	*C=0 W=0.608*	*C=0 W=0*	*0*	*7.523*	*3.233*	*0*	*0.522*	*C=0 W=1.135*	*C=0 W=3.241*
	27.0	*18.0*	*60*	*0.1*	*0*	*2*	*— 0.8*	*35.5*	*37.0*	*32.6*	*C=0.608 W=0*	*C=0 W=0*	*0*	*3.906*	*3.524*	*0.0001*	*0.228*	*C=0 W=0.204*	*C=0 W=0.558*

[1]For our calculations T_{ai} was assumed to be equal to T_{ao}, RH_{in} was assumed to be equal to RH_{out}.

T*able 55:* *Input and output data for the analysis of individual human body exergy consumption rate and PMV index at normal room conditions, visitor. The exergy values are expressed as rates,* W/m^2

Subject	T_{ai}[1] [°C]	T_{mr}[1] [°C]	RH_{in}[1] [%]	v_a [m/s]	Effective clothing insulation [clo]	Metabolic rate [met]	PMV index	T_{cr} [°C]	T_{sk} [°C]	T_{cl} [°C]	Cool (C), Warm (W) radiant exergy in [W/m^2]	Cool (C), Warm (W) convective exergy in [W/m^2]	Breath air in [W/m^2]	From inner part [W/m^2]	Exergy consumption [W/m^2]	Stored exergy [W/m^2]	Exhalation sweat out [W/m^2]	Cool (C), Warm (W) radiation out [W/m^2]	Cool (C), Warm convection out [W/m^2]
Visitor	*22.5*	*22.5*	*60*	*0.1*	*0.6*	*2*	*0.3*	*36.9*	*34.3*	*28.3*	*C=0* *W=0*	*C=0* *W=0*	*0*	*6.858*	*5.499*	*0.004*	*0.373*	*C=0* *W=0.255*	*C=0* *W=0.726*
	18.0	*27.0*	*60*	*0.1*	*0.6*	*2*	*0.1*	*36.9*	*34.1*	*27.8*	*C=0* *W=0.608*	*C=0* *W=0*	*0*	*8.442*	*5.690*	*0.003*	*0.539*	*C=0* *W=0.722*	*C=0* *W=2.096*
	27.0	*18.0*	*60*	*0.1*	*0.6*	*2*	*0.5*	*36.9*	*34.6*	*28.8*	*C=0.608* *W=0*	*C=0* *W=0*	*0*	*5.291*	*5.564*	*0.004*	*0.236*	*C=0* *W=0.003*	*C=0* *W=0.071*

[1] For our calculations T_{ai} was assumed to be equal to T_{ao}, RH_{in} was assumed to be equal to $RH_{out.}$

Table 56: *Input, output data for the analysis of individual human body exergy consumption rate and PMV index at extreme conditions, conventional system. The exergy values are expressed as rates, W/m²*

Influence factor	Subject	Clo material	v_a [m/s]	Effective clothing insulation [clo]	T_{ai} [1] [°C]	T_{mr} [1] [°C]	RH_{in} [1] [%]	Metabolic rate [met]	Room [m³]	PMV index	T_{cr} [°C]	T_{sk} [°C]	T_{cl} [°C]	Cool (C), Warm (W) radiant exergy in [W/m²]	Cool (C), Warm (W) convective exergy in [W/m²]	Breath air in [W/m²]	From inner part [W/m²]	Exergy cons. [W/m²]	Stored exergy [W/m²]	Exhalation, sweat out [W/m²]	Cool (C), Warm (W) radiation out [W/m²]	Warm convection out [W/m²]	Cool convection out [W/m²]
Conventional system: T_{ai}=T_{mr}: 15–35°C, RH_{in}: 30–96 %, constant T_{sk}, T_{cr} for patient, calculated for visitor and healthcare worker	*Healthcare worker*	*Cotton*	0.1	0.6	15.0	15.0	30.0	1.1	163.4	−1.44	36.8	29.5	23.3	C=0 W=0	C=0 W=0	0	6.290	4.376	0	0.563	W= 0.506	0.844	0
					35.0	35.0	80.0			3.00	37.2	36.8	35.9	C=0 W=0	C=0 W=0	0	1.120	0.929	0.081	0.010	W= 0.008	0.011	0
					35.0	35.0	96.0			3.00	37.6	37.3	36.2	C=0 W=0	C=0 W=0	0	0.831	0.323	0.235	0.002	W= 0.013	0.018	0
					35.0	35.0	30.0			2.10	37.0	36.0	35.6	C=0 W=0	C=0 W=0	0	4.564	4.378	0.004	0.171	W= 0.003	0.004	0
					15.0	15.0	80.0			−1.06	36.8	29.7	23.4	C=0 W=0	C=0 W=0	0	5.458	3.787	0	0.287	W= 0.520	0.865	0
					15.0	15.0	96.0			−0.99	36.8	29.8	23.4	C=0 W=0	C=0 W=0	0	5.322	3.683	0	0.244	W= 0.524	0.872	0
	Burn patient	*Naked*	0.1	0	15.0	15.0	30.0	2.0		−1.74	35.5	37.0	25.3	C=0 W=0	C=0 W=0	0	10.823	6.703	0	1.008	W= 0.782	2.330	0
					35.0	35.0	80.0			2.81	35.5	37.0	36.3	C=0 W=0	C=0 W=0	0	1.518	1.443	0.003	0.017	W= 0.014	0.037	0
					35.0	35.0	96.0			3.00	35.5	37.0	37.1	C=0 W=0	C=0 W=0	0	0.399	0.173	0.048	0.002	W= 0.036	0.091	0
					35.0	35.0	30.0			0.24	35.5	37.0	36.2	C=0 W=0	C=0 W=0	0	7.522	7.170	0.001	0.309	W= 0.011	0.029	0
					15.0	15.0	80.0			−1.35	35.5	37.0	25.7	C=0 W=0	C=0 W=0	0	9.107	5.263	0	0.506	W= 0.840	2.498	0
					15.0	15.0	96.0			−1.27	35.5	37.0	25.8	C=0 W=0	C=0 W=0	0	8.839	4.998	0	0.427	W= 0.860	2.554	0
	Visitor	*Cotton*	0.1	0.6	15.0	15.0	30.0	2.0		−0.84	36.9	31.9	23.3	C=0 W=0	C=0 W=0	0	9.798	6.719	0	1.025	W= 0.511	1.543	0
					35.0	35.0	80.0			3.00	38.0	37.3	36.1	C=0 W=0	C=0 W=0	0	2.364	1.596	0.356	0.020	W= 0.010	0.026	0
					35.0	35.0	96.0			3.00	38.6	38.0	36.4	C=0 W=0	C=0 W=0	0	2.089	0.727	0.648	0.006	W= 0.017	0.044	0
					35.0	35.0	30.0			2.14	37.3	36.4	35.6	C=0 W=0	C=0 W=0	0	8.492	8.140	0.311	0.015	W= 0.003	0.009	0
					15.0	15.0	80.0			−0.67	36.9	32.4	23.6	C=0 W=0	C=0 W=0	0	8.491	5.782	0.522	0.0005	W= 0.544	1.641	0
					15.0	15.0	96.0			−0.64	36.9	32.6	23.7	C=0 W=0	C=0 W=0	0	8.285	5.611	0.444	0.0002	W= 0.555	1.673	0

[1] For our calculations T_{ai} was assumed to be equal to T_{ao}, T_{ai} was assumed to be equal to T_{mr}, RH_{in} was assumed to be equal to RH_{out}.

Table 57: *Input and output data for the analysis of individual human body exergy consumption rate and PMV index at extreme conditions, LowEx system. The exergy values are expressed as rates, W/m²*

Influence factor	**Subject**	**Clo material**	**v_a [m/s]**	**Effective clothing insulation [clo]**	**T_{ai} [1] [°C]**	**T_{mr} [1] [°C]**	**RH_{in} [1] [%]**	**Metabolic rate [met]**	**Room [m³]**	**PMV**	**T_{cr} [°C]**	**T_{sk} [°C]**	**T_{cl} [°C]**	**Cool (C),Warm (W) radiant exergy in [W/m²]**	**Breath air in [W/m²]**	**From inner part [W/m²]**	**Exergy consump. [W/m²]**	**Stored exergy [W/m²]**	**Exhalation, sweat out [W/m²]**	**Cool (C), Warm (W) radiation out [W/m²]**	**Warm convection out [W/m²]**	**Cool convection out [W/m²]**
LowEx system: $T_{ai} \neq T_{mr}$, RH_{in}=80 %, constant T_{cr}, T_{sk} for patient, calculated for visitor and healthcare worker	*Healthcare worker*	*Cotton*	*0.1*	*0.6*	*15.0*	*35.0*	*80.0*	*1.1*	*163.4*	*0.35*	*36.9*	*34.5*	*31.1*	*C=0 W= 3.055*	*0*	*4.851*	*2.588*	*0.004*	*0.293*	*C=0 W= 1.943*	*3.073*	*0*
					35.0	*15.0*	*80.0*			*2.60*	*36.9*	*34.3*	*29.7*	*C= 3.055 W=0*	*0*	*0.578*	*3.066*	*0.081*	*0.010*	*W=0 C= 0.223*	*0*	*0.333*
	Burn patient	*Naked*	*0.1*	*0*	*15.0*	*35.0*	*80.0*	*2.0*		*−1.04*	*35.5*	*37.0*	*31.7*	*C=0 W= 3.055*	*0*	*8.155*	*2.552*	*0*	*0.511*	*C=0 W= 2.114*	*6.034*	*0*
					35.0	*15.0*	*80.0*			*1.23*	*35.5*	*37.0*	*34.6*	*C= 3.055 W=0*	*0*	*0.627*	*3.660*	*0.0003*	*0.017*	*C= 0.001 W=0*	*0*	*0.003*
	Visitor	*Cotton*	*0.1*	*0.6*	*15.0*	*35.0*	*80.0*	*2.0*		*0.42*	*36.9*	*35.0*	*29.2*	*C=0 W= 3.055*	*0*	*8.945*	*5.553*	*0.019*	*0.534*	*C=0 W= 1.505*	*4.370*	*0*
					35.0	*15.0*	*80.0*			*3.00*	*37.3*	*36.3*	*31.4*	*C= 3.055 W=0*	*0*	*1.632*	*4.153*	*0.063*	*0.019*	*C= 1.106 W= 0*	*0*	*0.282*

[1] For our calculations T_{ai} was assumed to be equal to T_{ao}, RH_{in} was assumed to be equal to RH_{out}.

ANNEX K. OBJECTIVES OF DOCTORAL DISSERTATION AND CANDIDATE'S CONTRIBUTION

Table 58: *Objectives of doctoral dissertation, time schedule, location, candidate's contribution, performer, supervisor, filed*

No.	Objective	Time schedule, location	Candidate's contribution	Performer	Supervisor, field
1	**To develop a method** for holistic manipulation of problems and find alternative solutions to them.	October 2005–June 2006 UL Faculty of Health Sciences; UL FGG, KSKE	Development of the method for aircraft environment (Dovjak et al., 2006): upgrading of Asimow (1962) methodology of engineering design.	M. Dovjak	Prof. Dr. A. Krainer (method development) M. Sc. K. Likar, Senior Lecturer (method development) Dr. J. Horvat, Senior Lecturer (chemical factors in aircraft environment) Assist. Prof. Dr. M. Bizjak (chemical factors in aircraft environment) Assist. Prof. Dr. A. Kraigher (biological factors in aircraft environment) Prof. Dr. V. Butala (heating, ventilation, air–conditioning systems in aircrafts)
		October 2007–September 2008 UL FGG, KSKE	Transformation of the method into HE, upgrading regarding the specifics of HE.	M. Dovjak	Prof. Dr. A. Krainer (transformation of the method into HE)
		January 2008 – January 2012 UL FGG, KSKE; Department of Civil Engineering, Technical University of Denmark	Execution of the first free steps of feasibility study. 1. Definition of needs, demands and conditions: analysis of real state conditions in the field of physical factors, chemical factors, biological factors, and individualization in HE (721 sources from relevant literature, from 1934 to 2012); creation of databank; comparison of actual conditions with regulated demands for HE; selection of impact factors and matrix construction. 2. Design problem: detection and classification of the problem (high energy use for H/C) 3. Synthesis of the possible solutions: searching for alternative solutions for detached problem in the matrix of physical factors. Alternative solutions: improvement of building envelope together with effective H/C system that enables optimal conditions for individual user with minimal possible energy use for heating and cooling.	M. Dovjak	Burn patients: J. Cepuder, M.D. (needs, demands and conditions for burn patients) R. Mulh, M.D. (needs, demands and conditions for burn patients) S. Dovjak, Working group for prevention of hospital–acquired infections, Community Heath Centre Ljubljana (needs, demands and conditions for healthcare workers; biological factors in HE) Prof. Dr. A. Krainer (detection and classification of the problem, matrix design, alternative solutions) Prof. Dr. M. Shukuya (detection and classification of the problem, matrix design, alternative solutions) Prof. Dr. A. Melikov (detection and classification of the problem, matrix design, alternative solutions)

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No.	*Objective*	*Time schedule, place*	*Candidate's contribution*	*Performer*	*Supervisor, field*
2	***To solve the problem** of high energy use for heating and cooling holistically, using the structure of building envelope and effective H/C system at different locations.*	*September 2008–November 2009* *UL FGG, KSKE; Department of Civil Engineering, Technical University of Denmark*	*Execution of the analysis of exergy consumption patterns for space heating: analysis of building envelope characteristics of the test room at different locations, calculation of energy use, comparison with the results of exergy analysis, definition of the most effective interventions for decreasing energy use.* *The results were published in our publication (Dovjak et al., 2010a, 2010b, 2011e, 2011g).*	*M. Dovjak*	*Prof. Dr. A. Krainer* *(energy analysis, definition of interventions)* *Prof. Dr. Shukuya* *(exergy analysis, definition of interventions)* *Prof. Dr. B. W. Olesen* *(exergy analysis, definition of interventions)* *R. Perdan* *(designer of the software for the calculation of energy use)*
3	***To solve the problem** of high energy use for heating and cooling by keeping at the same time the attained optimal conditions for individual user (required and comfort conditions) and designed (minimal possible) exergy consumption rate inside the human body*	*March 2009* *Department of Civil Engineering, Technical University of Denmark*	*Testing of H. Asada Rev. 2009 (Iwamatsu and Asada, 2009) spread sheet software for the calculation of human body exergy balance.*	*Working group of COST action C24* *M. Dovjak*	*Dr. H. Asada* *(software designer)*
		March 2009–May 2009 *Department of Civil Engineering, Technical University of Denmark*	*Exergy analysis of thermal comfort for average test subject: calculation of hbExCr, PMV for average test subject in experimental conditions (T_{ai} 25.0–25.5 °C, T_{mr} 25.0–25.5 °C, RH_{in} 50 –80 %).* *The results were also published in our publication (Dovjak et al., 2011b).*	*M. Dovjak*	*Working group of COST action C24 Analysis and design of innovative systems with LowEx for application in build environment* *Prof. Dr. Shukuya* *(exergy analysis)* *Prof. Dr. B. W. Olesen* *(exergy analysis)*
		March 2009–May 2009 *UL FGG, KSKE; Department of Civil Engineering, Technical University of Denmark*	*Comparison of human body exergy consumption and thermal sensation vote (TSV).* *The results were published in our publication (Simone et al., 2010, 2011a, 2011b).*	*Working group of COST action C24* *M. Dovjak*	*Working group of COST action C24 Analysis and design of innovative systems with LowEx for application in build environment* *Prof. Dr. M. Shukuya* *(exergy analysis of thermal comfort)* *Prof. Dr. B. W. Olesen* *(energy analysis of thermal comfort)* *Dr. H. Asada* *(software designer)*
		April 2009–May 2009 *UL FGG, KSKE; Department of Civil Engineering, Technical University of Denmark*	*Upgrading the software for the calculation of human body exergy balance of an individual user.*	*Dr. H. Asada* *M. Dovjak*	*Dr. H. Asada* *(software designer)*

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No.	Objective	Time schedule, place	Candidate's contribution	Performer	Supervisor, field
3a	*To compare the two H/C systems (conventional system and LowEx system) from exergetic point of the view. It includes exergy analysis of thermal comfort conditions for individual user and measured energy use for heating and cooling of active space.*	*June 2009–February 2011* *UL FGG, KSKE*	*Execution of exergy analysis of thermal comfort conditions for individual test subject: calculation of hbExCr, PMV for burn patient, healthcare worker, and visitor exposed to normal and extreme conditions created in a room with LowEx and conventional system (T_{ai} 15.0–35.0 °C, T_{mr} 15.0–35.0 °C, RH_{in} 30 –96 %).* *The results were also published in our publication (Dovjak et al., 2010c, 2011a, 2011c, 2011d, 2011f).*	*M. Dovjak*	*Prof. Dr. A. Krainer* *(exergy analysis of individual thermal comfort conditions)* *Prof. Dr. Shukuya* *(exergy analysis of individual thermal comfort conditions)*
		January 2010– February 2010 *UL FGG, KSKE*	*Upgrading the ICSIE system, additional installations and measuring equipment.*	*M. Dovjak*	*R. Perdan* *(upgrading the ICSIE system)* *Prof. Dr. A. Krainer* *(upgrading the ICSIE system)*
		March 2010, June 2010 *June 2010–July 2010* *UL FGG, KSKE*	*Execution of measurements of energy use for H/C of test room for both systems in time separation, control and monitoring of microclimate conditions with ICSIE system.* *The results were also published in our publication (Dovjak et al., 2011a, 2011c, 2011e, 2011f).*	*M. Dovjak*	*R. Perdan* *(measurements, ICSIE system)* *Prof. Dr. A. Krainer* *(measurements)*
		March 2010–September 2010 *UL FGG, KSKE*	*Data processing, system comparison, selection of days with comparable boundary conditions.*	*M. Dovjak*	*R. Perdan* *(data processing)* *Prof. Dr. A. Krainer,* *(system comparison)* *Prof. Dr. M. Shukuya* *(system comparison)*
3b	*To find the connection between optimal conditions (required and comfort conditions) and individual human body exergy consumption rate.*	*June 2010–February 2011* *UL FGG, KSKE*	*System comparison regarding the calculated hbExCr and PMV for individual user. Finding the connection between optimal conditions (required and comfort conditions) and individual human body exergy consumption rate.*	*M. Dovjak*	*Prof. Dr. A. Krainer,* *(system comparison)* *Prof. Dr. Shukuya* *(connection between optimal conditions and individual human body exergy consumption rate)*

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No	*Goal*	*Time schedule, place*	*Candidate's contribution*	*Performer*	*Supervisor, field*
3c	*To find out which H/C system enables at the same time attainment of optimal conditions (required and comfort conditions) for individual user, designed (minimal possible) human body exergy consumption rate and minimal energy use for heating and cooling.*	*June 2010–February 2011* *UL FGG, KSKE*	*System comparison regarding of hbExCr and measured energy use for heating and cooling. Finding which H/C system enables at the same time attainment of optimal conditions for individual user, designed (minimal possible) hbExCr and minimal energy use for heating and cooling.*	*M. Dovjak*	*Prof. Dr. A. Krainer,* *(connection between optimal conditions, designed human body exergy consumption rate and minimal energy use)* *Prof. Dr. Shukuya* *(connection between optimal conditions, designed human body exergy consumption rate and minimal energy use).*
4	***To select the solution** of the H/C system that enables attainment of optimal conditions (required and comfort conditions) for individual user, designed (minimal possible) human body exergy consumption rate and minimal energy use for H/C at the same time.*	*June 2010– February 2011* *UL FGG, KSKE*	*Selection of H/C system that enables attainment of optimal conditions (required and comfort conditions) for individual user, designed (minimal possible) hbExCr and minimal energy use for H/C at the same time.*	*M. Dovjak*	*Prof. Dr. A. Krainer,* *(selection of the system)* *Prof. dr. M. Shukuya* *(selection of the system)*
5	***To enable active regulation** and control of required and thermal comfort conditions for individual user.*	*September 2010– February 2011* *UL FGG, KSKE*	*Presentation of active regulation and control of required and thermal comfort conditions for individual user: individualization of personal space in HE.*	*M. Dovjak*	*Prof. Dr. A. Krainer* *(active regulation)* *Prof. Dr. M. Shukuya* *(active regulation)*

I have only just a minute,
Only 60 seconds in it,
Forced upon me–can’t refuse it
Didn’t seek it, didn’t choose it.
But it is up to me to use it.
I must suffer if it loses it.
Give account if I abuse it,
Just a tiny litter minute–
But eternity is in it.

(Anonymous)

A poetic expression of exergy and exergy destruction.