# ZBORNIK RADOVA / *Proceedings*

# interklima 2007

**19. MEĐUNARODNI SIMPOZIJ I IZLOŽBA O GRIJANJU, HLAĐENJU I KLIMATIZACIJI**
***19. INTERNATIONAL SYMPOSIUM AND EXHIBITION OF HEATING, REFRIGERATING AND AIR CONDITIONING***
**8. KONFERENCIJA O TERMOGRAFIJI / *8. CONFERENCE ON THERMOGRAPHY***

Zagreb, CROATIA, 19. i 20. travnja 2007. / *April 19-20, 2007*




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## 8. KONFERENCIJA O TERMOGRAFIJI

**u sklopu**

**IZLOŽBE GRIJANJA, HLAĐENJA, VENTILACIJE, KLIMATIZACIJE, SANITARIJA, OBRADE PITKIH VODA, TERMOENERGETSKIH POSTROJENJA, KOGENERATIVNIH SUSTAVA**

## 19th INTERNATIONAL SYMPOSIUM AND EXHIBITION OF HEATING, REFRIGERATING AND AIR CONDITIONING

## 8th CONFERENCE ON THERMOGRAPHY

***CROATIA - Zagreb, 19. i 20. travnja 2007.***

***Zagrebački velesajam, Dvorana "Brijuni"***

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## ORGANIZATORI / *Organized by*

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## ZNANSTVENI SAVJET (abecednim redom)

## *Scientific Committee (alphabetical order)*

# Program rada

**četvrtak, 19. travnja 2007.**

| | |
|---|---|
| 8:30 - 9:15 | registracija sudionika |
| 9:15 - 9:30 | otvaranje Simpozija - pozdravi: predstavnici Zagrebačkog velesajma, Fakulteta za strojništvo Univerze v Ljubljani, Fakulteta strojarstva i brodogradnje Sveučilišta u Zagrebu, Ministarstva znanosti, obrazovanja i športa, Grada Zagreba... |

## Znanstveni dio - Dan Frana Bošnjakovića

| | |
|---|---|
| | predsjedavajući: Galović, Novak, Boras |
| 9:30 - 9:45 | uvodno izlaganje:<br>*A. Galović*<br>IN MEMORIAM F. BOŠNJAKOVIĆ |
| 9:45 - 10:05 | *M. Vujanović, N. Duić, A. Galović*<br>THREE-DIMENSIONAL NUMERICAL SIMULATION OF THE NITROGEN OXIDES FORMATION IN AN OIL-FIRED FURNACE |
| 10:05 - 10:25 | *D. Dović, S. Švaić*<br>EXPERIMENTAL AND NUMERICAL STUDY OF THE FLOW AND HEAT TRANSFER IN PLATE HEAT EXCHANGER CHANNELS |
| 10:25 - 10:45 | *A. Can, A. Babadağli*<br>IMPORTANCE OF FORM AND PLACEMENT OF HEATING ELEMENTS FROM THE ASPECT OF COMFORT, ENVIRONMENT, HEALTH AND ECONOMY |
| 10:45 - 11,05 | *A. Can, B. Karaçavuş, E. Buyruk*<br>AN EXPERIMENTAL STUDY ON SOLAR ENERGY STORAGE |
| 11:05 - 11:30 | coffee break |
| 11:30 - 11:50 | *D. Kaya, E. Buyruk, A. Can, A. Fertelli*<br>NUMERICAL STUDY OF HEAT TRANSFER CHARACTERISTICS OF EXTENDED SURFACES |
| 11:50 - 12:10 | *D. Hrastović, V. Soldo*<br>APPLICATION OF CARBON DIOXIDE IN THE TRANSCRITICAL REFRIGERATION CYCLE |
| 12:10 - 12:30 | *I. Boras, S. Švaić, I. Gospić*<br>DETERMINATION OF TEMPERATURE DISTRIBUTION AT ASPHALT CARRIER SHIP STRUCTURE |
| 12:30 - 12:45 | rasprava i završne napomene |
| 12:45 - 14:00 | cocktail |

# Program rada

**četvrtak, 19. travnja 2007.**

**8. konferencija o termografiji**

| | |
|---|---|
| | predsjedavajući*:* Švaić, Mendel, Andrassy |
| 14:00 - 14:15 | uvodno izlaganje<br>*S. Švaić, M. Andrassy*<br>30 YEARS OF IR THERMOGRAPHY AT THE FACULTY OF MECHANICAL ENGINEERING AND NAVAL ARCHITECTURE IN ZAGREB, LABORATORY FOR APPLIED THERMODYNAMICS |
| 14:15 - 14:35 | *F. Mendel*<br>THE AUSTRIAN AND EUROPEAN SOCIETY FOR IR THERMOGRAPHY |
| 14:35 - 14:55 | *Ž. Hrs Borković, M. Zidar*<br>ENERGY AUDITS OF FAMILY HOUSES TO IMPROVE ENERGY EFFICIENCY |
| 14:55 - 15:15 | *J. Aderhold*<br>QUALITY CONTROL OF WOOD-BASED AND COMPOSITE MATERIALS BY INFRARED THERMOGRAPHY |
| 15:15 - 15:45 | coffee break |
| 15:45 - 16:05 | *I. Benkö*<br>TESTING PILOTS IN A VACUUM-CHAMBER FOR EMERGENCY CASES |
| 16:05 - 16:25 | *G. Traxler, W. Palfinger*<br>LIMITATIONS OF 1D HEAT FLOW THERMOGRAPHY ON COMPOUNDS USING FLASH SOURCES |
| 16:25 - 16:45 | *I. Boras, S. Švaić*<br>THE APPLICATION OF INFRARED THERMOGRAPHY IN THERMAL ANALYSIS OF OPEN PLATE HEAT EXCHANGER |
| 16:45 - 17:05 | *S. Švaić, I. Boras*<br>APPLICATION OF THERMOGRAPHY FOR DETERMINATION OF HEAT CONDUCTIVITY OF PREFABRICATED INSULATING PRODUCTS |
| 17:05 - 17:20 | rasprava i završne napomene |

# Program rada

**petak, 20. travnja 2007.**

**Stručni dio**

| | |
|---|---|
| 8:30 - 9:15 | registracija sudionika |
| 9:15 - 9:30 | pozdravni govori |
| 9:30 - 11:00 | *V. Kolega*<br>PRAVILNIK O OZNAČAVANJU ENERGETSKE UČINKOVITOSTI KUĆANSKIH UREĐAJA<br><br>*Z. Poša*<br>TOPLINSKA BILANCA LJUDSKOG TIJELA U VODI<br><br>*K. Osman*<br>AUTOMATSKI STABILNI SUSTAV ZA GAŠENJE POŽARA<br><br>*K. Osman*<br>ISPITIVANJE PROTUUDARNOG VENTILA ZA SKLONIŠTA<br><br>*posebna prezentacija: F. Kritovac*<br>DOBRA KLIMA (kolaž autorskih fotografija o tome kako su i gdje sve smješteni klima-uređaji - zapanjujući primjeri s područja Zagreba snimljeni u razdoblju 2003. - 2007. godine) |
| 11:10 - 11:30 | coffee break |
| 11:30 - 13:30 | OTVORENI NADZORNO-UPRAVLJAČKI SUSTAV ZA REGULACIJU HVAC-INFRASTRUKTURE U KONTEKSTU INTELIGENTNE ZGRADE (BMS)<br>*(Ninoslav KURTALJ, dipl. ing., Elma Kurtalj d.o.o.)*<br><br>IZGRADNJA NOVIH ENERGETSKIH POSTROJENJA TE UPRAVLJANJE I ODRŽAVANJE ENERGETSKOG SUSTAVA U KBC-u ZAGREB - BOLNICA REBRO - NOVI ISKORAK U POSLOVANJU HEP TOPLINARSTVA d.o.o.<br>*(Robert KRKLEC, dipl. ing., direktor HEP Toplinarstvo d.o.o.)*<br><br>PROJEKTI ENERGETSKE UČINKOVITOSTI U BOLNICAMA U IZVEDBI HEP ESCO d.o.o.<br>*(mr. sc. Gordana LUČIĆ, dipl. ing., direktorica HEP ESCO d.o.o.)*<br><br>MODEL IMPLEMENTACIJE MJERA ENERGETSKE UČINKOVITOSTI NA KBC REBRO<br>*(Hrvoje GLAMUZINA, dipl. ing., rukovoditelj Odjela za pripremu izvedbu projekata HEP ESCO d.o.o.)*<br><br>TPK OROMETAL - 50 GODINA S VAMA<br>*(Dragutin DRUŽINEC, dipl. ing., direktor TPK Orometal d.d.)* |
| 13:30 - 15:00 | svečani cocktail suorganizatora u prigodi značajne obljetnice - poziva Vas TPK Orometal |

# Predgovor

**Međunarodni simpozij INTERKLIMA** održava se u sklopu Međunarodnog sajma grijanja, hlađenja, klimatizacije i obrade pitkih voda na Zagrebačkom velesajmu od 1969. godine. Simpozij i sajam su bijenalnog karaktera.

INTERKLIMA 2007 održat će se u dvije cjeline i to:

• 1. dan - znanstveni dio: Dan Frana Bošnjakovića i 8. konferencija o termografiji

• 2. dan - stručni dio s prezentacijama tvrtki.

Radovi za znanstveni dio simpozija pišu se i iznose na engleskom, a za stručni dio na hrvatskom jeziku.

Simpozij INTERKLIMA, kao najstariji simpozij takve vrste u ovom dijelu Europe, postao je tradicionalno mjesto susreta znanstvenika i stručnjaka iz područja klimatizacije, grijanja i hlađenja. To je također mjesto gdje brojni izlagači sa sajma INTERKLIMA izravno mogu predstavljati svoje novosti okupljenim stručnjacima.

Očekujemo da će i ovaj, 19. po redu Međunarodni simpozij o grijanju, hlađenju i klimatizaciji te 8. konferencija o termografiji okupiti ugledne stručnjake koji će tijekom dvodnevnog rada razmijeniti brojna istraživačka i radna iskustva iz tematskih područja simpozija.

Uz sudjelovanje na simpoziju INTERKLIMA sudionici imaju pravo na ulaznicu za jednokratan posjet sajmu INTERKLIMA na Zagrebačkom velesajmu.

# Foreword

***The INTERKLIMA International Symposium*** *has been held in the context of the International Fair of Heating, Refrigeration, Air-conditioning and Potable Water Treatment as an event of the Zagreb Fair since 1969. Both the symposium and the fair have a biennial character.*

*INTERKLIMA 2007 will be composed of two sections:*

*• 1st day - scientific section The Fran Bošnjaković Day and the 8th Conference on Thermography*

*• 2nd day - specialized events including presentation of companies.*

*Papers intended for the scientific section of the symposium will be prepared and presented in English.*

*The INTERKLIMA, as the oldest symposium of the kind in this part of Europe, has grown into a traditional venue for meeting of scientists and experts in the field of air-conditioning, heating and refrigeration. This is also the place where numerous exhibitors may introduce their novelties to attending experts.*

*We expect this, 19th in the row International Symposium on Heating, Refrigeration and Air-conditioning and the 8th Conference on Thermography to gather prominent experts. During two symposium days they will be given the opportunity to exchange numerous research and practical experiences of the thematic areas covered by this event.*

*All participants of the INTERKLIMA Symposium will receive a ticket for a visit to the INTERKLIMA Exhibition at the Zagreb Fair site.*

# Sadržaj

# Sadržaj

# Sadržaj

# Znanstveni dio

## Dan Frana Bošnjakovića

# Three-dimensional numerical simulation of the nitrogen oxides formation in an oil-fired furnace

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## ABSTRACT

In this paper the three dimensional numerical simulation of the nitrogen pollutant formation in real-size oil fired furnace is presented. AVL's finite volume based commercial CFD code FIRE$^{TM}$ was used to simulate turbulent flow field, combustion, nitrogen containing pollutants, radiation and related phenomena by solving the mathematical equations, which govern these processes using a numerical process. The mathematical models of oil combustion, radiation and nitrogen oxides formation was implemented in commercial CFD code FIRE$^{TM}$ in order to predict heat transfer process and nitrogen oxides emissions.

**Keywords: pollution, $NO_x$, combustion, radiation CFD**

## INTRODUCTION

The nitrogen oxides ($NO_x$) emitted from combustion systems are atmospheric pollutants that contribute to photochemical smog, acid rain, global warming and depletion of stratospheric ozone. The global emission of $NO_x$ into the atmosphere has been increasing over the past 50 years. This significant increase is mainly attributed to human activities, particularly motor vehicles, electric power plant boilers and gas turbines, and other industrial, commercial, and residential sources that burn fossil fuels. Stringent regulations have been established to limit the total NO emissions emitted in combustion processes, and the prediction of these pollutants has been a major driving force in combustion modelling.

The generic designation $NO_x$ refers to the summation of all oxides of nitrogen, namely NO, $NO_2$, $N_2O$, $N_2O_2$, $N_2O_3$, $N_2O_4$ and $N_2O_5$. The most damaging of the hazardous nitrogen compounds formed during combustion are nitric oxide (NO), nitrogen dioxide ($NO_2$) and nitrous oxide ($N_2O$). However, combustion sources emit $NO_x$ mostly in the form of nitric oxide representing 90 to 95 percent of the total $NO_x$ emissions. Considering that the most of the $NO_x$ emitted by combustion sources is NO with only a small fraction appearing as $NO_2$, and that other nitrogen oxides are emitted in negligible concentrations, the presence and effects of $NO_2$, $N_2O$, $N_2O_2$, $N_2O_3$, $N_2O_4$ and $N_2O_5$ during combustion processes are ignored in this study.

This paper presents a review of modelling work performed at Department of Energy, Power Engineering and Environment (DEPEE) regarding the prediction of $NO_x$ formation in combustion systems. The modelling work has involved approach that couples a simplified description of the $NO_x$ reaction process with combustion and flow process, taking into account sufficient details to adequately describe $NO_x$ reaction process and to allow coupling with the turbulent mixing process in CFD simulation. AVL's finite volume based commercial CFD code FIRE$^{TM}$ was used to simulate turbulent flow field, combustion, nitrogen containing pollutants, radiation and related phenomena by solving the mathematical equations, which govern these processes using a numerical process. The mathematical model of oil-combustion is semi-empirical, based on a single step irreversible reaction scheme, including the impacts of evaporation, induction, kinetics and coke combustion in an Arrhenius type of expression. The radiation was solved using the discrete transfer radiation method [1, 2], where the radiative properties of participating media were evaluated with the weighted sum of gray gases model [3]. The amount of NO produced in combustion process in oil-fired furnace were calculated in a post-processing mode based on numerical solution of the steady state transport equation for nitric oxide. The source term, which represents the production of NO, was calculated from contributions from two predominant chemical kinetic mechanisms: extended Zeldovich mechanism and Fenimore prompt mechanism [4]. Coupling with the turbulent mixing process was required special consideration when calculating the time-mean kinetic rates for the reaction mechanisms, which was taken into account by means of presumed probability density function of a normalized temperature.

## OIL COMBUSTION MODEL

The oil combustion and release of heat as the consequence of combustion is very complicated process, which depends on one side upon the complex structure of chemical processes for gas-phase combustion and on the other side upon the droplet size and related complex phenomena as atomization and spray combustion. Detailed modelling of the combustion process is therefore time consuming task, and usually a compromise has to be searched for computationally affordable solutions. Hence, the alternative approach of efficient combustion modelling is using Arrhenius type expression for the fuel concentration rate of change with coefficients determined on the basis of experimental data for particular type of fuel. The mathematical model of oil-combustion used in this work is empirical [5], based on experimental data, using data for single droplet combustion. The total time of oil-combustion consists three parts: time of evaporation and induction, time of oxidation and time of coke combustion:

$$\tau = \tau_{ei} + \tau_{ox} + \tau_{cc} \tag{1}$$

The time of evaporation and induction:

$$\tau_{ei} = 9,434 \cdot 10^{-7} \cdot e^{\frac{10^5}{RT}} + 0,45695 \cdot d_0^2 \tag{2}$$

The second stage of oil-combustion is oxidation, time of oxidation:

$$\tau_{ox} = \frac{d_0^2}{0,0032 \cdot (T - 273,15) - 1,79} \tag{3}$$

The time of coke combustion is estimated with regard to time of evaporation and induction and time of oxidation:

$$\tau_{cc} = 0,75 \cdot (\tau_{ei} + \tau_{ox}) \tag{4}$$

The combustion rate k is defined as reciprocal value of total time of oil-combustion:

$$k = \frac{b}{\tau} \tag{5}$$

where $b$ is combustion velocity coefficient used for the purpose of switching on the influence of oxygen diffusion

b = 33,3 if $Y_{O2} < 0{,}03$
b = 1 if $Y_{O2} > 0{,}03$

The reaction rate of oil-combustion depends of fuel mass fraction and to oxygen mass fraction, which can be expressed as:

$$\overline{r_B} = -\rho \cdot \frac{\partial \overline{Y_B}}{\partial t} = -\rho \cdot k \cdot \overline{Y}_B^{\nu_B} \cdot \overline{Y}_{O_2}^{\nu_{O_2}} \tag{6}$$

where it is only sensitive to oxygen mass fraction when it is low, delimited in the model by 3%:

$\nu_B = 1$
$\nu_{O_2} = 1$ if $Y_{O_2} < 0{,}03$
$\nu_{O_2} = 0$ if $Y_{O_2} > 0{,}03$

This oil-combustion model is robust and works well for a wide range of typical industrial oil flames, as tested in previous work [6-9].

**RADIATION MODEL**

For CFD based oil-combustion furnace simulation, radiation is a dominant energy transport mechanisms to surrounding surfaces and correct calculation of the thermal radiation is needed to accurately predict the temperature field and the heat fluxes at walls of the furnaces. This is very important especially when an accurate modelling of $NO_x$ reactions is required, since the formation of nitrogen pollutant species is strongly temperature-dependent. Thus, radiative energy transport is very important element in modelling of combustion systems and in this work for calculating radiation heat transfer the discrete transfer radiation method (DTRM) [2] is used. This method is based on the solution of radiative heat transfer equation for some representative rays fired from the domain boundaries. Rays are fired from the boundary cells into a finite number of solid angles that cover the radiating hemisphere about each cell.

Figure 1 shows example of 2D illustration of the domain subdivided into a finite number of control volumes. An arbitrary ray is shown for boundary face *P*.

The change of radiant intensity leaving point *R* and along the ray until it reaches *P* is tracked, yielding recurrence relation:

$$i'_{n+1} = i'_n (1-\varepsilon) + \varepsilon \frac{\sigma T^4}{\pi} \tag{7}$$

The radiative properties were modelled in this work by using weighted sum of grey gases model (WSGGM) [3].

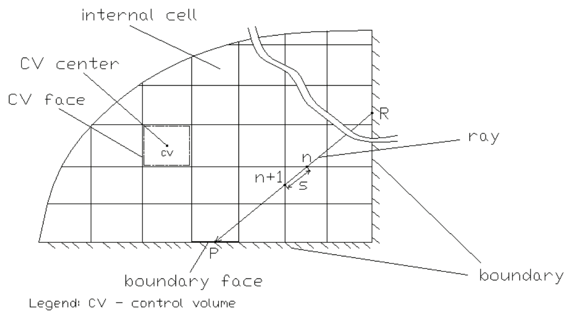


Figure 1. Ray tracing in the DTRM

It is customary to assume that radiant surfaces are gray and obey the Lambert cosine law, leading to the following boundary relation that is needed at the beginning of the incremental ray tracing path in Eq. (8):

$$i'_0 = C_R \frac{q_{out}}{\pi} = \frac{q_{in}}{\pi}(1-\varepsilon_w) + \varepsilon_w \frac{\sigma T_w^4}{\pi} \tag{8}$$

In the original DTRM [2] there is no correction of the outgoing boundary radiation flux, i.e. $C_R$=1, and it is due to Coelho and Carvalho [1] later on that a correction to the standard DTRM is added in order to preserve a conservation of energy as:

$$C_R = \frac{\sum\limits_{boundary\_faces\_j} A_j q_{out,j}}{\sum\limits_{\substack{starting\\ points\_j}} \left( q_{out,j} \left( \sum\limits_{\substack{i=ending\_points\\ of\_ray\_j}} \cos\Theta_{i,j}\Delta\Omega_{i,j} A_i / \pi \right)\right)} \tag{9}$$

The total hemispherical irradiation on a boundary cell face is obtained by collecting the incoming radiant intensities for all the rays fired from that face:

$$q_{in} = \int\limits_{\vec{s}\cdot\vec{n}<0} i' \vec{s}\cdot\vec{n} d\Omega \approx \sum_{kk=1}^{N_{rays}} i'_{kk} \cos\Theta_{kk} \Delta\Omega_{kk} \tag{10}$$

As each radiosity ray propagates through the domain it exchanges radiative energy with the participating medium, yielding a contribution to the source term in the energy equation for the medium (fluid). For a control volume that is intersected by a ray, the energy source due to radiant intensity change along one ray is calculated as:

$$\overline{S}_{k_j} = (i'_{n+1} - i'_n) A_j \cos\Theta_j \Delta\Omega_j \tag{11}$$

Total energy source within a control volume is obtained by collecting the source terms, according to Eq. (11), due to all the rays that intersect that control volume:

$$\overline{S}_k = \sum_{intersecting\_rays\_j} \overline{S}_{k_j} \tag{12}$$

**$NO_x$ MODEL**

The prediction of $NO_x$ emissions may be decoupled from the generalized combustion and radiation model and executed after the flame structure has been predicted because total amount of nitrogen oxides formed in combustion are generally low and does not affect the flame structure. Solving the NO pollutant model equation jointly with the combustion model equations is far more complex. As a consequence, CFD solver is used in a post-processing step to solve NO transport equation based on a flow field and combustion variables obtained from the converged solution. The transport equation for NO mass fraction, accounting for the convection, diffusion, production and consumption of NO:

$$\frac{\partial(\bar{\rho}\tilde{Y}_{\mathrm{NO}})}{\partial t}+\frac{\partial(\tilde{u}_i\bar{\rho}\tilde{Y}_{\mathrm{NO}})}{\partial x_i}=\frac{\partial}{\partial x_i}\left(\bar{\rho}D_t\frac{\partial\tilde{Y}_{\mathrm{NO}}}{\partial x_i}\right)+\bar{S}_{\mathrm{NO}} \tag{13}$$

The source term $S_{NO}$ in Eq. (13) is predicted from contribution from two predominant mechanisms: extended Zeldovich mechanism and Fenimore prompt mechanism. The extended Zeldovich mechanism NO mechanism involves reaction between atmospheric nitrogen and the atomic oxygen produced in the high temperature regions of the flame and the subsequent reactions with the atomic nitrogen. This process is described by a set of chemical reactions:

$$\mathrm{N}_2+O\overset{k_1}{\leftrightarrow}\mathrm{NO{+}N} \tag{14}$$

$$\mathrm{N}+\mathrm{O}_2\overset{k_2}{\leftrightarrow}\mathrm{NO{+}O} \tag{15}$$

$$\mathrm{N}+\mathrm{OH}\overset{k_3}{\leftrightarrow}\mathrm{NO}+\mathrm{H} \tag{16}$$

The rate coefficients for reactions (14-16) used in present model are based on the evaluation of Hanson and Salimian [10].

Using the quasi-steady assumption that the rate of consumption of free nitrogen atoms becomes equal to its formation rate, the expression for the overall rate of thermal NO formation process is given by:

$$\frac{dc_{\mathrm{NO}}}{dt}=2k_{1f}c_{\mathrm{O}}c_{\mathrm{N}_2}\frac{\left(1-\dfrac{k_{1b}k_{2b}c_{\mathrm{NO}}^2}{k_{1f}c_{\mathrm{N}_2}k_{2f}c_{\mathrm{O}_2}}\right)}{\left(1+\dfrac{k_{1b}c_{\mathrm{NO}}}{k_{2f}c_{\mathrm{O}_2}+k_{3f}c_{\mathrm{OH}}}\right)} \tag{17}$$

Since the thermal NO mechanism involves O and OH radicals, it is necessary to couple the thermal NO reactions to the fuel oxidation reactions. However, compared to the fuel oxidation reactions, the overall rate of NO formation by the thermal mechanism is slow and, can be assumed that thermal formation reactions can be decoupled from the fuel oxidation mechanism. In this situation, the equilibrium values of temperature and concentrations of $O_2$, $N_2$, O and OH are assumed. Using this approach, oxygen atoms have been assumes to be in equilibrium with $O_2$. However, an improvement to this method is that value for O and OH can be obtained using the partial equilibrium assumption or using O and OH radical concentration when they are well predicted using and advance chemistry model. In the partial equilibrium assumption, the concentration of O and OH can be expressed:

$$c_{\mathrm{O}} = 36,64 T^{0,5} c_{\mathrm{O}_2}^{0,5} \exp\left(\frac{-27123}{T}\right) \tag{18}$$

$$c_{\mathrm{OH}} = 2,129 \times 10^{2} T^{-0,57} \exp\left(-\frac{4595}{T}\right) c_{\mathrm{O}}^{0,5} c_{\mathrm{H_2O}}^{0,5} \tag{19}$$

Prompt NO was firstly identified by Fenimore (1971) that the rate of NO generated during combustion of hydrocarbon substances is considerably higher than that predicted by the Zeldovich mechanism. This enhanced NO formation is attributed to the presence of hydrocarbon species, which result from fuel fragmentation during the combustion process. Prompt NO is formed by the reaction of atmospheric nitrogen with hydrocarbon fragments, which is subsequently oxidized to form NO. The prompt NO mechanism forms NO from nitrogen much earlier in the flame than the thermal NO mechanism, as its name suggests.

$$\mathrm{CH_x} + \mathrm{N_2} \leftrightarrow \mathrm{HCN} + \mathrm{N} + \ldots \tag{20}$$

The model used in the present study to predict prompt NO concentration is calculated from the De Soete [11] global model as:

$$\frac{dc_{\mathrm{NO}}}{dt} = kf c_{\mathrm{O}_2}^{b} c_{\mathrm{N}_2} c_{fuel} \exp\left(-\frac{E}{RT}\right) \tag{21}$$

Values of $k$ and $E$ are experimental constants [12]. In the above equation $f$ is an empirical function designed to account for the effect of various aliphatic hydrocarbon fuels and air/fuel ratio effects, and is given as:

$$f = 4.75 + A_1 n - B_1 \phi + B_2 \phi^2 - B_3 \phi^3 \tag{22}$$

where $n$ is a number of carbon atoms in the fuel and $\phi$ is the equivalence ratio. $A_1$, $B_1$, $B_2$ and $B_3$ take the values of 0,82, 23,2, 32, 12,2, respectively.

Oxygen reaction $b$ order depends on flame conditions. According to De Soete [11], oxygen reaction order is uniquely related to oxygen mole fraction in the flame:

$$\begin{aligned} & X_{\mathrm{O}_2} \le 4,1 \times 10^{-3} \rightarrow b = 1,0 \\ & 4,1 \times 10^{-3} \le X_{\mathrm{O}_2} \le 1,11 \times 10^{-2} \rightarrow b = -3,95 - 0,9 \ln X_{\mathrm{O}_2} \\ & 1,11 \times 10^{-2} < X_{\mathrm{O}_2} < 0,03 \rightarrow b = -0,35 - 0,1 \ln X_{\mathrm{O}_2} \\ & X_{\mathrm{O}_2} \ge 0,03 \rightarrow b = 0 \end{aligned} \tag{23}$$

Chemistry-turbulent interactions are modelled by integration of chemical kinetic rates (source term $S_{\mathrm{NO}}$) with respect to fluctuating temperature and appropriate probability density function (PDF).

## NUMERICAL SIMULATION

The presented models were implemented through a user functions in commercial CFD code FIRE™ and numerically tested and investigated for IJmuiden furnace of the International Flame Research Foundation (Netherlands). This is a horizontal tunnel furnace constructed of refractory brickwork, approximately square in cross section, with an arch roof, as shown in Figure 2. The fuel and air streams are injected through a burner system in one end wall, the usual arrangement being for a single horizontal flame fired axially along the length of the furnace and exhausting through a chimney at the far end of the furnace. The fuel is injected into a pre-combustion chamber formed by a divergent nozzle followed by a cylindrical extension of variable length. Air is supplied in two streams, swirled primary and secondary.

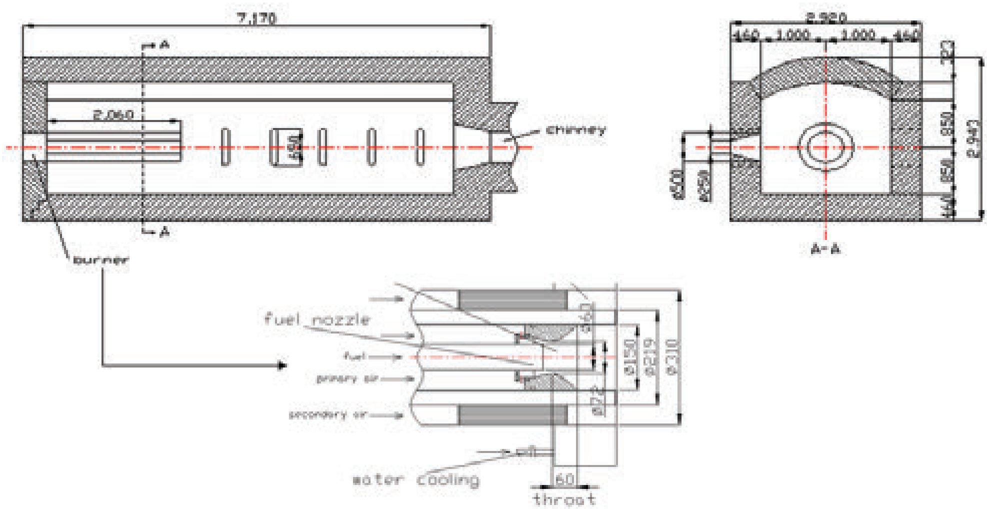


Figure 2. IJmuiden furnace and burner configuration

Three-dimensional steady-state simulations of the combustion process in this furnace were performed using the AVL's CFD package FIRE™, which uses conventional numerical methods and differencing schemes for a completely arbitrary mesh. This simulation was used to provide the general flow and heat transfer characteristics of the furnace. Steady oil-combustion and standard species transport module was used for the simulation of combustion process. Turbulence was modelled using the k-ε model, with standard wall functions while radiative heat transfer is computed using the discrete transfer radiation method . The nitric oxides formation was predicted with a post-processing computation using converged solution of a pre-calculated flame structure. The $NO_x$ post-processor was predicted thermal NO formation via the extended Zeldovich mechanism, where O and OH concentrations were based on partial equilibrium assumptions. In addition, prompt NO mechanism was incorporated for total prediction of NO formation in furnace.

Numerically simulated, adiabatic and non-adiabatic (DTRM) temperature axial profiles for oil-fired IJmuiden furnace are shown in Figure 3. It is noticeable that adiabatic profile over-predicts the flame temperature, especially at the downstream locations. Non-adiabatic temperature profile is in much better agreement with the experimental measurements, indicating the need for accurate radiative heat transfer model.

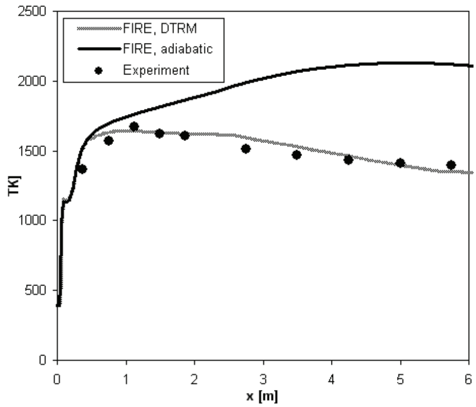


Figure 3. Axial profile of temperature along axis symmetry of the burner

Figure 4 shows temperature and NO distribution in a vertical plane along the furnace axis. The trends in the results show that NO distribution is well characterized by the temperature distribution. Higher concentration of NO mass fraction can be observed in the area of highest temperature in the furnace. The total amount of NO mass fraction produced by combustion is overall small, indicating that production of thermal NO is significant only at the temperatures above approximately 1500°C due to high activation energy. Actually, atmospheric nitrogen has a strong triple bond, and is extremely stable. Thus, it can be only decomposed at the very high temperatures within the flame. At temperatures lower than 1000°C, the amount of thermal NO formed is reduced, while at temperatures below 760°C NO is either generated in much lower concentrations or not at all.

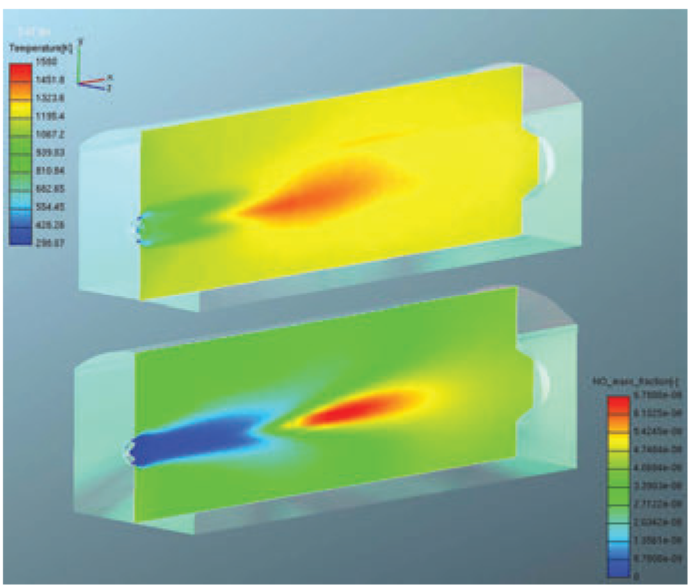

Figure 4. Temperature and NO mass fraction distribution

## CONCLUSION

The three dimensional numerical simulation of the nitrogen pollutant formation in real-size oil fired furnace was numerically simulated and predictions of temperature and NO mass fraction have been presented. Steady oil-combustion model was used for combustion simulation while the discrete transfer radiation method (DTRM), together with the WSGGM, was used for radiation predictions, showing the importance and need to accurately predict the temperature field and nitrogen pollutants. The nitric oxides formation was predicted with a post-processing computation using converged solution of a pre-calculated flame structure. The chemical kinetic mechanisms used for $NO_x$ modelling were limited to sufficiently few homogeneous reactions in order to approximate the essential features of the NO reaction process, allowing the reduction of calculation time and to make this time-consuming method more attractive for industrial application.

## NOMENCLATURE

| | | |
|---|---|---|
| $A$ | $[m]$ | area |
| $C_R$ | [-] | correction factor |
| $D_t$ | $[m^2/s]$ | molecular diffusivity |
| $E$ | $[J/mol]$ | activation energy |
| $M$ | $[kg/mol]$ | molecular weight |
| $N$ | [-] | total number |
| $S$ | $[kg/m^3s]$ | source |
| $T$ | $[K]$ | temperature |
| $Y$ | [-] | mass fraction |
| $a$ | $[1/m]$ | absorption coefficient |
| $b$ | [-] | combustion velocity coefficient for oil |
| $c$ | $[mol/m^3]$ | concentration |
| $d$ | $[mm]$ | droplet initial diameter |
| $f$ | [-] | correction factor |
| $i$ | $[W/m^2sr]$ | radiant intensity |
| $k$ | $[kg/kgs]$ | constant of reaction rate |
| $k$ | $[m^3/mols]$ | reaction rate coefficient |
| $l$ | $[m]$ | distance that a ray makes in a control volume |
| $n$ | [-] | number of carbon atoms |
| $\vec{n}$ | [-] | unit boundary cell face normal vector |
| $q$ | $[W/m^2]$ | radiation heat flux |
| $r$ | $[m]$ | radius |
| $\vec{s}$ | [-] | unit ray direction vector |
| $t$ | $[s]$ | time |
| $u_i$ | $[m/s]$ | Cartesian velocity component |
| $x_i$ | $[m]$ | Cartesian co-ordinate |
| $\Theta$ | $[rad]$ | polar angle |
| $\Omega$ | $[sr]$ | solid angle |
| $\varepsilon$ | [-] | total emissivity |
| $\chi$ | $[1/s]$ | scalar dissipation rate |
| $\nu$ | $[m^2/s]$ | kinematic viscosity |
| $\rho$ | $[kg/m^3]$ | density |
| $\sigma$ | $[W/m^2/K^4]$ | Stefan-Boltzmann constant |
| $\tau$ | $[s]$ | time |
| $\theta$ | [-] | equivalence ratio |

**Subscripts and Superscripts**

| | |
|---|---|
| $0$ | beginning of incremental path |
| $B$ | fuel |
| $H$ | hydrogen |
| $HCN$ | hydrogen cyanide |
| $N$ | nitrogen |

| | |
|---|---|
| *NO* | nitric oxide |
| *O* | oxgen |
| *OH* | hydroxide |
| *R* | radiative |
| *cc* | coke combustion |
| *ei* | evaporation and induction |
| *in* | incoming |
| *in* | incoming |
| *n* | rays entrance in intersected control volume |
| *n+1* | rays exit from intersected control volume |
| *out* | outgoing |
| *ox* | oxidation |
| *rays* | rays |
| *spec* | species |
| *w* | wall |
| − | Reynolds average |
| ~ | Favre average |

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# EXPERIMENTAL AND NUMERICAL STUDY OF THE FLOW AND HEAT TRANSFER IN PLATE HEAT EXCHANGER CHANNELS

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## ABSTRACT

This paper presents an experimental and numerical study of the flow and heat transfer in corrugated chevron channels of plate heat exchangers. 3D CFD simulations were performed on two channel models, each consisting of 16 cells, with chevron angles $\beta$=28°and $\beta$=65° under laminar, transitional and fully turbulent flow conditions. Applied the RNG $k$-$\varepsilon$ and the RES models for turbulent flow simulation are presented and discussed. Computationally obtained flow patterns are analyzed and compared with recorded ones from the visualization tests. Different methods for evaluation of the simulated hydraulic and thermal characteristics are discussed and compared with experimental data from present measurements and literature.

**KEYWORDS:** Plate heat exchangers, numerical simulation, pressure drop, heat transfer

## 1. INTRODUCTION

Plate heat exchangers (PHEs) are nowadays used in a wide range of applications for heating, cooling, evaporation and condensation purposes. Due to their low volume/surface ratio i.e. compactness, high overall heat transfer coefficient, low production and operational costs, they have found an increasing usage in place of conventional shell&tube heat exchangers. A number of investigations have been conducted with the aim of exploring the complex flow mechanisms that occur in a corrugated PHE channels. Also, several results from the thermal-hydraulic tests have been reported revealing considerable disagreement between derived expressions for prediction of the heat transfer and pressure drop [1]. A limited number of CFD studies on the flow through the smallest repeat unit of a channel (a single cell of a sinusoidal form, Figure 1) can be found in the literature [2]; [3]. Those authors applied periodic flow conditions combined with imposed constant wall temperature and/or constant wall heat flux conditions. This work provides results from the 3D CFD analysis of the flow through 16 cells of two plate heat exchanger channels with different chevron angles: $\beta$=28° and $\beta$ =65° (Figure 1). Such approach allowed for investigation of the flow characteristics and flow pattern formation starting from the channel entrance to the fully developed flow region. Additionally, avoiding a need for implementation of the periodic boundary conditions, as is case when the flow through only one cell is simulated, the heat transfer characteristics can also be presented in terms of the total heat flux exchanged between the central channel fluid and neighbor channel fluid under conditions that occur in a real heat exchanger, instead of determining only the heat transfer coefficient. The advantage of such approach is that it enables direct comparison of a computed with measured channel overall heat transfer capacity. Presented heat transfer and pressure drop characteristics are related to the fully developed flow conditions in the laminar, transient and turbulent region corresponding to the Re-numbers from *Re*=10 to *Re*=6000.

## 2. MODEL DETAILS

Three dimensional simulation of the PHE channel flow was performed using the computer code FLUENT. Similarly, like in a majority of computer codes within the field of the computational fluid dynamics, the finite volume method is employed here to convert the governing differential equations to algebraic equations suitable to be solved numerically. The computational domain consisting of 16 smallest repeat channel units (cells) is shown on Figure 1. Each cell (meshed using software package GAMBIT) is covered with 56160 hexahedral/wedge control volumes each consisting of 8/6 nodes placed on element edges, respectively, and distributed within each cell by means of the cooper meshing scheme. This ensured that all of cell cross sections perpendicular to the mean flow direction are equally meshed. It was especially important to ensure that the fine grid in the near wall region (generated using a boundary layer option) is uniformly swept along the wall surface when initially created at cell flow inlet or outlet surface. The channel geometry (Figure 1) used in the simulation is consistent with the previously tested geometry, whereby the chevron angle was varied from $\beta$=28° to $\beta$=65° as the most influencing parameter on the flow, while the other parameters such as pressing depth *b* and corrugation width *l* (i.e. pitch *p*) were kept constant. Flow conditions considered here correspond to the following Re-numbers: *Re*=10, *Re*=700÷900 and *Re*=4600÷6000. This means that the present simulation should involve laminar, transitional and turbulent flow modeling. Fluids used in simulation were water at *Re*=(10÷6000) and aqueous solution of glycerol (85% of glyc. by weight) at *Re*=(10÷15) with identical properties to those of the test fluids used in the previously performed tests.

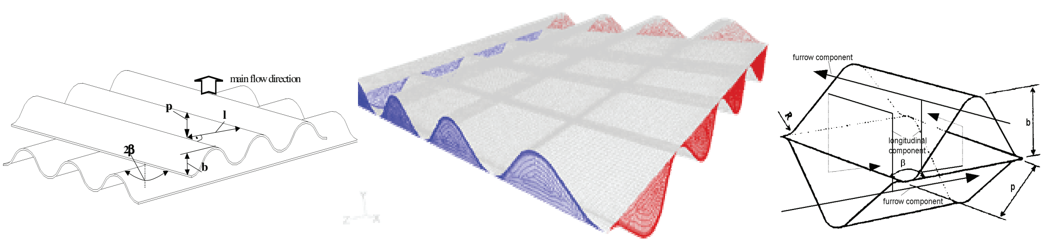



Figure 1. Geometry of a chevron channel composed of two corrugated plates together with the computational domain consisting of 16 cells and detail of a position of the flow components within particular cell

### 2.1 Governing equations and flow modeling

The standard set of equations discretized in order to solve laminar and turbulent flow comprises the equation for conservation of mass, conservation of momentum (Navier-Stokes equations) and conservation of energy. Written in the *i*-th direction for incompressible laminar flow with constant fluid properties and neglecting viscous dissipation terms they are given as

$$\frac{\partial u_j}{\partial x_j} = 0 \tag{1}$$

$$\frac{\partial u_i}{\partial t} + \frac{\partial (u_i u_j)}{\partial x_j} = -\frac{1}{\rho}\frac{\partial p}{\partial x_i} + \nu \frac{\partial}{\partial x_j}\left(\frac{\partial u_j}{\partial x_i} + \frac{\partial u_i}{\partial x_j}\right) + \frac{1}{\rho} f_i \tag{2}$$

$$\frac{\partial T}{\partial t} + \frac{\partial (u_j T)}{\partial x_j} = a\left(\frac{\partial^2 T}{\partial x_j^2}\right) + \frac{1}{\rho c} S_T \tag{3}$$

where $f_i$ is the body force per unit volume and $S_T$ is the volumetric rate of heat generation.
A useful generalized representation of these equations is

$$\frac{\partial \phi}{\partial t} + \frac{\partial (u_j \phi)}{\partial x_j} = \Gamma \frac{\partial^2 \phi}{\partial x_j^2} + S \tag{4}$$

Here, the dependent variable $\phi$ stands for a variety of quantities such as temperature, velocity component, turbulent kinetic energy etc, while $S$ is the source term and $\Gamma$ is the diffusion coefficient. The finite volume method employed here is based on integrating the conservation Equation (4) about each control volume within a computational domain. The face values of $\phi$ are interpolated from the upstream cell center values using the second order upwind scheme.
In discretization of the momentum equation, the standard interpolation scheme was used to obtain the face values of pressure between centers of adjacent control volumes. For higher Re number flows (here at $Re$>4600) where additional swirling is expected, the PRESTO scheme was used instead [4], since a large pressure gradients are expected between control volume centers. The SIMPLE pressure-velocity coupling algorithm was used to derive the equation for pressure from the discrete continuity equation. No benefits in terms of a faster convergence have been observed when the SIMPLEC algorithm [5] was used instead.
All calculations were performed in a double precision segregated steady state solver.
In the simulations of flows at $Re$>700 two different models were employed for turbulence modeling, namely the RNG $k$-$\varepsilon$ model and the Reynolds stress transport model (RES). In both approaches the Reynolds averaged equations need to be solved. Those equations can be written in the following form (incompressible flow with constant fluid properties):

$$\frac{\partial u_j}{\partial x_j} = 0 \tag{5}$$

$$\rho\frac{\partial u_i}{\partial t} + \rho\frac{\partial (u_i u_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \mu \frac{\partial}{\partial x_j}\left(\frac{\partial u_j}{\partial x_i} + \frac{\partial u_i}{\partial x_j}\right) - \rho\frac{\partial}{\partial x_j}\left(\overline{u_i' u_j'}\right) \tag{6}$$

Here, we can see that the additional terms appear, termed as the Reynolds stresses, in order to account for the effect of turbulence:

$$\rho \overline{u_i' u_j'} = \mu_t \left( \frac{\partial u_j}{\partial x_i} + \frac{\partial u_i}{\partial x_j} \right) - \frac{2}{3} \left( \rho k + \frac{\partial u_j}{\partial x_i} \right) \delta_{ij} \tag{7}$$

In order to close Equation (6) the Reynolds stresses which contain instantaneous values of velocity are modeled based on the Boussinesq hypothesis (Equation 7) in the $k$-$\varepsilon$ model, or in the RES model by means of solving the transport equations for each term of the Reynolds stress tensor together with one additional scale determining equation (normally for $\varepsilon$). In the RNG $k$-$\varepsilon$ model two transport equations for $k$ and $\varepsilon$ were solved (Equations (8)-(9)) while the turbulent viscosity $\mu_t$ is computed as a function of $k$ and $\varepsilon$ (Equation (10)):

$$\rho \frac{\partial k}{\partial t} + \rho \frac{\partial (u_j k)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + G - \rho \varepsilon \tag{8}$$

$$\rho \frac{\partial \varepsilon}{\partial t} + \rho \frac{\partial (u_j \varepsilon)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + C_1 G \frac{\varepsilon}{k} - C_2 \rho \frac{\varepsilon^2}{k} - R_\varepsilon \tag{9}$$

where $G$ represents the generation of turbulent kinetic energy due to the mean velocity gradients.

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \tag{10}$$

with $C_\mu$=0.0845, $C_1$=1.42, $C_2$=1.68.

In the standard $k$-$\varepsilon$ models turbulent Pr-numbers for $k$ and $\varepsilon$ are taken as constants $\sigma_k$=1.0 and $\sigma_\varepsilon$=1.3, respectively.

The RNG (renormalization group) based $k$-$\varepsilon$ model used in the present simulation has been proved to be more reliable and accurate for a wider class of flows than the standard $k$-$\varepsilon$ model. The main difference relative to the standard $k$-$\varepsilon$ model lies in the analytical derivation of the $\varepsilon$ transport equation.

Also, the RNG theory provides an analytical formula (more details in [6]) for turbulent *Pr* numbers $\sigma_k$ and $\sigma_\varepsilon$ instead of using constant values, as well as the differential formula for the effective viscosity which accounts for low Re-number effects.

In the RES model the turbulent kinetic energy $k$ needed for modeling a specific term in the transport equation for Reynolds stresses $\rho \overline{u_i' u_j'}$ is obtained directly from the normal Reynolds stress tensor $k = \frac{1}{2} \overline{u_i' u_i'}$.

Values of $k$ near the wall are computed from the equation derived by contracting the exact transport equation for Reynolds stresses [7] which is essentially identical to Equation (8) used in the $k$-$\varepsilon$ model.

The dissipation tensor $\varepsilon_{ij}$ in the RES model is defined as

$$\varepsilon_{ij} = \frac{2}{3} \delta_{ij} \rho \varepsilon \tag{11}$$

where the scalar dissipation rate $\varepsilon$ is obtained from the transport equation similar to the one from the $k$-$\varepsilon$ model (Equation (9)). The turbulent viscosity $\mu_t$ is computed similarly to the $k$-$\varepsilon$ model (Equation (10)) taking $C_\mu$=0.09.
Convective turbulent heat transfer is modeled based on the Reynolds' analogy to turbulent momentum transfer where the modeled energy equation takes the following form for both the $k$-$\varepsilon$ and the RES model:

$$\frac{\partial T}{\partial t}+\frac{\partial u_j T}{\partial x_j}=\frac{1}{\rho}\frac{\partial}{\partial x_j}\left(\left(k+\frac{c\mu_t}{\Pr_t}\right)\frac{\partial T}{\partial x_j}\right)+\frac{1}{\rho c}S_T \qquad (12)$$

In the standard $k$-$\varepsilon$ and RES model $Pr_t$=0.85 while in the RNG model $Pr_t$ is calculated from the above mentioned analytical formula [6] as a function of $\mu_{mol}/\mu_{eff}$ which is an advantage of the RNG model since it was experimentally confirmed that the turbulent Pr-numbers vary with the molecular Pr-number and turbulence [7].
In both described models the enhanced wall treatment was applied to resolve the viscosity affected region all the way down to the wall. The enhanced wall treatment combines two-layer modeling (viscosity affected region/fully turbulent region) with enhanced wall functions. The standard wall functions modeling would, instead, bridge this region between the wall and fully turbulent region. Such approach ensures accuracy when both coarse (wall function) meshes are used in high Re-number flows and also when fine low Re-number meshes are generated. The enhanced wall functions were derived by smoothly blending an enhanced turbulent wall law with the laminar wall law according to the following Equation (13) from Kader [8] which gives a reasonable representation of the velocity profiles within the laminar sublayer, buffer region and fully turbulent outer region:

$$u^+=e^{\Gamma}u^+_{lam}+e^{\frac{1}{\Gamma}}u^+_{turb} \qquad (13)$$

In the two-layer modeling the domain is subdivided in two regions – fully turbulent and viscosity affected. In the fully turbulent region the $k$-$\varepsilon$ model and RES models are employed while in the viscosity affected region the one equation-model of Wolfstein [9] is employed.
Two layer formulation of turbulent viscosity is smoothly blended with previously stated high Re-number definition for $\mu_t$ from outer region (Equation (10)).
The similar procedure is used to ensure a smooth transition between algebraically specified $\varepsilon$ (Equation (14)) in the inner region and $\varepsilon$ obtained from the transport equation valid for outer region of turbulent boundary layer (Equation (9)).

$$\varepsilon=\frac{k^{3/2}}{l_\varepsilon} \qquad (14)$$

Enhanced thermal wall functions are derived following the same approach used for the profile of $u^+$ (Equation (13)).

## 2.2 Boundary conditions

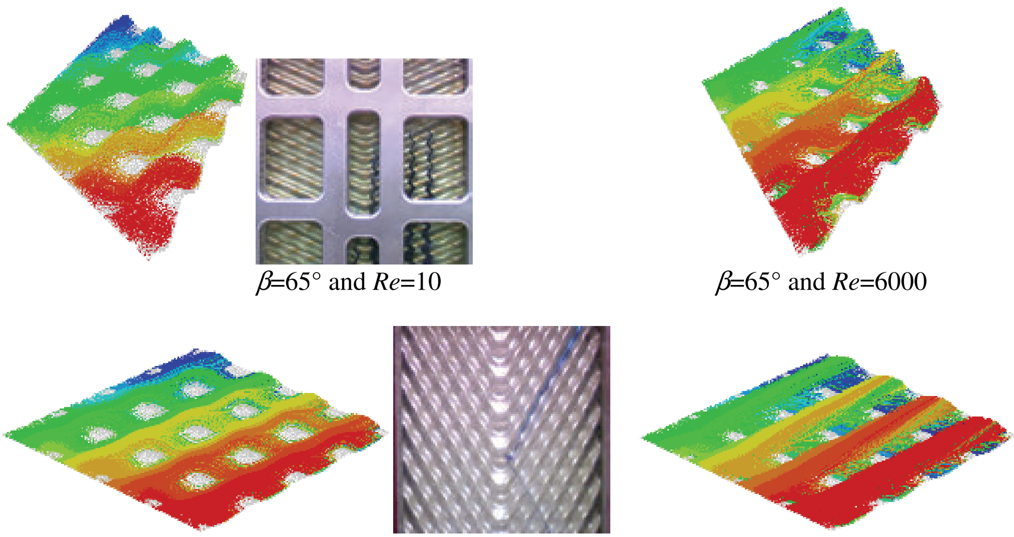
Inlet faces of the whole computational domain (Figure 1) were treated separately as the upper and lower corrugation velocity inlets while the opposite faces as flow outlets. At inlets of a particular channel half the velocity vector was aligned with the corresponding corrugation. This was supposed to be a good approximation of the actual flow condition at the channel entrance. The magnitude of the resultant velocity was determined from a measured volume flow rate and actual flow cross sectional area of the tested channel $A_{cross}$. The specification of the turbulent boundary conditions was the same for the $k$-$\varepsilon$ and the RES model i.e. in terms of the turbulence intensity ($I=u'/u_{avg}$= 1%) and turbulent viscosity ratio ($\mu_t/\mu$=10). The $k$ and $\varepsilon$ values are then estimated from these quantities. In the RES model the Reynolds stresses at inlets are derived under the assumption of isotropy of turbulence ($\overline{u_i' u_j'} = 0$, while the normal Re-stress component in each direction is obtained from $\overline{u_i'^2} = \frac{2}{3}k$ ). The outflow boundary conditions were applied at outlets of the simulated channel on which zero diffusion fluxes for all flow variables were imposed in the direction normal to the exit plane (Neumann boundary conditions). The zero diffusion flux condition applied at outflow faces means that the conditions of the outflow plane are extrapolated from within the domain and have no impact on the upstream flow, which is consistent with fully developed flow assumption. Thermal boundary conditions were defined in terms of the external heat transfer coefficient $\alpha_{ext}$ and external fluid temperature $T_{f,ext}$, known from the measurements. The heat flux to the wall is then computed as

$$q = \alpha_{ext}\left(T_{f,ext} - T_{wall}\right) \tag{15}$$

# 3. RESULTS AND DISCUSSION

## 3.1 Flow pattern

Simulated flow distribution over the lower channel plate is shown on Figure 2 for different chevron angles and Re- numbers. The flow pattern recorded during the previously performed visualization tests [10] is provided for comparison. At flow conditions corresponding to the mean channel Re-number $Re$=(10÷15) the criss-cross and wavy longitudinal flow pattern appear in the channels with $\beta$=28° and $\beta$=65°, respectively. The simulated and recorded flow patterns cannot be easily compared at higher Re-numbers due to the large mixing induced, that lead to the dispersion of the dye introduced in the channel flow core.



$\beta$=65° and *Re*=10 $\beta$=65° and *Re*=6000

$\beta$=28° and *Re*=15 $\beta$=28° and Re=4667

Figure 2. Simulated and experimentally recorded flow patterns in the channels with $\beta$=65° (wavy longitudinal pattern) and $\beta$=28° (criss-cross pattern) at low Re-numbers flow and fully turbulent high Re flow (*Re*>4667)

Simulated flow patterns confirm assumptions made in the previous works about existence of the two flow components, namely the furrow and the longitudinal one. The furrow component refers to the part of a fluid which does not change direction when passing throughout successive cells, while the longitudinal component changes direction within each cell flowing from one plate to the opposite one, as schematically shown on Figure 1.

Such a change of direction is believed to be accompanied by strong mixing within the cell, resulting in higher heat transfer coefficient as well as in higher pressure drop. An increase of the Re-number and aspect ratio $b/l$ together with a decrease of the chevron angle $\beta$ would promote the furrow component which results in the criss-cross (or furrow) flow pattern. The opposite conditions would promote the longitudinal component whose dominance is responsible for the wavy longitudinal flow pattern. It should be mentioned that the simulated turbulent flow field was shown to be qualitatively independent of the turbulence model employed in simulation.

**3.2 Pressure drop**

Figure 3 shows computed pressure drop of the simulated channels together with the measured data expressed in terms of the total pressure drop per unit of developed channel area and the Fanning friction factor $f$. The pressure drop is obtained as the difference between the total pressures (area weighted averages of node values) at cell inlets and outlets. Presented values for pressure drop comprise effects of the channel entrance parts where the shear is normally higher than in the part with fully developed flow conditions. The Fanning friction factor $f$ is calculated using the expression

$$f = \frac{\Delta p}{4\left(\frac{L}{d_h}\right)\frac{\rho u_m^2}{2}} \tag{16}$$

and given in Figure 3 together with the measured data and experimental data from the literature [1]. In the case of the channel with $\beta$=28° the characteristic length $L$ is taken as the distance between the entrance and exit face along a single furrow provided that is the principal flow direction. In the case of the dominating longitudinal flow in the $\beta$=65° channel, $L$ is defined as the shortest diagonal distance between two corners of simulated geometry. This way of defining a characteristic length with the respect to the main flow direction for particular channel could only ensure the Fanning friction factor to be comparable with measured values for the whole channel. In general, the simulated pressure drop (expressed in $\Delta p/m^2$) relatively slightly differ from the measured values at higher Re numbers (up to 10%, in average) while large discrepancies are present in the laminar flow at $Re$=(10÷15) when a highly viscous glycerol was set as a fluid. When comparison is made in terms of the Fanning friction factor $f$, then the differences between the simulated and measured results are lower for plate with $\beta$=65° (2%÷12%) and much larger for $\beta$=28°plate (8%÷42%). This is believed to be a consequence of the present definitions of a characteristic length $L$ and mean velocity $u_m$, which cannot completely reflect a rather complex flow distribution in the corrugated channel characterized by variable flow cross sections.

### 3.3 Heat transfer

As stated before, the heat transfer characteristics of the simulated channel are evaluated in terms of the total heat flux exchanged with the fluid flowing out of the channel (Figure 3). Heat flux reported refers to the single cell “far” from the channel entrances where the flow is proved to be both hydrodynamically and thermally fully developed (see Figure 2). The average heat flux for the whole simulated geometry is somewhat higher being affected by developing regions of more intensive heat (and momentum) transfer. Indeed, when comparing computed results with experimental ones, only the fully developed conditions are relevant figure of merit as they prevail in a real heat exchanger. The heat transfer coefficient could be evaluated based on the average channel flow heat flux, average wall temperature and averaged fluid temperature as defined by Equation (17). Such averaging, however, leads to the computational errors since the heat flux, wall temperature and corresponding mean fluid temperature can not be properly “coupled” in order to yield correct values for heat transfer coefficient at certain cross sections of the channel. In order to mitigate this affect to some extent, another approach of obtaining heat transfer coefficient based on the mean logarithmic temperature difference $\Delta t_m$ is used instead, according to Equations (18)-(19). Such approach takes into account the change of the mean fluid temperature along the channel and yields (5-10)% better results, but still it can only include the mean values for a fluid temperature and wall heat flux. This finally results in a higher deviations from the measured values-up to 30 % (Figure 3) relative to the case when comparison is made in terms of the heat fluxes, when they go up to 15% (Figure 3). That is the main reason why simulations of the flow through one cell with periodical conditions cannot give a correct picture of the thermal characteristics as they do not allow use of the external convective heat transfer boundary conditions applied here, and therefore simulation of the exchanged heat. Simulated heat fluxes are (5÷15)% higher than the measured ones at the flow corresponding to $Re$>700 while deviations become again large for laminar flow, but they are less serious compared to the case with pressure drop calculations. The RES model gives better results for both simulated channels and all turbulent flow conditions (Re> 700) than the RNG $k$-$\varepsilon$ model. Thermal characteristics expressed in terms of $Nu/Pr^{1/3}$ are up to (10÷30)% higher than the measured ones at $Re$>700. In the laminar flow simulations the values of $Nu/Pr^{1/3}$ are close to the measured values only for channel with $\beta$=65° when water was set as a fluid instead of glycerol.

$$\alpha_{16cells} = \frac{q_{av}}{\left(T_{wall,av} - T_{f,av}\right)} \tag{17}$$

$$\frac{1}{\alpha_{16cells}} = \frac{1}{k} - \frac{1}{\alpha_{ext}} \tag{18}$$

$$k = q_{devel.} / \Delta t_m = q_{devel.} / \frac{\left(t_{in} - t_{f,ext}\right) - \left(t_{out} - t_{f,ext}\right)}{\ln \dfrac{\left(t_{in} - t_{f,ext}\right)}{\left(t_{out} - t_{f,ext}\right)}} \tag{19}$$

$$Nu = \frac{\alpha_{16cells} d_h}{\lambda} \tag{20}$$

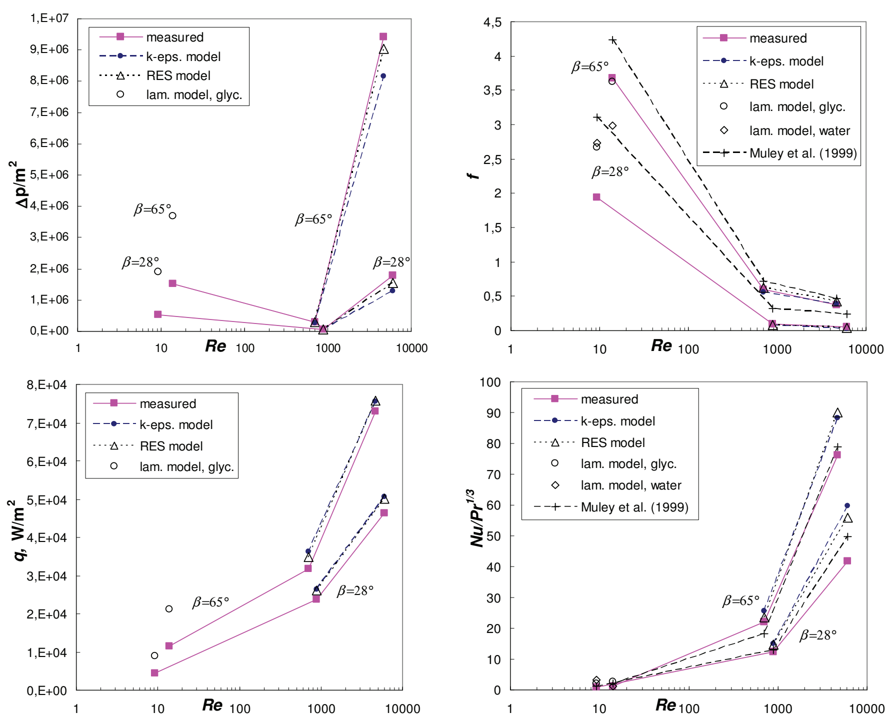



Figure 3. Simulated hydraulic and thermal characteristics of modeled channels with $\beta$=28° and $\beta$=65° compared to the experimental data from present measurements and literature (Muley *et al.*, [1])

## 4. CONCLUSION

Numerical simulations of the flow distribution, momentum and heat transfer in corrugated chevron channels of plate heat exchangers were performed under laminar, transient and fully turbulent flow conditions on the model exchanger consisting of the 16 cells. The RNG $k$-$\varepsilon$ and RES models were used for turbulent flow simulations. Computed flow patterns compared with recorded ones confirm assumptions about occurrence of two flow components responsible for the criss-cross flow pattern in low angle channels ($\beta$<45°) dominated by the furrow component and for the wavy longitudinal flow pattern in high angle channels ($\beta$>45°) dominated by the longitudinal component. Higher velocities and deeper corrugations (higher aspect ratio $b/l$) would promote the furrow component which flows along particular furrow, while the longitudinal component, which changes direction in each cell, is promoted by opposite flow conditions and geometry. Comparison with performed thermal and hydraulic tests indicates that the pressure drop and heat transfer calculations give reliable results only for the transient and fully turbulent flows at $Re$>700, whereat the RES model was shown to yield more accurate results than the $k$-$\varepsilon$ model. In this case the simulated and measured results differ in average from 5% to 15 % when expressed in terms of the pressure drop per channel surface unit and exchanged heat flux. Larger deviations of the friction factor $f$ from the measured values are attributed to the inappropriate definitions of the characteristic length and the mean velocity which do not fully reflect a complex flow distribution and variable channel geometry. The same deviations were observed in presentation of the thermal characteristics by means of $Nu/Pr^{1/3}$. Beside the inappropriate definition of the characteristic length, this is also believed to be a consequence of unavoidable averaging of the channel wall and fluid temperature in calculation of the heat transfer coefficient. For this reasons, computationally less expensive simulations performed on the single cell using a periodic boundary conditions are suspected to give a correct picture of the thermal characteristics, as they do not allow use of external heat transfer boundary conditions i.e. simulation of the heat flux exchanged between two fluids as a relevant figure of merit for comparison with experimental data.

## NOMENCLATURE

| | | | | |
|---|---|---|---|---|
| $A_{cross}$ | cross-sectional free flow area (m$^2$) | | **Subscripts** | |
| $b$ | corrugation depth, | (m) | *av* | average |
| $c$ | specific heat | (J kg$^{-1}$K$^{-1}$) | *devel.* | developed |
| flow | | | | |
| $d_h$ | hydraulic diameter, $2b/\Phi$ | (m) | *ext* | external |
| $f$ | Fanning friction factor (Eq.) | | (–) | $f$ |
| | fluid | | | |
| $k$ | turbulent kinetic energy | (m$^2$/s$^2$) | *i, j, k* | directions |
| $L$ | characteristic channel length | | (m) | *lam* |
| | laminar | | | |
| $l_\varepsilon$ | length scale | (m) | *m* | mean |
| $Nu$ | Nusselt number | (–) | *t* | turbulent |
| $p$ | pressure | (N/m$^2$) | | |
| $Pr$ | Prandtl number | (–) | | |

| | | |
|---|---|---|
| $q$ | heat flux | (W/m$^2$) |
| $Re$ | Reynolds number, $u_{mean}d_h\rho/\mu$ | (–) |
| $T$ | temperature | (K) |
| $t$ | time | (s) |
| $U$ | overall heat transfer coeff. | (Wm$^{-2}$K$^{-1}$) |
| $u_i$, $u_j$ | velocity components | (m/s) |
| $\alpha$ | heat transfer coefficient | (Wm$^{-2}$K$^{-1}$) |
| $\beta$ | chevron angle | (°) |
| $\delta_{i,j}$ | Kronecker delta | (–) |
| $\varepsilon$ | dissipation rate | (m$^2$/s$^3$) |
| $\lambda$ | thermal conductivity | (W/m$^{-1}$K$^{-1}$) |
| $\rho$ | density | (kg/m$^3$) |
| $\mu$ | dynamic viscosity | Pa s |

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# IMPORTANCE OF FORM AND PLACEMENT OF HEATING ELEMENTS FROM THE ASPECT OF COMFORT, ENVIROMENT, HEALTH AND ECONOMY

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## ABSTRACT

People want to have good conditions in their living and existing places. Therefore heating and cooling methods are developed according to different application areas, different heat transfer principles. When we heat a place, the suitability of the placement of heating element and applied heating method need to be examined from the aspect of the factors like effect on human health, occurred comfort conditions and economic etc. according to design and operation cost. In this paper heating methods were evaluated on selection of heating systems and heating elements considering on the such factors as usage place, energy economy, environment, health, comfort and economic.

KEYWORDS: Heating systems, Placement of heating elements, Energy Economy

## INTRODUCTION

Energy generation is one of the most important factors for communities to keep on their lives. In the global sense, the issue of increasing energy cost, supply and usage has gained great importance. The most economical way of using limited sources which exist on the earth and the conscious way of consumption energy must be done. From this point of view, heating has the greatest ratio in energy consumption. Applications like heating of the places through the surfaces of the structure elements like floor, ceiling and wall were carried out in the ancient ages. As the first time in history, the oldest remains are left by the South Western Anatolian civilization in 1200 B.C. it is known that at those times in China and Tibet, about in the year 80 B.C. in Rome, much later than that in Russia and in many of the North European countries, heating was used. An old application about floor heating "Hypokaustenheizung des Römerkastells Saalsburg" is shown in Figure 1 [1]. Heating is carried out by the distribution of the chimney gases of the fireplace or a furnace trough the walls and the floor of the location by the help of the canals.

There are three basic reasons to consider and study about heating. The first one is that the most important energy-containers coal, oil and natural gas will be exhausted in a short time. The second one is that, the carbon dioxide that is left into the atmosphere after the consumption and combustion of those has been lessened by the protocol of Kyoto and the must to protect the national climate, and the third one is the ongoing increase in the consumption of energy due to the increase in the world's population. New technologies are being developed in order to have the best benefit from the energy used in heating and to have the comfort conditions in the cheapest way.

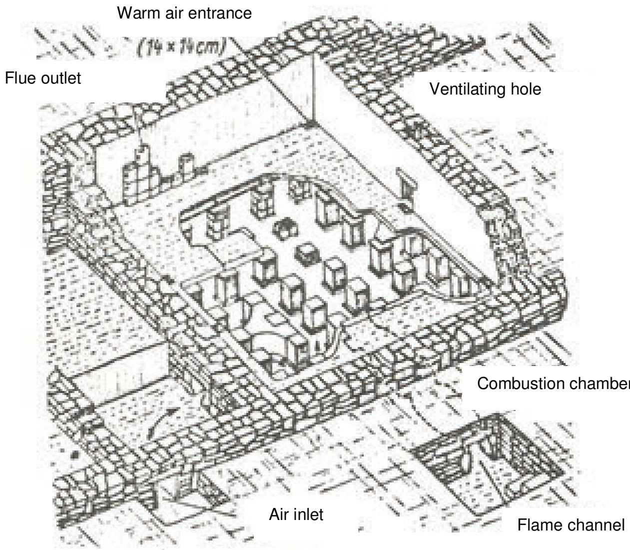



Figure 1. An application on heating, Old Roman Age

The methods and materials used for heating should be evaluated due to comfort, health and their interaction with the environment. The role that is played in each of the expenses during investment and operation will show us whether it is applicable or not by decreasing the energy used in heating and the heating element of the method. On the other hand the shape of the volume that will be heated, is also important. As examples, we can give houses, which are continuously used, hospitals and schools, which contain a big amount of people in them, places in ice-skating centres where audience sits and airplane hangars and factories which have a really high roof.

Heating is, as everybody knows, keeping the temperature of the inside of the volume, between some certain values where a certain amount of comfort is necessary against the natural conditions where it is impossible for the man to live. Also the shape of the activity that people show is important for the heating comfort. Whether due to the height in the volume or the temperature difference which is between the inner and the outer mediums and which is constant, are the elements that affect the heating comfort and also economical usage of energy. Heat transfer is carried out in three ways due to their usages in the heating systems [2,3]: 1) Conduction heat transfer, 2) Convection heat transfer 3) Radiation heat transfer.

**HEATING SYSTEMS**

The heating systems which are used today are classified into three groups according to heating types.

**Traditional Systems**

Heating is carried out by convection. The hot air, steam or hot water gained by a central source, is carried to the place where it is going to be heated by circulation pumps, fans, pipes/canals. The first purpose here is to heat air of medium. And for the second, the people and the machines etc get heated. For the heat generation, there must be heating central or boiler house. It is so hard to do a local or a regional heating.

**Radiation Heating Systems**
All objects radiate rays due to their surface temperatures. Those waves are infrared waves, when they hit the objects or the heating surfaces, the molecules in the objects are set in motion and then heat is got out. The infrared waves can't be seen with naked eye and move directly through the air without heating it. The energy in the waves is absorbed by the cold surfaces. While some of the waves are absorbed by the surfaces and objects, some are reflected back. In that heating system, heat transfer is carried out by radiation.

The approximate ratios of the heat transfer types which are effective in the heating systems are shown in Figure 2.

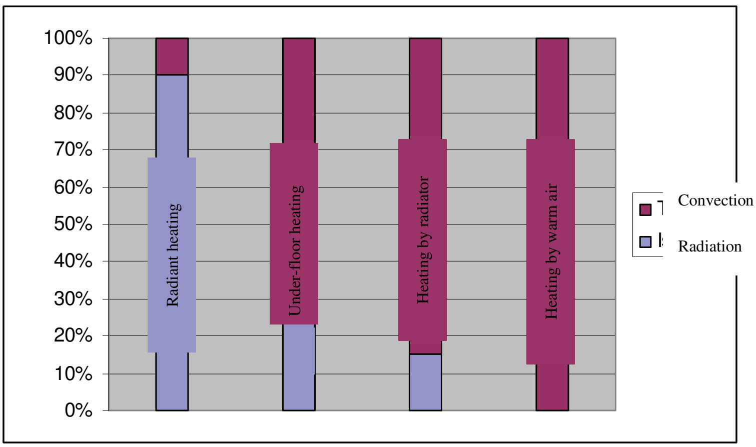



Figure 2. The approximate ratios of the heat transfer types in the heating systems

In general, radiant heating systems are used in museums, big bazaars and shopping centres, schools, factories, industrial centres and complexes, workshops, auto services and showrooms, gyms, mosques, supply depots, plane hangars, greenhouses, animal farms.

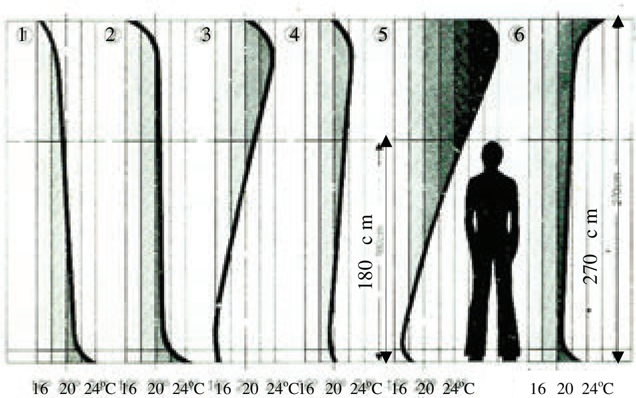
*Table* 1. Heating systems

| Heating Systems According to Heating Types | | |
|---|---|---|
| Central Heating | Local Heating | Regional Heating |
| Hot Water Heating (Between 90 $^{o}$C- 110 $^{o}$C ) Superheated Water Heating (Between 110 $^{o}$C-190 $^{o}$C ) Low Pressure Steam Heating (1,5 bar, 110 $^{o}$C) High Pressure Steam Heating (>1,5 bar, >110 $^{o}$C) Vacuum Steam Heating (0,05 bar – 0,75 bar,>110 $^{o}$C) | Heating is carried out by the direct heating equipments | Buildings more than one are heated by a single heating centre |

<table>
<tr><td>Hot air</td><td colspan="2"></td><td></td></tr>
<tr><td colspan="4">Heating Systems According to Their Heat Sources</td></tr>
<tr><td>Solid Fuelled</td><td colspan="2">Liquid Fuelled</td><td>Gas Fuelled</td></tr>
<tr><td>Wood<br>Coal ( mine- lignite )</td><td colspan="2">Fuel-oil<br>Diesel</td><td>Natural Gas<br>LPG<br>NPG</td></tr>
<tr><td colspan="4">Heating Systems According to Heat Transfer Type</td></tr>
<tr><td colspan="2">T traditional Heating Systems</td><td colspan="2">Radiation Heating Systems</td></tr>
<tr><td colspan="2">Convectors, hot air fans, air conditioning units</td><td colspan="2">Low density radiant heating systems<br>Medium density radiant heating systems<br>High density radiant heating systems</td></tr>
</table>

## HEAT TRANSFER MECHANISM AND TEMPERATURE GARDIENT IN THE VOLUME

Heat transfer mechanism is fulfilled by the direct convection from the radiators with an approximate temperature of 80 °C to the volume in the heating systems with traditional radiators and in the floor heating systems by convection from the floor with the temperatures between 25-80 °C to the volume, from the radiators to the volume the radiation ratio which can be neglected, becomes about 50% of the total heat transfer in the floor heating. As a result of this; the temperature difference between the inner walls and inner surfaces of the outer walls can be kept about 1-2 °C. In Radiator heating systems, this temperature difference is about 6-7 °C. The heated air moves to the upper sides of the volume. The places of the volume with the highest temperature in the floor heating systems are the surface of the floor. The relocating hot air from the floor will get colder and colder and there will be a decrease in the movements of the air partials in the upper sides of the volume. The heat gradient volume cross- section for different heating systems is shown in Figure 3.




1 Ideal heating

2 Floor heating

3 With radiator heating In the inner wall

4 With radiator heating In the outer wall

5 Heating with warm air

6 Heating from the Ceiling

Figure 3. The heat gradient volume cross- section for different heating systems

Heating from the floor and ceiling gives theoretical temperature distribution that is accepted as ideal.

**THE POSITION OF THE HEATING ELEMENTS IN THE VOLUME**

While choosing heating systems and heating method the fact like volume, using type, heating volume function, health, energy economy and comfort are taken into consideration. The positions of the heating elements in the volume can be like being hanged on the walls, being positioned in the ceiling from the inner and outer sides.

In the radiator, the air comes from downside and goes upward while getting heated. As shown in the Figure 4, the air flow is carried out only by convection. So the efficiency of the heating radiator just depends on the air flow. The thermal efficiency of the radiators is carried out due to the European control norm EN 442 and when the radiator is 110mm high from the floor. But, in the real applications according to the application locations and the aesthetic choices, those measurements that are advised at the standard are not applicated in practice.

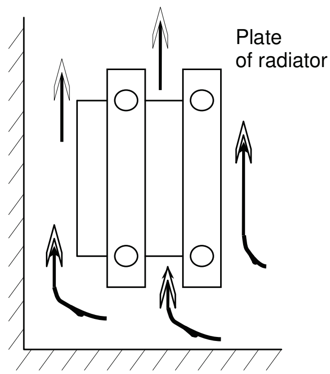



Figure 4. Operation principle of the radiator core

Çopur et al.[2] have done some researches about the affects on the energy economy, the protection of environment, and thermal comfort, and have compared the temperatures of the waters in the heating systems by doing the necessary calculations and have determined the temperatures of the installations which are required. Dağsöz et al. [5] have done researches about the problems and lacks which are faced in central heating systems and have decided that in order to do some economy in there should be made some new corrections. Tiryaki [6] and Inam [7] have researched the factors like, the wave lengths of the devices according to temperature, the length of the device, the losses in the chimneys, the efficiency of the device and the properties of the device while making a choice in devices and how economic using the radiant systems is, in projecting the radiant heating systems and have shown that it is so important that care should be taken in choosing the radiant , and advantages can be obtained when the radiant heating systems are used in high buildings like factories and hangars.[6,7,8]. Seifer and Richter [9] have taken the room of a house with low energy whose thermal need is 485 W, and have made energy analysis of how to locate the heating devices according to the positions of the walls and researches about the affects on the thermal comfort, and they have determined that when the heating device is located in front of the windows, at the outer walls and the inner walls, and the rates of air-exchange is important in determining the quality of the heaters, and that the heaters which are located at the outer walls can provide the lowest heat capacity.

In order to determine the affect of the distance between the radiators of the central heating system and the floor on the ratio of convection, Gu [10] has improved two methods, so he planned to give the convection ratios of the radiators the most suitable shape that they could have. He determined that the If the distance between the radiator and the floor is small than the heating capacity of the radiators will be small and after a certain distance he found out that the value becomes constant.

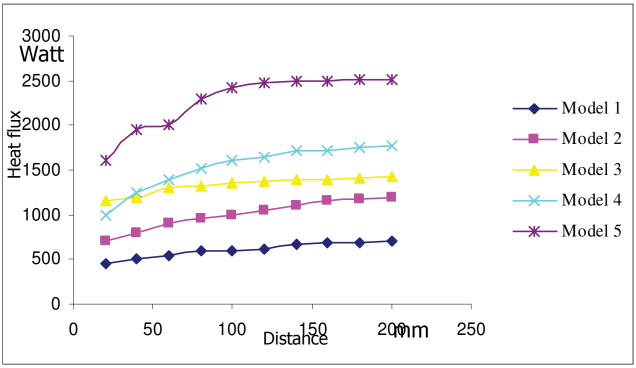



Figure 5. The influence of distance between the radiator and the floor on heat flux

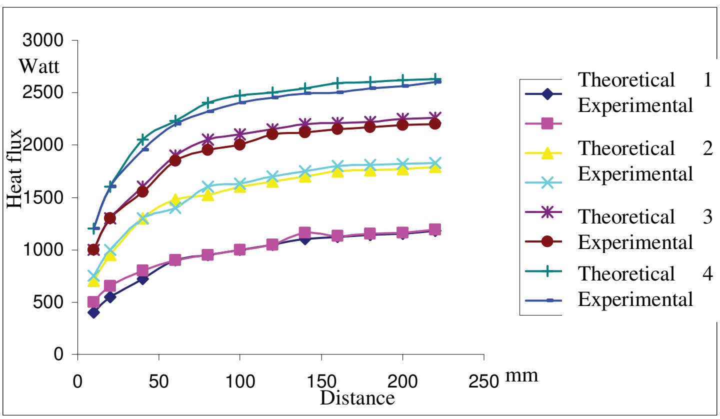



Figure 6. Theoretical and Experimental Results of Effect of Floor Distance on Radiator Core

In the radiant heaters in which the heat is transformed by radiation, the temperature changing can be about 1-2°C due to the floor height in the design where the 20°C inner temperature was taken as the essential point.

In the systems operating with radiator which is working with air and hot water from the traditional heating systems, a temperature difference of 5 °C on the floor whose volume height is 8-10 m in the same design temperature can be like 15-25 °C through the ceiling. This sort of conditions effect the increment on energy consumption and it also causes the discomfort in living medium.

The purpose of this study is to determine the main factors in designating the heating style while choosing the heating system, by taking into consideration the points like the usage style of the volume, heating volume function, health, energy economy and comfort. The situations of the heating elements in the volume can be like being hanged on the walls, being put on the ceiling from inside to outside, being hidden in the floor etc.

**COMPARISON OF HEATING SYSTEMS**

In the heating by traditional radiators, volume air is heated primarily. Then this air is separated in the volume by convection. In the traditional heating systems, the floor temperature is lower than the comfort temperature. In Figure 7, traditional heating systems and radiant heating systems are compared due to their energy economy and temperature layering.

The temperature in the regions near the ceiling is higher because of the upward movement of the heated air. This causes the increase in the thermal losses that occur by the air circulation in the volume and heating the unnecessary space which is not used.

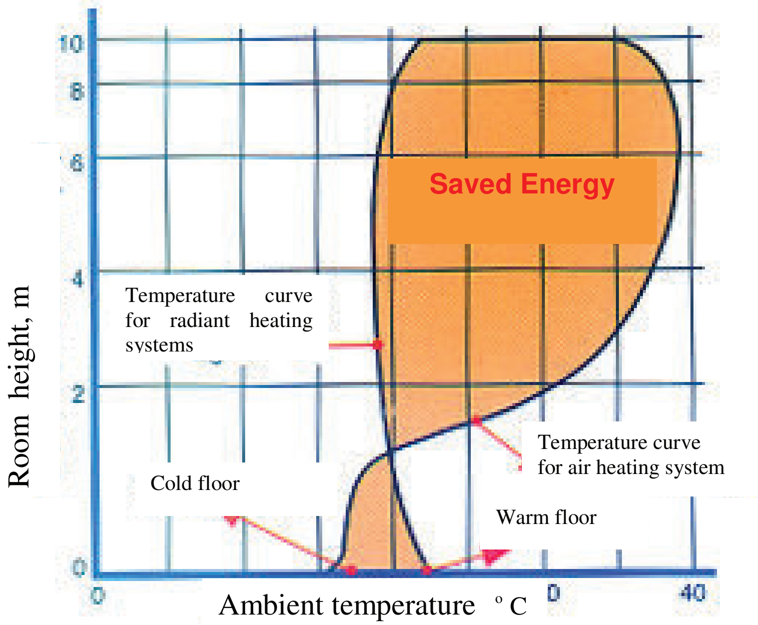



Figure 7. The comparison between the traditional heating systems and the radiant heating systems

But in the radiant heating the floor temperature is higher than the comfort temperature and on the other hand in the regions near to the ceiling the temperature value is dramatically lowers than the value in the traditional heating systems. For this reason this causes a significant energy economy.

**The Forming and Positioning Condition of the Heating Elements**

The opinions about the positioning the heating elements at the inner sides of the volumes are becoming widespread [10]. Energy circumstances like physiological ones are being evaluated by doing researches and based on computer simulation program. Due to the obtained results, suggestions for the ratios having different air changes are listed in literature.

**The heating regions of the radiant heaters**

In the heating system applications, it is possible to heat all the sections or a part of them in the region by positioning the heaters in a suitable way. As it is important how economical the heat is produced, how homogeny the heat is separated into the region is also important.

As the air heated will go up by having a lower density, the places where the temperature is high are the places which are close to the ceiling and as the temperature is low in the places with low height which should be heated, there exist energy losses. Just as the height of the ceiling and the change in air flow have a significant importance on energy economy, in the same way the positioning of the heaters in a suitable way has on effect on energy economy, too.

If a radiator gives the same amount of heat in the same conditions with small surface, then its thermal efficiency is higher. Because of their aesthetic outlook, panel radiators are widely preferred. Steel radiators with smooth surfaces are preferred because of their properties like dust catching and easiness to be cleaned.

**The Air Conditioner Modules**
The air conditioner modules used in recent years has the principle of heat transmission caused by the radiation which occurs between the hot and cold surfaces. Here, heat moves from hot surfaces to cold surfaces. The cold object gets warmer by receiving heat. By the air conditioner modules, directly the air is heated up. Especially, under the modules of the air conditioner, the upper surfaces of the floor, the walls, the tables and the furniture. But, in counterpart to that, in the event of "cooling" the process is just functioned just in an opposite way to this. The air conditioner module has the duty of energy supplying. The basic assumption is to have a constant inner volume temperature and is to obtain a volume air conditioner. It is possible to evaluate the advantages of the air conditioner in some parts as their affects on health, economy and environment.

In the evaluation of them considering health, we can list the entries below:
- Maximum hygiene conditions are obtained.
- Much higher temperatures of wall and floor can be obtained.
- Maximum rate of comfort should be reached.
- As there is no air flow, a very healthy medium can be formed.
- There is no noise at all which can be called as "disturbing"
- The best physical conditions can be obtained.

The following advantages can be listed considering their economical direction
- It is possible to use the volume economically and also gain space.
- It is possible to have homogeny heat dispersion in the volume.
- Energy economy can be made.
- Less material expenses are used.
- Less operation expenses are used.
- It is possible to have the cooling and the heating in the same system.
- Has a short time period of amortization.

And the environmental advantages can be listed as
- Energy economy can be made.
- With the energy possession made, the environment can be less polluted much.
- The natural energy sources are consumed with protection.

The comparison made between the traditional heating systems where heat is transferred by convection and conduction and heating systems where heat is transferred by radiation for economy, interaction with environment and comfort, is given in the Table 1 [12]

*Table* 2. The comparison between the traditional heating and radiant heating methods

| Traditional heating methods | Radiant heating methods |
|---|---|
| When the heating system is on the floor, there can be losses from surfaces | All the surfaces that provide heating participate in heat transfer |
| Heat transfer is carried out by convection and conduction | Heat transfer is just carried out by radiation |
| Especially in the heating systems working with hot air, as fans are used, circumstances for comfort cannot be obtained | In the dusty regions, no movements of dusts exist which can ruin the comfort circumstances |
| System works on the heating of the volume | System Works on the heating of the surface |
| Heating is dense in the regions near to ceiling | Heating can be done for required place. |
| In order to reach the comfort condition, long time should be passes | Comfortable conditions are obtained in a short time |
| Because of the transportation of the heat, so many thermal losses exist | As the transportation of the heat is carried out by direct radiation, there exist no thermal losses at all |
| Can not get adopted to the improvements or changes in the regions which will be heated | can be reassembled due to the improvements or changes in the regions which will be heated, the present system elements can be removed |
| Heating system and elements can not be disassembled and used in an other heating region | Radiant heaters obtained from different volumes can be used in the regions which are wished to be heated |
| Because of the mechanic parts like pup, hot air or water devices, cauldron and burner; a maintenance is often necessary, there can be dangers like leaks, dirtiness and fire | There are no mechanical parts at all. Operation and maintenance is not often necessary |
| It's general efficiency is about 80% | It's general efficiency is about 50% |

**Conditions Related to Insulation**

The results of the isolation applications can be evaluated in three directions. The first one is that it makes increments on the energy transformation efficiency and energy economy. Because of this, heat losses can be lowered and the amount of the material that will be used for heating and the capacity of radiator and cauldron will decrease. The expenses for the first investment of the installation will decrease and beside that, as much less fuel will be consumed, there will be economy on spending. Secondly, there will be a decrease in the environment pollution. Thirdly is the thermal comfort obtained by application.

Windows and the doors with glass are the structural elements which have the highest value of thermal loss. But beside that, on sunny days, there can be a thermal gain through the windows. By using a window with double glass, the thermal loss can be decreased to minimum. In order to decrease the thermal losses that happen by the air leak through the opening parts of the doors and the windows, some applications like gasket and the thermal loss are also decreased when the shutters of the windows are closed in the evenings, and by placing self-closing mechanisms on the outer doors or positioning revolving doors, decreasing the time period of the door.

The isolation of rear side of radiator which is the thermal source in the room has a significant importance in decreasing the thermal losses. When the room temperature is 20$^{o}$C, the back of the radiator can be 45$^{o}$C. Because of the aesthetic reasons, when the front of the radiators is covered, that temperature value is much more increased. According to the laws of thermodynamics, the more difference there is between two mediums the more heat transfer occurs. That means, the most and the fastest heat transfer from the volume is carried out in the back of the radiators. The most suitable way is to put an isolating material in the walls that are located at the back of the radiators. But in the buildings which were not constructed like that at the construction period, the isolation for the backside of the radiator can be easily carried out by the polyethylene sheets aluminium covered. Beside their property that they are good isolators, it is again a big advantage that they can reflect the heat into the volume. After realizing those applications, in the residences heated with individual heating or central heating by holding the volume temperature same, there can be decrement in the fuel consumption by 5 %. In the central heating system, the expense for the heating does not change when an isolation made for a single residence is carried out but a higher level of comfort in heating can be obtained with a temperature increase of 3 %.

The condensation on the inner surfaces of the volume walls, come into existence if the inner surface temperature which is in touch with the medium air that is a mixture of dry air and steam, gets lower than the condensation temperature of the steam. The heat transfer coefficient k* which will prevent the structure material from facing a condensation is detected by the formula below.

$$k^{*} = \alpha_{inn} \frac{T_{inn} - T_{con}}{T_{inn} - T_{out}} \qquad (1)$$

In this way, the total heat transfer coefficient k for the wall between the inner medium and outer medium can be found.

$$k = \frac{1}{\frac{1}{k^{*}} - \left( \frac{1}{\alpha_{inn}} - \frac{1}{\alpha_{out}} \right)} \qquad (2)$$

Condensation temperature $T_{con}$ can be designated according to the relative humidity value and the $T_{in}$, the inner temperature of the air in the heating volume.

$$\varphi = \frac{P_{b}}{P_{bD}} \qquad (3)$$

For this, the saturated steam pressure value $P_{bD}$ which matches the inner temperature value is read from the table and the partial steam pressure in the mixture of dry air and water in the medium is obtained by multiplying it by the relative humidity value $\varphi$. The saturation value (dew point) which matches that designated pressure value is read from the table. If there is a different humidity percentage between the outer and inner mediums of the outer walls or because of the formation of condensation in the wall material, there can be a humidity transfer which is similar to heat transfer. In order to prevent that, it is suggested to use a steam-prevented with isolation material.

Considering the international and national standards of the thermal losses in the buildings, the thicknesses for the isolation are determined. According to that, the approximate value should be or less than 60 kWh/ ($m^2$ year) for the unit area of the building.

## CONCLUSION

Every single day, it is becoming more and more important to live in cheaper way and in a better comfort. For this, some new alternative systems have been developed and the problems in the present system are being tried to be solved.
We want the mediums where we are in, to be suitable to our physiological situation. If the weather temperature, the velocity and humidity of the air, air change rate, the pureness of the air (odour and the hanging particles in it) and the control of the air motions in the volumes used, are disturbing the people, the properties of the using heating system should be revised and the defects in the application should be corrected. For example, the extreme high and extreme low air temperatures in the place where we live, affect our lives directly.

On the other hand, the heating system should be economical by the means of low consumption rate, and the economical usage of the energy. The savings made for the fuel and energy can contribute to the home and national economy. It is important to give the answer to the questions as between which temperatures and in what kind of a place the work will be carried out, which heat transfer type will be suitable to that type, and beside the suitable types, which heating system will be used, and the points where the heating elements will be located in the heating system. Because by making a good decision in choosing the heating system and locating the system elements in the place in a correct way, comfortable life and the life quality increase and the costs on investment expenses will decrease and energy and savings will be made.

Saving energy and lowering the heat losses should be considered and should be related to building a new power plant with the way of making large economy.

**SYMBOLS**

| | |
|---|---|
| $\dot{q}$ | Heat flux (W/$m^2$) |
| $\lambda$ | Thermal Conductivity (W/mK) |
| A | Area ($m^2$) |
| $\alpha$ | Convection coefficient (W/$m^2$K) |
| T | Temperature ($^oC$) |
| $\varepsilon$ | Emissivity $(0 \le \varepsilon \le 1)$ |
| $\varphi$ | Relative humidity |
| k | total heat transfer coefficient (W/$m^2$K) |
| $\sigma$ | Stefan-Boltzmann constant (5, 67 x $10^{-8}$ W/$m^2$) |
| $\dot{Q}$ | Heat flux (W) |

*Indices*

| | |
|---|---|
| x | Toward of x axis |
| inns | Inner surface |
| outs | Outside surface |
| $\infty$ | Outer medium |
| conpo | Condensation point |
| b | Steam |
| bd | saturated steam |
| cd | Cold |
| ht | Hot |

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# AN EXPERIMENTAL STUDY ON SOLAR ENERGY STORAGE

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## ABSTRACT

Using solar energy for heating of buildings or supplying hot water when needed by storing solar energy under ground was investigated experimentally. For this aim, a heating system was built to store solar energy in cylindrical water tank placed under ground via solar collector.

In the present study it is obtained that energy saving was seen remarkably in Edirne in Turkey ($41^o$ 39' 54'' N) by storing solar energy to use it at other time considering that it would balance difference between consuming and generating energies.
During the experimental study, temperature of heat storage unit and surrounding under ground were measured. In the season of heat storage, efficiency of heat storage unit of cylindrical tank was obtained as 86-96 % and it was determined that 89-100 % of total heat was obtained from the sun.

## 1. INTRODUCTION

Nowadays the importance of renewable energy sources increased sharply due to increments of energy consumption. Energy needs are variable due to usage in industrial plants and in buildings. There are many investigations on limiting the energy consumption saving the energy needs in building sectors and many studies have been carried out to increase the renewable energy usage rate. Solar energy usage that means low temperature heat storage system usage increases remarkably in the way of method of specific heat capacity and method of transformed heat usage.

There have been many studies for years related to energy storage with underground heat. (Shelton, 1975), investigated theoretically the thermal relationship between the non-insulating semi-spherical heat tank and geological structure around the tank under a building. Problem of semi spherical energy tank placed underground was solved analytically and numerically. (Lund and Östmann, 1985), developed a numerical model to calculate performance of heat storage system of building heating by using heat pump in heating season. They have used the cylindrical heat exchanger placed underground by using solar energy. (Brunström et al., 1985), built a system and made the measurement on his system that was built in Sweden – Lykebo on heating system which contains the solar collector, rock cave, residence and auxiliary heating system.

(Lund and Pentolo, 1987), have made measurement on heating system at Keravo Solar Village built in Helsinki at 1981-1982. (İnallı, 1996), carried out theoretical analysis of solar heating system with cylindrical tank placed in underground. Temperature distribution was taken as two dimensional around the tank. Heat transfer problem was solved by using finite difference method and finite Fourier Transformation technique. (Nordell and Hellström, 2000) investigated the performance of seasonal storage of solar energy used in low temperature heating of building. They have used Trnsys and Minsun computer software program for this purpose. (Ünsal and Yumrutaş, 2000), carried out on analysis of solar assisted heat pump heating system by using spherical tank. (Schmidt et al., 2003), planned and used seasonal storage central solar assisted heating system in Hamburg at 1995. More than 50 % of annual space heating and water requirement were supplied from solar energy by seasonal storage.

In the present study, an experimental rig was built for the system of seasonal storage of solar energy. Heat requirement of system was 1.99 GJ/month for residence heating and hot water requirement. When space heating is not needed then solar energy was used for hot water usage and rest of energy is also stored in heat storage unit and its surrounding. Temperature distributions were determined experimentally in the volume of solar energy storage system and its surrounding as daily and monthly for the period of July, August and September. Temperature distribution and value of heat transfer were monitored and investigated by changing the daily hot water requirement.

## 2. EXPERIMENTAL STUDIES

### 2.1. Experimental Setup

Experimental rig contains two parts which are heat storage and heat collector units. Heat storage system was located underground. Figure 1 shows the schematic view of experimental rig.

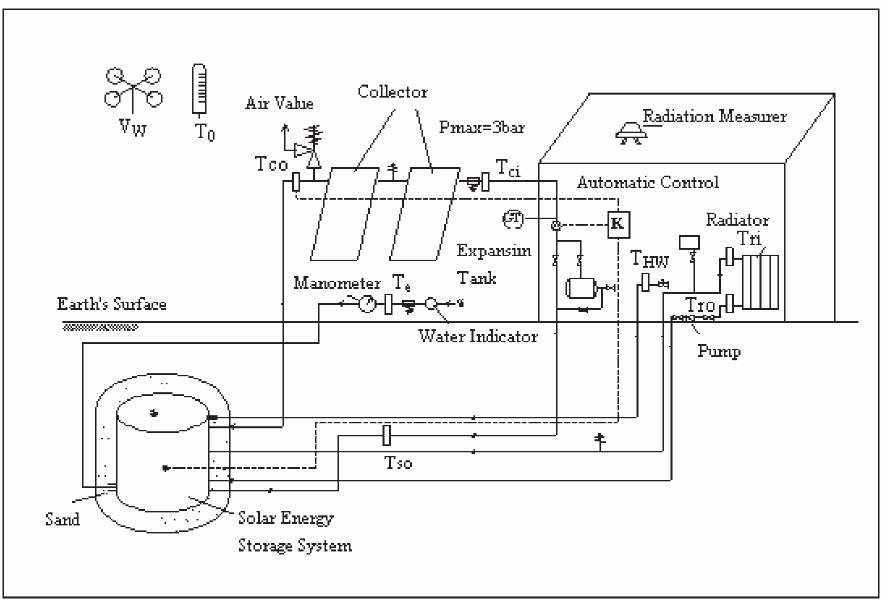


Figure 1. Schematic Overview of Experimental Installation

### 2.1.1. Heat Collector Unit

Available solar radiation that comes to solar collector, was determined with the method of Ø. Ø is the monthly average availability. Ø is defined as ratio of total radiation on critical radiation coming monthly on the collector to coming radiation to this surface (Hottel and Willer, 1953). Monthly available solar radiation for unit $m^2$ of collector and necessary collector surface area were determined. Parameters of solar collectors are given in Table 1.

Table 1. Parameters of solar collectors

| | |
|---|---|
| Location | Edirne |
| Collector Slope | 41° |
| Collector Type | Vacuum piped |
| Absorber Surface | $A_c$=4 $m^2$ |
| Optic Efficiency | $\eta_{op}$ = % 83,7 |
| Heat Loss Coefficient | $k_1$ =1,75 $W/m^2K$, $k_2$= 0,008 $W/m^2K$ |
| Collector Working Fluid | Ethylene Glycol-Water |
| Concentration Rate | % 25 |
| Control of collector operation | $\Delta T_{on}$ = 4 K, $\Delta T_{off}$ = 2 K |
| Recording Data | $\Delta t_{data}$ = 1 h |

### 2.1.2. Heat Storage Unit

Solar energy storing unit contains the cylindrical tank and the sand filled bulk around tank. Total diameter of tank (with sand filled shell) is 1350 mm and its height is 1400 mm. Solar energy tank diameter is 650 mm and height is 700 mm, it was built with given parameters in Table 2. Solar energy tank was insulated with 50 mm thick insulation material, connection tubes were insulated with 9 mm isolation material that was rubber foam material. Elements of solar energy tank were given in Figure 2.

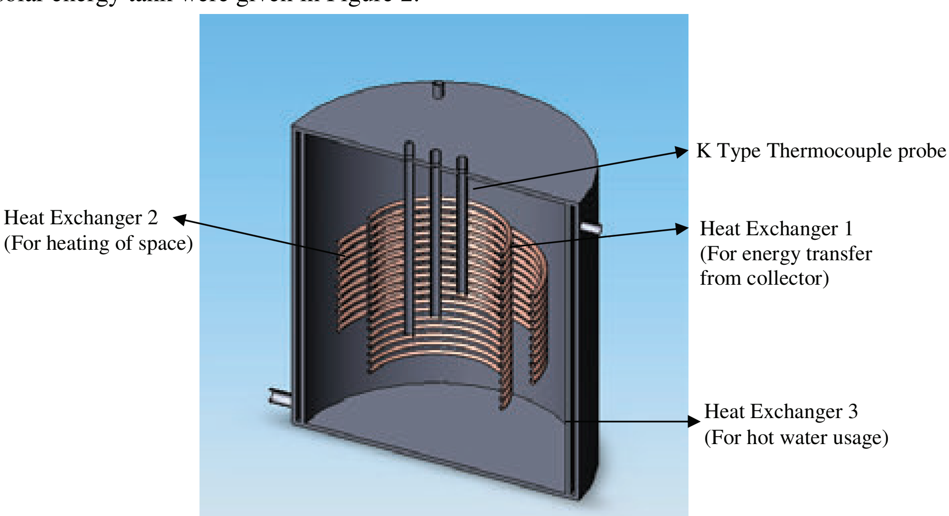


Figure 2. Solar Energy System

Collected energy by flowing fluid coming from solar energy collector is transferred heat storage unit placed in underground. This is transferred to the solar energy tank placed in heat storage unit. Heat storage material in solar energy tank is water. As can be seen in Figure 2, energy transfer is supplied via heat exchanger 1. Heat storage unit works automatically according to the temperature difference between the average water temperature in the heat storage tank and mixture of 25 % ethylene-glycol-water average temperature coming from collector.

Storing heat from seasonal solar energy is used for only hot water usage which is supplied from heat exchanger 3. Heat exchanger 2 is used for domestic space heating from stored solar energy. Radiator water circulates in heat exchanger 2 and pump was used to circulate water.

### 2.2. Measurements

Three groups of measurements were carried out. First one is related to collector. Collector outlet temperature $T_{co}$, collector inlet temperature $T_{ci}$, outlet temperature of working fluid flowing in collector from solar energy tank $T_{so}$ was measured respectively.

Secondly measurement of coming radiation to horizontal surface I, ambient temperature $T_o$, wind velocity $V_W$ values were monitored.

Thirdly, solar energy tank and its surrounding temperatures were measured. Locations of measurement point were given in Figure 3 and Table 2. Table 3 shows the total measurement values. Measuring device and accuracy of them were given in Table 4.

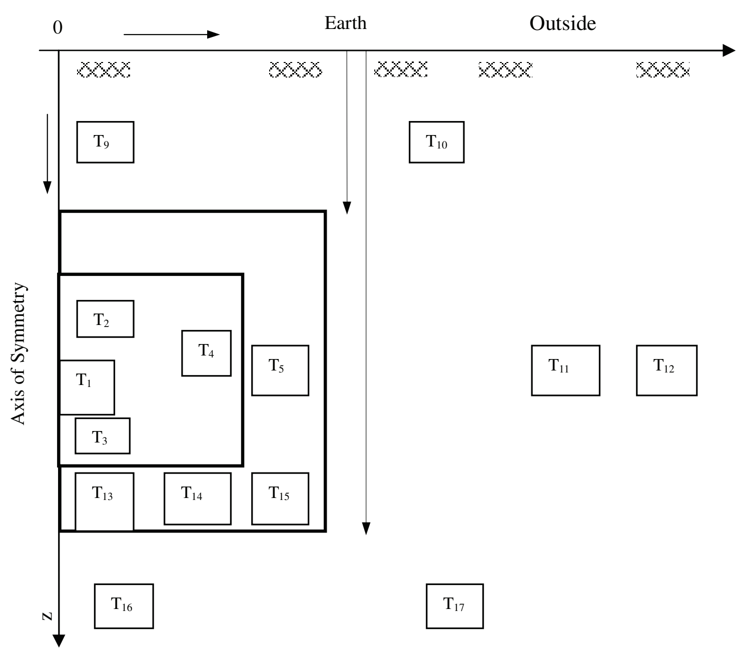


Figure 3. Measurement Points on Solar Energy Tank and its Surrounding

Table 2. Coordinates of Measurement Points

| $T_x$ | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| r (cm) | 0 | 6 | 6 | 29 | 52 | 20 | 30 | 55 | 6 | 75 | 85 | 100 | 20 | 30 | 55 | 6 | 75 |
| z (cm) | 110 | 95 | 117 | 104 | 110 | 55 | 55 | 55 | 25 | 25 | 110 | 110 | 165 | 165 | 165 | 195 | 195 |

Table 3. Parameters of heat exchanger in solar energy tank

| | Heat Exchanger 1 | Heat Exchanger 2 | Heat Exchanger 3 |
|---|---|---|---|
| Material | Copper | Copper | Galvanized sheet |
| Type | D= 8 mm, Axial D= 300 mm, 42coil, w = 5,7 mm | D= 10 mm, Axial D= 440mm, 26 coil, w = 5,2 mm | Storage tank with double wall $D_h$=20 mm |
| Fluid for transferring of heat | % 25 Ethylene Glycol- % 75 Water | Water | Water |
| Mass flow rate | 0,03 kg/s | 0,044 kg/s | 0,07 kg/s |
| Specific heat capacity | $C_p$ =3,821 ( kJ/kgK)* | $C_p$ =4,179 ( kJ/kgK)* | $C_p$=4,18 ( kJ/kgK)* |

* Specific heat capacity of fluids were determined according to the mean inlet and outlet temperatures of fluids

Table 4. Used Measurement Devices and Features

| | |
|---|---|
| Data Logger | 10 channels measurement, 5 of them is analog and 5 of them is digital, accuracy of $10^{-4}$. |
| Radiation measurement Device | 11,47.$10^{-6}$ µV/W/m$^2$ accuracy |
| Digital Thermometer | Measurement Range: -50 to 1300 °C, 0,3 %, ±1 °C accuracy |

## 3. THE EVALUATION OF RESULTS

Measurements were started on 11.07.2005. Total energy transferred to heat storage unit was 1338 MJ for the period of July. Amount of 456.6314 MJ of this energy is extracted with hot water for usage. 473.9417 MJ and 356.7340 MJ were stored by water and sand respectively.

In the period of August, 1995 MJ energy was transferred to the heat storage unit. 469.358 MJ of energy was extracted with hot water for usage. Rest of 827.042 MJ and 459.665 MJ were stored by water and sand respectively.

In the period of September, 1691 MJ energy was taken and 342.7338 MJ was extracted for hot water usage, 711.297 MJ and 402.3257 MJ were stored by water and sand respectively.

In the season of heat storage, values of average temperature of sand, water in the heat storage unit were given in Figure 4 a daily from hourly temperatures of July, August and September.

Generally tank water mean temperature of points 2 and 4 were higher. Because flowing fluid comes in from upside and goes out from bottom of the heat exchanger 1 and heated water moves up to upper points due to free convection. Lowest temperature of water in tank was obtained at the point of 3 . Average tank water temperature values decrease by going down on z direction from the centre of tank to bottom of tank (from the measuring point of 1).

Upper and bottom surface of sand temperatures closest to tank centre at r direction were higher. Because both heat loss is higher from side-walls and is far from heat exchanger as the shape of shell. As tank water temperature increases sand temperature increases being closer to the centre of r direction.

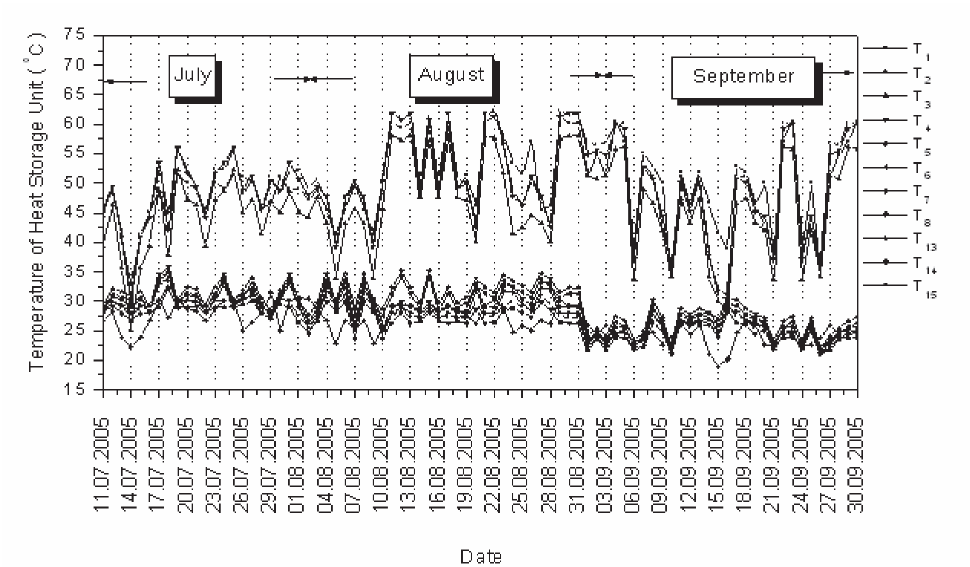


Figure 4. Daily Average Temperatures of Heat Storage Unit at Season of Heat Storage

In the season of heat storage period, underground temperature of cylindrical heat storage unit surroundings can be seen in Figure 5.

Bottom surface of ground temperature of heat storage unit is higher than upper surface of ground temperature due to the wind. Depending on this velocity of wind heat loss increases from upper surface. Higher ground temperatures of side surface were measured at period of August due to energy transfer for hot water usage and heat loss due to increasing average hot water temperature values. Ground temperature of heat storage unit's upper surface reached the minimum value especially at noon time due to increments of ambient temperatures.

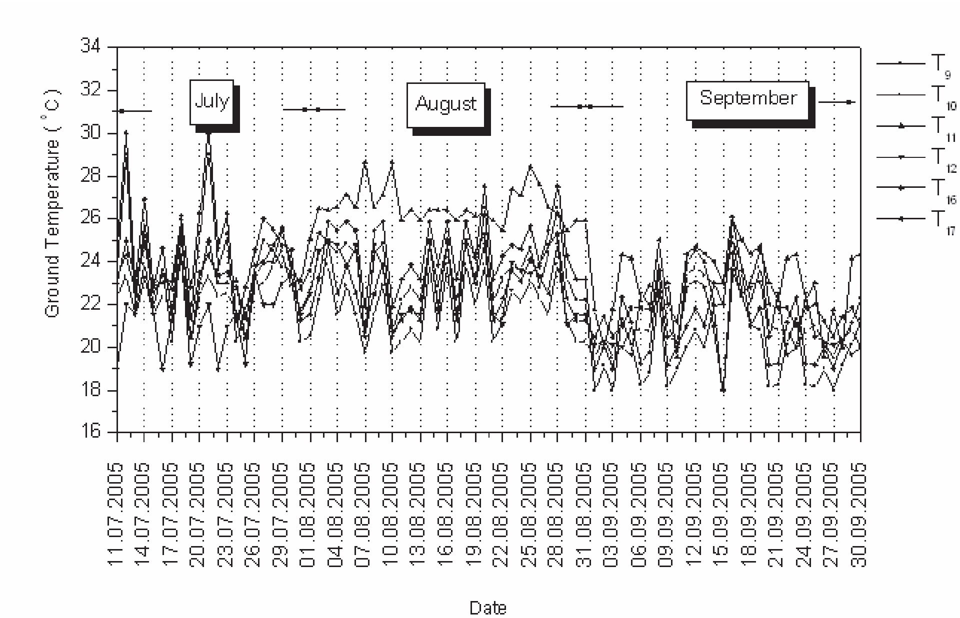


Figure 5. Daily Average Ground Temperatures around Heat Storage Unit

Daily maximum average water temperature and total radiation value coming onto the collector surface are given Figure 6 on solar energy tank at the season of heat storage.

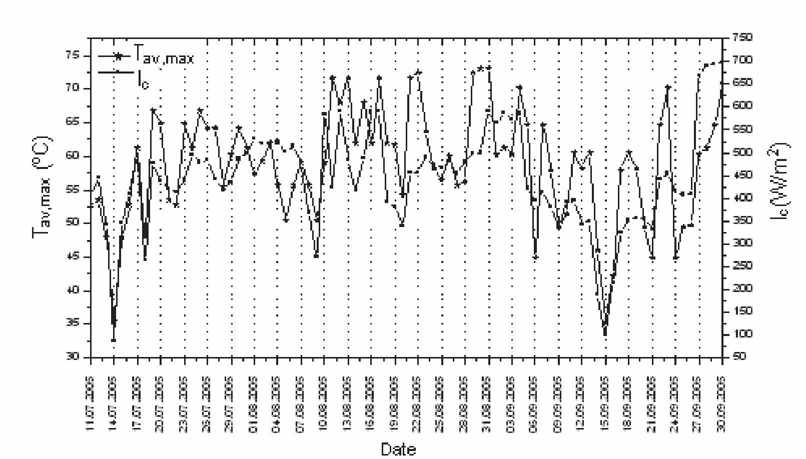


Figure 6. Daily Average Values of Reached Maximum Temperatures at Solar Energy Tank

According to the measurement results, monthly average temperature distributions of heat storage unit and its surrounding were given in Figure 7 at the season of heat storage. The highest average water temperature was obtained at August; the lowest average water temperature was obtained at July. The lowest ground and sand temperatures were obtained in September.

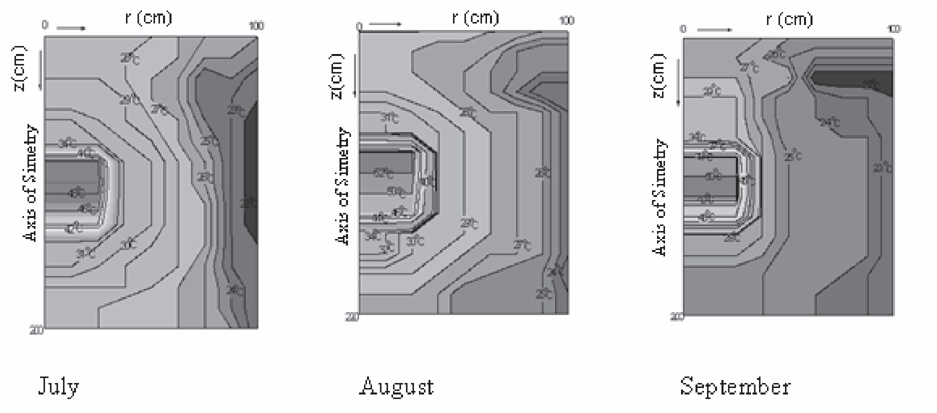


Figure 7. Temperature Distribution for Cylindrical Heat Storage Unit and it’s surrounding

Solar fraction rate, efficiency of heat storage unit and hot water usage and transferred energies are given in Figure 8 for storing seasonal solar energy for July, August and September. Circulation pump power was taken into consideration for net efficiency of heat storage unit.

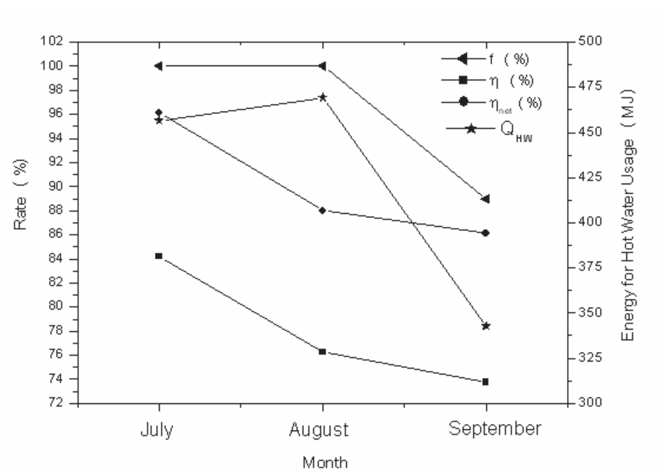


Figure 8. Monthly Solar Fraction Rate , Average Tank Efficiency and Taking Energy with Hot Water for usage

## 4. CONCLUDING REMARKS

Transferring energy to the heat storage unit at seasonal solar energy storage is totally function of cylindrical heat storage unit and its surrounding temperature; furthermore it depends on ambient air temperature, wind velocity, radiation, heat loss and taking energy for hot water usage from heat storage unit.

Total amount of 5.024 GJ energy was transferred in the season of July, August and September. 1.269 GJ of this energy was taken with hot water for usage, 2.012 GJ and 1.219 GJ of this amount were stored by water and sand respectively. This system was built for the aim of seasonal heat storage system by taking into consideration monthly minimal total radiation value of the year and monthly maximum total heat needs of the year. 89-100 % of necessary monthly heat needs in the season of heat storage was supplied from solar energy. Net stored efficiency varied from % 73 to % 83.

**Symbols**

A collector Area
C specific Heat
D diameter
f solar fraction rate
h height
I radiation flux comes on unit area of Collector
K heat loss coefficient
L distance between the heat storage unit to ground.
m mass flow rate
P pressure
Q energy
T temperature
t time
η efficiency
Ø monthly average availability

Subscript and Superscript

o ambient air
c collector
$c_i$ collector inlet
$c_o$ collector outlet
dç tank outlet
$H_W$ used water
$r_i$ radiation inlet
$r_o$ radiation outlet
op optic
p constant pressure
av average
h hydraulic

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# NUMERICAL STUDY OF HEAT TRANSFER CHARACTERISTICS OF EXTENDED SURFACES

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## ABSTRACT

In this study, effects on heat transfer of different type extended surfaces placed on a flat plate on two dimensional rectangular cross-sections are investigated with conjugated heat transfer approach. Surface and air inlet temperature were fixed for the condition of low turbulent flow. Effect of Reynolds number on heat transfer coefficient was also shown for triangular fin geometry. Three type of fin geometries were used which are circular, rectangular and triangular in the present study. For these geometries, temperature contours and local heat transfer coefficients are obtained for Re=500. Triangular fin with groove was also investigated and temperature contour and local heat transfer coefficient were shown. Firstly, problem is solved analytically for flow in a flat channel to obtain the correct results and then these are compared with results of commercial codes Ansys-Flotran and Fluent numerical analysis programs.

## INTRODUCTION

The extended surfaces definition is mostly used for a solid which has heat transfer by conduction within it's borders, as for between it's boundaries and surroundings is mostly used for a solid which has heat transition by convection and radiation. Although there are lots of different states that convection and radiation materialize together, one of the practices which is encountered mostly is the extended surfaces which are used in order to increase heat between a solid and the fluid in its surroundings. On a plane wall in order to increase being carried heat transfer between the wall surface and the fluid in its surroundings fluid velocity is raised and heat transfer coefficient can be increased as related to this and/or fluid temperature can be reduced. Nevertheless even it is being raised to the highest value may not be sufficient to get the desired heat transfer or it may be encountered with high costs. These costs are related to the power of ventilator or pump which is required for fluid act's being increased. By a third option heat transfer can be increased by increasing the surfaces where being carried heat transfer occurs. By using the fins which the fluid in its surroundings expands into it, it can be done from this wall. Thermal conductivity of fin material affects the heat distribution along the fin and heat transfer is also affected by this. Ideally, for heat change's being least from beginning to end of the fin, fin material should have high thermal conductivity. In the event of thermal conductivity's being infinite, the whole fin will be at the same heat with the surface and at the most heat transfer increments will be provided.

In today's technology, by increasing the amount of the heat transfer, energy saving efforts is very much. For this aim, one of the widely used methods is fins (extended heat transfer surfaces). The surfaces with fins raise convective heat and mass transfer by expanding surface area and raising the current of turbulence. The practice areas of the surfaces with fins are very various. Main usage areas can be listed as cooling of gas turbine engines, cooling of turbine blade, and of electronic devices and some heat exchanger used in chemical production foundation and in aviation, airplane. When not using appropriate fin, then heat transfer can be reduced instead of increment of heat transfer. It is required by taking up the fin material, it's type, it's orders of placement, it's assembly forms to surface and each of environment conditions, their being examined and being evaluated in the way that it will increase heat transition.

In heat exchangers in plate type, Tauscher and Mayinger [1] have examined as experimental and numerical the heat transfer enhancement in low turbulent and laminar flow cases with extended surfaces made of different installations. This research has been the beginning point to this research which has been presented. Lee and Abdel – Moneim [2] have searched numerically the flow patterns from horizontal surface and heat transfer for two dimensional turbulators by using CDF model. Acharya et al. [3] carried out the numerical and experimental examination of heat and fluid transfer for fluid area which progresses periodically in channel with fin. Liou, Chang and Hwang [4] and Liou, Hwang [5] have done studies about different fin height and the rates on different inclination for two pairs of turbulence supporter which are mounted one after another in improving channel flow ($1.2x10^4<Re<12x10^4$). Masliyah, Nandakumar [6] obtained heat transfer characteristics in tubes with triangle fin inside by using finite element method. They have determined that there were optimum numbers of fin in fin installations for highest heat transfer. Özceyhan and Altıntop [7] have done the analyses of heat transfer and thermal strain in pipes with cavity by using finite element technique. Taslim and Liu [8] have done numerical and experimental work to analyses the heat transfer structures which occur from a surface with 45 degree extension in a rectangle shaped channel. Hong-Min Kim and Kwang-Yong Kim [9] have examined optimization of wavy surfaces and extended surface about turbulence heat gain and they have carried out studies about different flow conditions such as fin height, the distance between them.

Some of results of this study were presented in GAP conference by Derya Kaya et. al. [10]. The effects of different fin types, different fin heights and channel interior surface temperature changes and triangular fin with groove have been examined numerically in the present study. First of all to demonstrate the accuracy of the results by doing analytic solutions in flat channel, and results have been compared for Fluent and Ansys-Flotran computer programs. The results were presented as temperature distribution and local heat transfer coefficient distribution for different geometries and different flow conditions.

## GEOMETRIC MODEL

Figure 1 shows the model that was used in the present study. Boundary conditions and geometry can be seen in this Figure.

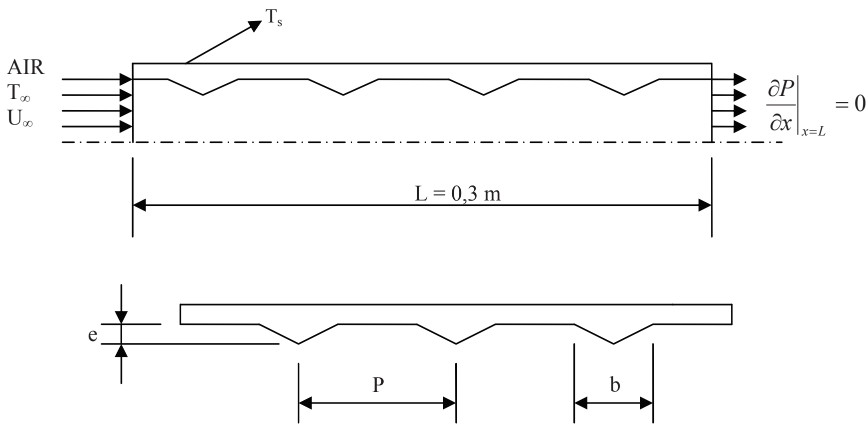



Figure 1. Model which belongs to fin structures used in study

- ✓ Fin shape : Triangular, circular, rectangular
- ✓ Fin height (e) : 1,5 mm, 3 mm
- ✓ Fin intervals (p/e) : 6,6-10-20
- ✓ Surface temperature of wall ($T_s$) : 303 K
- ✓ Reynolds number : 500
- ✓ Air temperature ($T_\infty$) : 350K

As thermal conductivity of fin material will affect heat distribution along the fin, from beginning to end of the fin for temperature change's being least, aluminum which has high thermal conductivity, cheap and light has been chosen as channel material. And by assuming that fins have been produced together with channel surface as a whole, thermal contact resistance between fins and surface was neglected. In this study k-ε turbulence model has been used in calculations.

For the most appropriate grid model's being be able to be prepared, fine grid should be formed in regions where variables such as velocity, pressure, and temperature changes strong. Because of this, in geometry finest grid on channel surfaces which were added fin has been preferred and in other zones sparser grid has been preferred. In channel with triangle fin for grid formation triangle element which is in accordance with geometry has been used. It has been determined that by comparing to solutions which has different element number (3000, 6000), grid structure which consists of 11000 triangle elements will be sufficient. For circular and rectangle channel with fin, solutions have been realized with grid structure which is consisted of triangle and 13000 elements.

## NUMERICAL METHODS

### Finite Element Method

Finite elements method is one of the approximate solution methods which solve the complex and difficult problems by an acceptable approximate engineers encounter. Finite elements method is a solution form that the complete solution is found by complex problems' being divided into easier inferior problems, by each ones' being solved in it's own.

In most of the engineering problems, manager differential equations' analytical solutions can not be done. Existing analytical analysis is only available for simple conditions. In this respect the method of differential equations' numeric solutions is applied. Finite elements method is used for solutions of the problems related to strain analysis, mechanics of fluids, heat transfer, mechanical vibrations and etc. In finite elements method, by calculation area's being separated to the region in finite number, mesh points are formed. As a result of some decomposition methods which are applied, variables (velocity, pressure, temperature and etc.) on the mesh points have been tied to each other by algebraic correlations. That is to say these algebraic correlations replace of differential equations. Finite elements method is used with methods such as variation formulations and weighted residual methods. Infinite elements method is mainly preferred for complex geometries. Although this method has been improved for the calculation of strain in solids and deformation, it has been improved on the point of being used on flow cases and heat transfers, too. For this aim, Ansys-Flotran packet program which is used at this study, by finite elements approach, has been improved for the solution of the flow cases. The basis concept of finite elements method is a continual quantity's such as heat, pressure or displacement being transformed in to a model which is formed by little and continual parts' uniting. According to this method, original geometry is placed by serial of elements which contain points on the surface and inside of the material.

**Finite Volume Method**

Finite volumes method as similar to finite elements method is based on the principle of by dividing the geometry which will be solved into portions to find a solution for each of these portions and then by uniting these solutions to find the general solution of the problem. And finite volumes method as similar to finite elements method is based on finite differences method. But in terms of development it is rather sensitive than finite differences method. As being different from finite elements, this method uses a technique which is based on control volume for transforming flow equations to algebraic equations which can be solved as numerical. In other words this technique is based on the principle of taking flow equations' integration in each control volume. This integration result provides the equations which characterize each control volume coming to the light.

## THE EVALUATION OF RESULTS

As the aim of this research is the examination of the heat transfer structures on surfaces to which fin has been added, as the beginning with the purpose of comparison solutions for plain channel which has smooth surface has been provided. Initially, for the chosen geometry analytical solution has been done. Later, for same geometry and border provisions with Ansys-Flotran based on finite elements and Fluent packet programs based on finite volumes solutions have been executed. In Table 1 the results obtained by numeric methods has been compared to results which are calculated as analytical.

Table 1. Comparison of numerical and analytical results for a flat channel

| | Analytical | Ansys | Fluent |
|---|---|---|---|
| Exit velocity, m/s | 1.38 | 1.332 | 1.336 |
| Exit temperature, K | 310 | 306.702 | 307.76 |
| Heat transfer coefficient, W/m$^2$K | 15.92 | 18 | 14.9 |

In Figure 2, local distribution of heat transfer coefficient on flat channel surface and temperature distribution of channel entrance section have been demonstrated. From the graphic, it can be seen that local distribution of coefficient of heat transfer which belongs to packet programs of Ansys and Fluent are well agreed with each other.

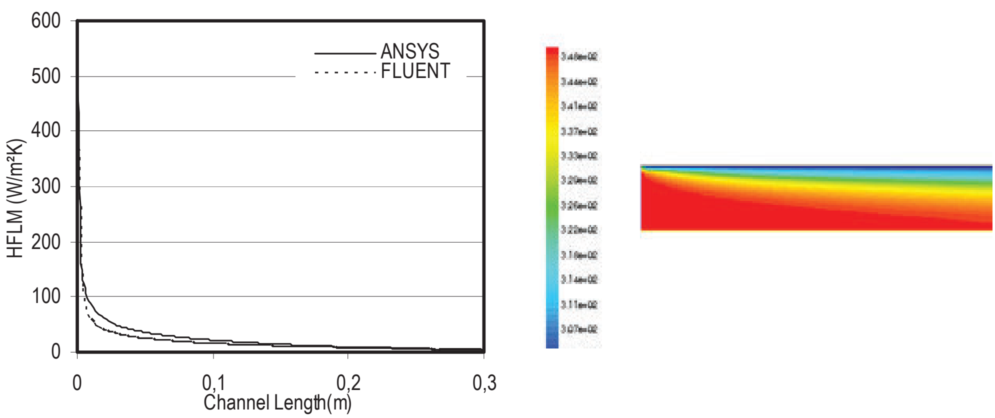



Figure 2. Distribution of local heat transfer coefficient on flat channel surface and temperature distribution of channel entrance section

In the second part of the study, the effects of triangle fins which are placed on plain channel surface to heat transfer have been examined by Fluent program. In Figure 3, when looking at the graphic which shows distribution of local heat transfer coefficients on channel surface to which triangle fin has been added, it is seen that graphic like in the plain channel has inclination which continues by decreasing from maximum value. The increase in values of heat transfer coefficient and undulations formed by fins, are the expected diversities considered to graphic which belongs to plain channel. As long as progressed along the channel the decrease in undulations which fins have done is because of the decline of the temperature diversity which occurs by cooling of fluid decrease. Again same graphic shows the solution results in values of different Reynolds number. As long as the velocity increases, then heat transfer coefficient increases, too.

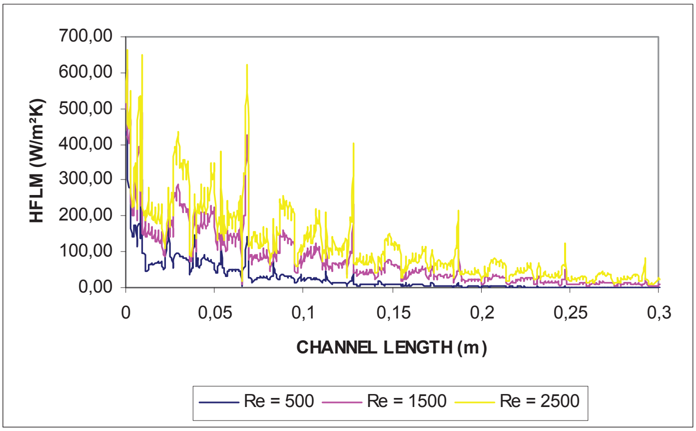
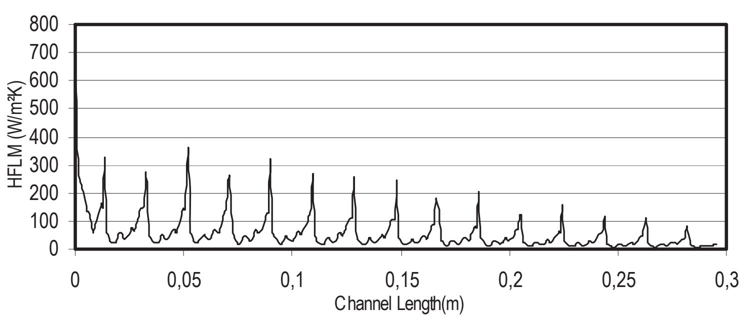
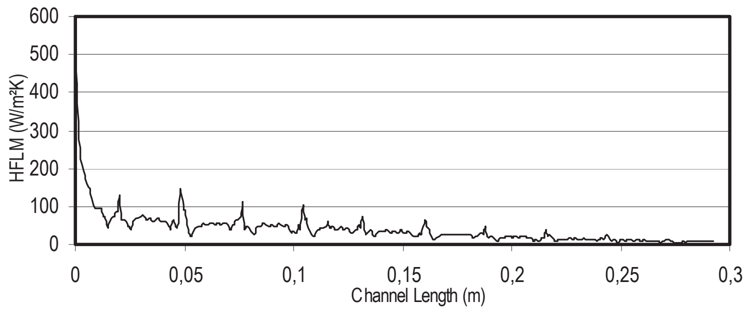



Figure 3. Distribution of local heat transfer coefficients for different Re number with triangle fin ($T_\infty$ = 350 K, $T_s$ = 303 K, p/e=6.6 ve e=1.5 mm)

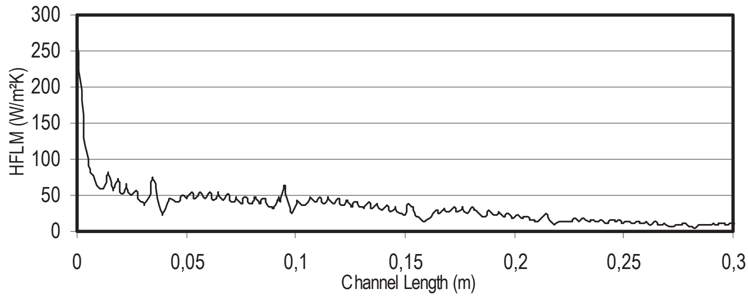
Figure 4 shows the local heat transfer coefficient distribution for different type of fin spacing (p/e) with e=3 mm. When p/e rate is low, fin interval decreases and number of fins in the same surface increase. Because of the number of fins' being excessive both area of heat transfer surface area increases and turbulence formation develops earlier. Because of these reasons, in the case of p/e rate's being low, it is expected that heat transfer coefficient increases. From the graphics, when p/e rate increases then values of heat transfer coefficient decrease. In this situation, the most suitable fin interval can be considered as p/e= 6.6. Fins with their widening of heat transfer surface and their fluid undulations which they will form are designed in order to increase heat transfer in the same volume. If appropriate fin intervals (number of fins) are not determined, as extra pump or ventilator power will be needed in order to increase flow power because of the flow's mixing much, it will be fell short off the aimed goal. Because of this, the aimed increase in heat transfer should also meet the pressure decrease. Although we obtain the most coefficient of heat transfer, when it is considered in point of view of both heat transfer increase and friction factors, it is understood that p/e= 6.6 rate won't be appropriate. Because of these reasons, it should be stated that the design of fins at intervals which is thin and won't prevent the flow will be productive.




a ) p/e = 6,6



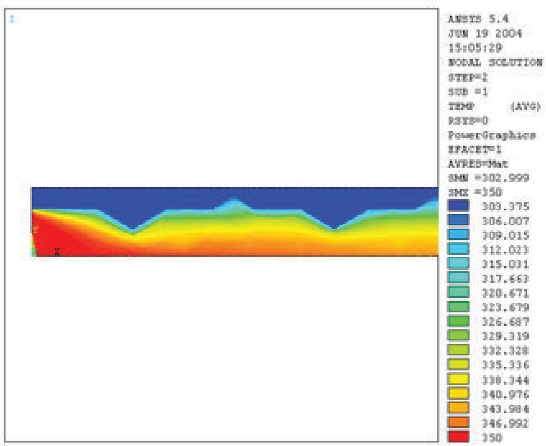

b) p/e = 10




c ) p/e = 20

Figure 4. Distributions of local heat transfer coefficient for different fin intervals (p/e) on triangular fins (Re = 500, $T_s$ = 303 K, e=3 mm)

In Figure 5, for triangle fin with groove for Re= 500, p/e= 6.6 and e=1.5 conditions temperature distribution at entrance of the channel and heat transfer coefficient distribution have been shown. As it is seen on figures, temperature contour change can be seen both after fin and also after groove. Results of this, heat transfer coefficient changes much wavy when compared with Figure 6. As related to shape of the surface then temperature change's being excessive, the energy which is transferred will be more for this geometry. Distribution of local heat transfer coefficients for different type of fin geometries without groove which are shown in Figure 6 in accordance with temperature contours, while heat transfer coefficient is 28 $W/m^2K$ for triangle fin which is shown in Figure 3, this value for circular and rectangular geometries have reached 30.8 $W/m^2K$ and 37 $W/m^2K$ respectively. In addition to this, the increase of the mixture which has come into existence will also increase the heat transfer. But when doing the choice of appropriate fin and appropriate fin distances, it is required that pressure decline also should be taken into consideration. Because of this reason, it is required the geometry which both won't prevent flow and provide energy saving in the desired amount should be chosen.



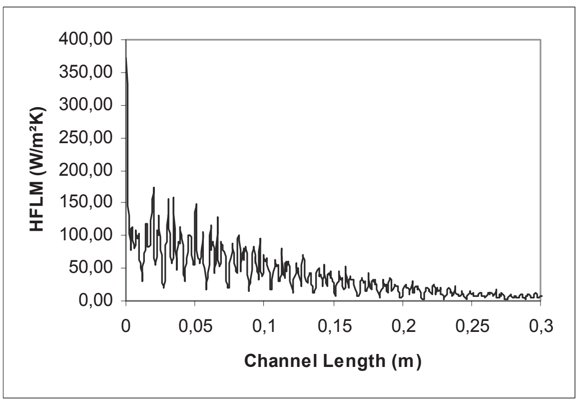
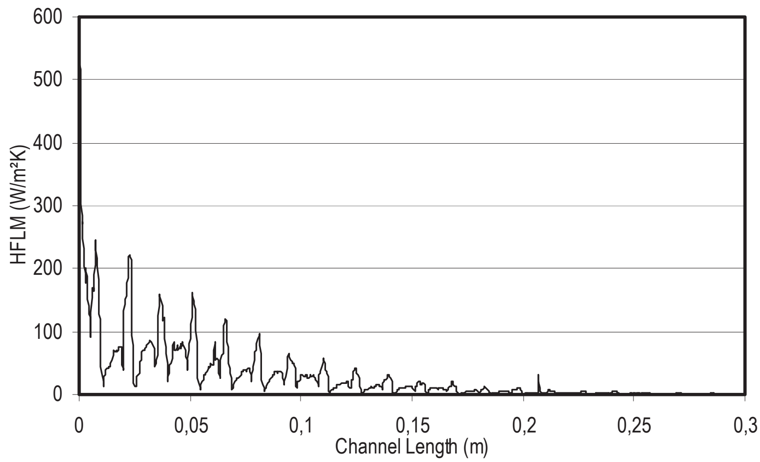
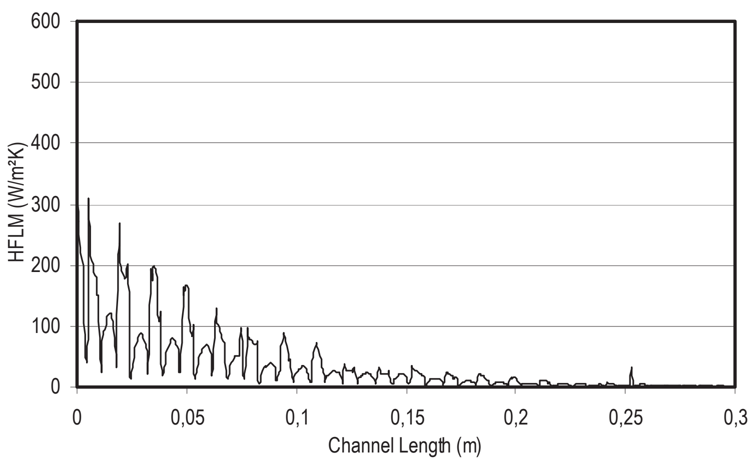




Figure 5. Temperature contour at entrance of the channel and Heat Transfer Coefficient Distribution for the Triangular Fin with Grooved Surface (Re = 500, p/e=6.6 ve e=1.5 mm)




(a) Circular




(b)Rectangular

Figure 6. Distribution of local heat transfer coefficients for circular and rectangular fins ($T_s$ = 303 K, p/e=6.6, e=3 mm, Re = 500)

## CONCLUDING REMARKS

In the presented study, distribution of the heat transfer coefficient on surfaces with extension has been done in different geometry and flow conditions. The necessity that fins execute heat transfer by their features of broadening surfaces of heat transfer and forming mixed flow, in fin design these two conditions need to be taken into consideration in the way they will balance each other has been put forward. From this point it should be emphasized that haphazard design can not be done for the heat transfer increase's being realized with fins. In fin design on channel surface, it is determined that the design which is thin and will be formed by dense intervals that won't prevent flow will provide the best result in heat transfer.

It is required that the numerical study which has been done should be supported as experimental and in addition to this the results in higher flow velocities and for more complex geometries should be found, the results' with pressure declines being given should be planned in the forth stages of this study.

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# APPLICATION OF CARBON DIOXIDE IN A TRANSCRITICAL REFRIGERATION CYCLE

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## ABSTRACT

During the last century, the refrigeration industry was forced to change refrigerant technology due to commitments to the Montreal protocol. The industry is redesigning technology towards the application of new environment friendly refrigerants that may be used for a longer period. Refrigerants should be chlorine free, which means that they do not have impact on the ozone layer, and at the same time they should have low global warming potential.

Emphasis in this paper is given to the use of carbon-dioxide R744 and ammonia R717 because of their low impact on the atmosphere. The cascade system technology is known with $CO_2$ as the refrigerant in the first stage and $NH_3$ as the refrigerant in the second stage of the cascade. This paper elaborates the use of $CO_2$ in the transcritical area, i.e. above the critical point where there is no vapour condensation. The paper deals also with problems connected with high working pressure of $CO_2$, thermodynamic properties and design of transcritical heat pumps and their components.

Compared to other refrigerants, $CO_2$ has one specific property: low critical temperature 31,1 °C at 73,8 bar. The operating pressure for gas cooling will be extremely high, above 100 bar. In the subcritical area the enthalpy change is influenced by the temperature change, while in transcritical area enthalpy change is affected by pressure. In the transcritical area, the coefficient of performance may be controlled by the refrigerant pressure after compression.

**Key words**: carbon-dioxide, natural refrigerants, transcritical heat pumps

## 1. INTRODUCTION

Carbon dioxide is investigated [1-3] as a *natural* solution to decrease the impact of refrigeration and air conditioning applications on global warming of anthropic origin. The use of $CO_2$ as a refrigerant looks promising in several applications like cascade refrigeration systems, commercial refrigeration systems, heat pump water heaters and mainly for systems characterised by relatively high leakage potential, as in automotive AC.

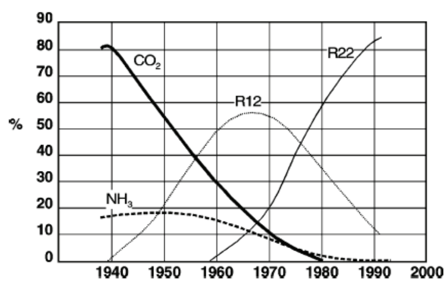



Figure 1.1 Percentage use of main primary refrigerants in existing marine cargo installations [3]

Around 1930, the chemical industry developed new CFC refrigerants which were considered to be almost ideal because of their properties. Many factors caused the withdrawal of $CO_2$ from use, e.g. problems related with high operating pressures and leakage prevention with current technology, loss of capacity and efficiency due to leakage, reduction of performance at high ambient temperatures, aggressive marketing of CFC manufacturers, lower CFC system price, inability of $CO_2$ system manufacturers to enhance heat pumps.

It was realised in the 1980s that CFCs were causing many problems as impact on the ozone layer reduction and on global warming. The chemical industry developed new compounds, HCFCs and HFCs, which could replace existing CFC refrigerants, but there was also an endeavour to push them out from use.

In 1992 Lorentzen and Pettersen published the first paper on $CO_2$ application as refrigerant in vehicle industry heat pumps [4]. The comparison of thermodynamical performance of HFC and $CO_2$ systems shows lower performance for $CO_2$. The environmental comparison indicates better results of $CO_2$. New programs where developed for $CO_2$ usage in the vehicle industry, like the RACE program of the European industrial consortium, European program COHEPS about $CO_2$ heat pumps usage and activities connected with $CO_2$ in IEA-the international energy agency. There is an additional restriction for refrigerants in the vehicle industry, which need to have GWP lower than 150 [3].

## 2. THERMODYNAMICAL PROPERTIES OF CARBON DIOXIDE R744

The use of refrigerants containing chlorine like chlorofluorocarbons CFC (R12) and chlorofluorohydrocarbons HCFC (R22) is completely eliminated, while the application of hydroflurocarbons HFC (R134a) is doubtful because they affect the climate through the global warming effect. The research and development of new chemical synthetic compounds is abandoned, and it is is directed towards the use of natural refrigerants like water, air, hydrocarbons, ammonia and carbon-dioxide. Some of these refrigerants are limited with flammability regulations. Attention is given to the application of carbon dioxide R744 and ammonia R717 because of their favourable characteristics (Table 2.1). While selecting the right refrigerant it is necessarily to regard its flammability, thermodynamical properties, transport properties and total global warming potential.

Table 2.1 Physical properties of refrigerants

| | R12 | R22 | R134a | R407C | R410A | R717 | R290 | **R744** |
|---|---|---|---|---|---|---|---|---|
| ODP/GWP | 1/8500 | 0,05/1700 | 0/1300 | 0/1600 | 0/1900 | 0/0 | 0/3 | **0/1** |
| Flammability / toxicity | no/no | no/no | no/no | no/no | no/no | yes/yes | yes/no | **no/no** |
| Molecular mass [kg/kmol] | 120,9 | 86,5 | 102,0 | 86,2 | 72,6 | 17,0 | 44,1 | **44,0** |
| Critical pressure [bar] | 41,1 | 49,7 | 40,7 | 46,4 | 47,9 | 114,2 | 42,5 | **73,8** |
| Critical temperature [°C] | 112 | 96 | 101,1 | 86,1 | 70,2 | 133 | 96,7 | **31,1** |
| Volumetric cooling capacity, [kJ/m$^3$] | 2734 | 4356 | 2868 | 4029 | 6763 | 4382 | 3907 | **22545** |
| First commercial usage | 1931 | 1936 | 1990 | 1998 | 1998 | 1859 | - | **1869** |

Basic properties of $CO_2$:

- $CO_2$ has a low critical temperature of 31,1°C, and therefore the characteristic process runs in the transcritical region, above the critical pressure of 73,8 bar.
- At 0 *°C* the saturation pressure is 34,81 *bar*.
- The volumetric cooling capacity is 3-10 times higher than for CFC, HCFC, HFC and HC.
- In the transcritical region, the pressure affects the coefficient of performance, because of that the high pressure side is controlled. The process is transcritical, so the gas cooling temperature doesn't affect the COP.
- For water heating heat pumps, the optimal COP of 4,5 is gained at the pressure of 90-95 bar, which is extremely high if compared with freon systems (condensation pressures for freons are in the range from 10 to 20 bar, for R410A up to 35 bar)
- High working pressures of 30-100 bar take effect on the system design and armature, but the explosive energy of $CO_2$ systems is not much higher than that of standard systems because the working volume is smaller. The compression ratio is lower, thus the performance of the compressor is higher.
- The transcritical change of state of $CO_2$ requires specific heat exchanger design, here called the gas cooler, while the same exchanger in standard systems is called the condenser.
- Transcritical temperature gliding is useful in systems for heating air or water. With the right exchanger design, $CO_2$ outlet temperatures from the gas cooler may be a few degrees above the inlet temperature of cooling air or water, and it is possible to achieve a high coefficient of performance.

Figure 2.1 represents a typical schematic of a $CO_2$ refrigeration cycle [2]. In $CO_2$ systems, the gas cooler replaces the condenser because the refrigerant is at supercritical pressures and no liquid forms in the component. Superheated refrigerant (1', Figure 2.2) flows into the compressor, where it is elevated to a high-pressure and high-temperature state (2'). Then the supercritical refrigerant flows through the gas cooler, transferring heat to the environment, while being cooled down to point 3. Afterwards, it exchanges heat with the low-pressure refrigerant in the internal heat exchanger (IHX) and flows into the expansion device (e.g. capillary tube) where it is throttled to low-pressure and low temperature (4'). Finally, the refrigerant in the evaporator absorbs heat and evaporates. The cycle 1-2-3-4-1-represents a cycle without IHX.

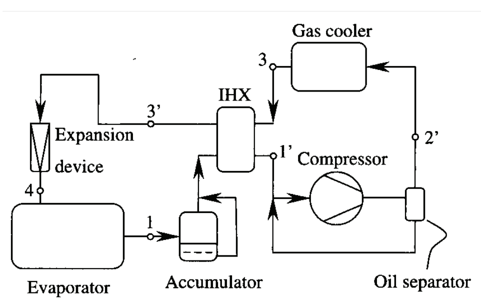



Figure 2.1 Carbon dioxide refrigeration cycle

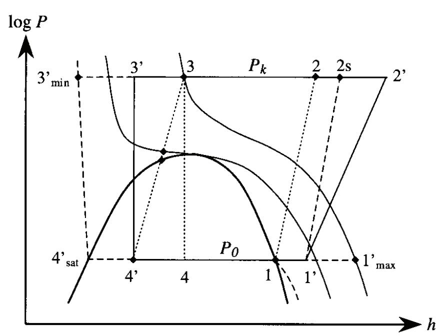



Figure 2.2 *p-h* diagram of carbon dioxide transcritical refrigeration cycle

## 3. APPLICATION OF $CO_2$ IN TRANSCRITICAL AREA

In the transcritical area, pressure and temperature are mutually independent. In the subcritical area, temperature has the highest impact on enthalpy change (Figure 3.1), while in transcritical area pressure has the highest effect. It is necessary to control the discharge compressor pressure because the pressure affects the specific heat. The compressor power depends on the discharge pressure, and coefficient of performance COP also depends on the pressure. In standard systems, the coefficient of performance decreases with the increase of the discharge pressure, but in transcritical process the condition are more complex.

Figure 3.2 shows the dependence of the coefficient of performance and some other variables like specific cooling capacity $q_o$, specific compressor power $w$, transcritical pressure $p_k$ and the outlet temperature of vapour $\vartheta_{ex}$ ($\vartheta_3$) at the gas cooler exit on the pressure of the high temperature side. The diagram shows the theoretical dependence based on an ideal heat cycle with evaporating temperature $\vartheta_e = 5$ °C, and gas cooler outlet temperatures $\vartheta_{ex} = 35$ °C and 50 °C [3].

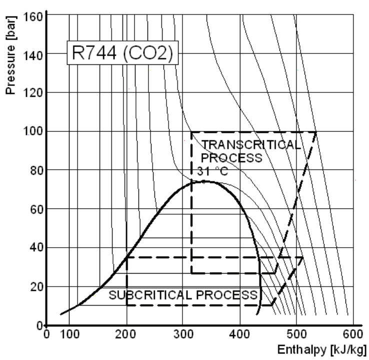



Figure 3.1 Transcritical $CO_2$ process in *p-h* diagram

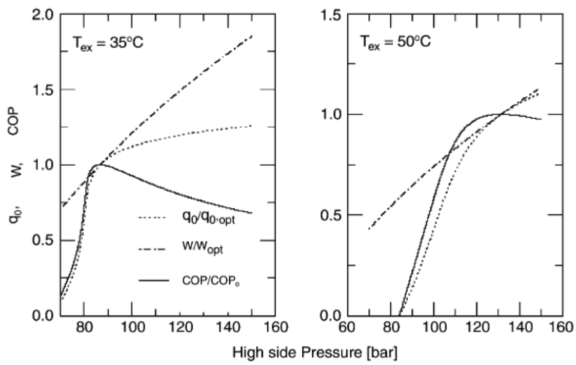



Figure 3.2 Influence of $q_o$, $w$, $p_k$, $\vartheta_{ex}$ on heat pump COP

The COP increases with the increase of compressor discharge pressure, but at some point the trend changes as a consequence to increased compression work. The $\vartheta_{ex}$ curve becomes steeper with the pressure increase and the effect of rising pressure is reduced. Isentropic compression is almost linear. Maximal, optimal performance for $\vartheta_{ex} = 35$ *°C* is at the pressure of 87 bar, and at the outlet temperature of 50 °C, the optimal pressure is 131 bar. Heat pump thermodynamical coefficient of performance depends on a couple of factors but the main one is the discharge pressure from the compressor. Changes of the discharge pressure also affects the cooling capacity, but also the heating capacity of a heat pump. The optimal pressure is increasing almost linearly with the increasing temperature of outlet vapour from a gas cooler $\vartheta_{ex}$.

### 3.1 Thermodynamical losses of transcritical heat pump

If the same evaporating temperatures and the same superheating after evaporator are presumed, and when transcritical and standard condensing cycles are compared, there is a difference in thermodynamical losses, which become much higher in transcritical processes.

The theoretical $CO_2$ process has higher losses at pressure reduction. Because the working point is close to the critical point, the cooling capacity loss will be high because of pressure reduction, and because the evaporating heat near the critical point becomes small. The real $CO_2$ process differs slightly from the theoretical process, and the superheat will not be as high.

Figure 3.3 shows the variation and temperature gliding in a $CO_2$ transcritical process while the single phase transcritical fluid cools down in a constant pressure process. The ideal cooling curve of the transcritical vapour follows the heating curve of water or air on the other side of a gas cooler. The best cost effectiveness of transcritical $CO_2$ systems is achieved in heating of domestic hot water, because the transcritical temperature curve follows the temperature of water. A heat pump for water heating from 10 to 80 *°C* is ideal with $CO_2$ as refrigerant.

Figure 3.4 shows the effect of the gas cooler (or condenser) outlet temperature on the coefficient of performance, normalized by the COP at 40 °C (evaporating temperature is 0 °C). It is necessary to reduce the outlet gas cooler temperature as close as possible to the entering temperature of cooling fluid, in order to decrease the effect of the pressure valve. While the ideal COP for R134a is increased by about 40 % through a 10 °C condenser outlet temperature reduction, the effect on the $CO_2$ cycle COP is nearly 70 % [3].

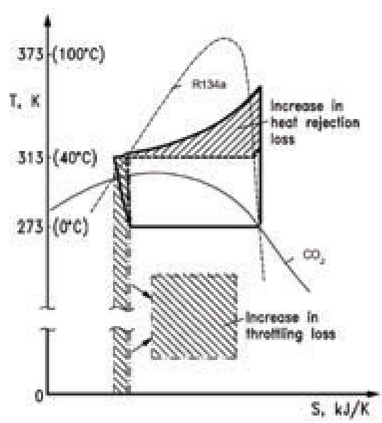



Figure 3.3 Comparison of $CO_2$ and R134a at same evaporating temperature

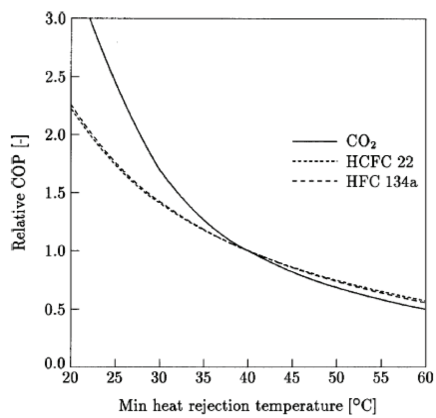



Figure 3.4 Relative change of cooling COP at varying refrigerant exit temperature from condenser/gas cooler

A great advantage of $CO_2$ systems is its insensibility of heating performance to evaporating temperature changes. This characteristic reduces additional energy needed for heating, if compared to standard systems.

Basically, the one stage theoretical transcritical process may be additionally enhanced in order to increase energy efficiency. Theoretically, it is possible to make many corrections in the basic system, e.g. multiple stage compression, multiple stage expansion, mid-exchanger for subcooling, using expansion work. Transcritical $CO_2$ heat pumps may be used for simultaneous heating and cooling because of specific working temperature conditions. When compared to the single stage process, the two stage process shows a performance increase of 15 % to 25 % depending on the concept used in the heat pump.

### 3.2 Pressure control on high-pressure side of heat pump

The pressure at the high temperature side of a $CO_2$ heat pump may be in the subcritical or the transcritical area, and it depends on the condensation temperature, cooling of transcritical vapour in the gas cooler, on the design and the application of the heat pump. While working in subcritical area, the system will perform as standard freon systems, where the condensation pressure depends on the condensation temperature, though the performance of the system will be pure because the critical point is close to the working point. In the transcritical area, the system pressure will depend on the mass of working fluid *m*, volume of working fluid *V* and temperature of working fluid *T*.

The properties of the working fluid can be described with:

$$p = p(v,T) = p\left(\frac{V}{m},T\right)$$

There are three ways to control high temperature side pressure:
- change of mass in the system $m$
- change of volume on the high temperature side $V$
- change of temperature on the high temperature side $T$

The first two methods make possible an active pressure control, while the third control method is passive depending on changes of room temperature. The temperature is controlled by means of volume and mass of refrigerant in order to adjust pressure depending on the temperature of outside air, with the goal to achieve optimum performance. In case of system leakage, the relation between temperature and pressure will change along with performance, which will usually drop. The heat pump will work mostly in transcritical area, but it is necessary to design the system for subcritical, condensing work also. The system will work in condensing mode when the ambient temperature drops below a critical point. In the vicinity of the critical point, condensation heat of is small, so the performance will be low.

## 4. COMPONENTS OF $CO_2$ HEAT PUMP

Compressors in $CO_2$ systems work at high pressures and large pressure differences (cca 80 bar), but the compression ratio is low, due to smaller volume ratio. Therefore the expansion loss of the compressor is also smaller. High working pressures demand a specific design of compressors. In Figure 4.1 ideal pressure-volume diagrams for compression of R134a and $CO_2$ with equal cooling capacity at 0 °C [6] are presented. The volumetric capacity of $CO_2$ is significantly larger than that of R134a, thus the dimensions of a $CO_2$ compressor are smaller or close to R134a compressor and the compressor walls are thicker. Compressors in $CO_2$ heat pumps have cylinder volumes up to seven times smaller than those for R134a. $CO_2$ vapour has high density because of high working pressures and vicinity of critical point. For the same working power, $CO_2$ compressor swept volume will be smaller (Figure 4.2).

$CO_2$ reduces the viscosity of oil when it dissolves in it. Large differences between evaporation and gas cooling (condensation) pressure may cause leakage to become a severe problem for the compressor. Therefore reciprocating compressors are used for $CO_2$, and scroll compressors are avoided. Additional enhancements may be accomplished by installing an expander instead of the reduction valve or by multistage compression. $CO_2$ compressors are designed to withstand working pressures up to 250 bar.

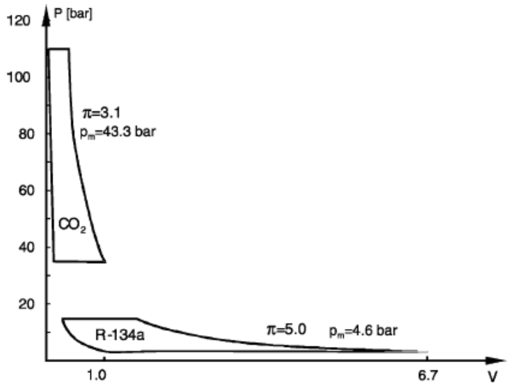
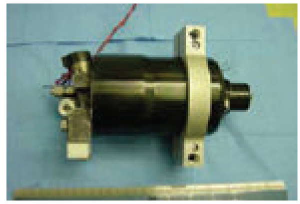
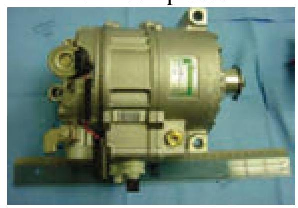



Figure 4.1 Compressor diagram p-v,
π - compression ratio [6]



R744 compressor



R134a compressor
Figure 4.2 Comparison of
compressor dimensions

Heat exchangers are made of microtubes produced by extrusion. The exchanger heat transfer surfaces are finned in order to increase the heat transfer on both the air side and the $CO_2$ side. High working pressure and characteristic properties of $CO_2$ demand the application of microtubes yielding high compactness of the exchanger. Extended surface and small dimensions of exchangers make them acceptable in cars and cold-store trucks. Standard pipe heat exchangers for $CO_2$ have increased mass because of high working pressures, which demands thicker walls.

At high working pressures the pressure drop is almost negligible, allowing higher flow velocities and thus better heat transfer coefficients. This is convenient for transcritical gas coolers. The proximity of the critical point increases the specific heat of $CO_2$ and high operating pressures result in a compact design with extruded microtubes and reduced dimensions of all the other components as valves, regulating elements etc.

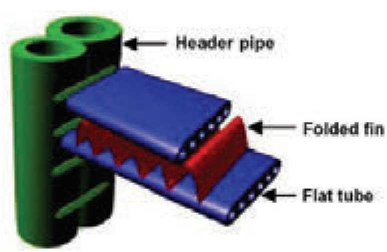



Figure 4.3 Design of microchannel heat exchangers [3]

## 5. EXAMPLES OF $CO_2$ USE IN TRANSCRITICAL AREA

Systems should be designed to make advantage of specific thermodynamical properties of $CO_2$ in the transcritical region. In every application it is necessary to respect the requested conditions as e.g. market demands, unit mass, systems for water heating with transcritical characteristics and new directives.

### 5.1 Heat pumps in automotive systems

Modern car engines with fuel injection lack waste energy for cabin heating and defrosting of the evaporator. $CO_2$ heat pumps can provide additional energy at an efficiency almost three times higher than with electric heaters. There are problems with microchannel evaporators, which are exposed to water, frost and ice and their defrosting. The evaporator may be defrosted using the exhaust gases or engine cooling liquid. Defrosting by system inversion is not acceptable because it would cause unacceptable temperature drops of the cabin air. One of the main problems is the energy shortage for heating and defrosting of the evaporator until the engine reaches adequate temperature, and energy transfer from the engine to the evaporator.

Air conditioning units with R134a, which is commonly used in today's systems, have a high leakage rate due to rubber pipes used in heat pumps to reduce vibrations. $CO_2$ enables the design of compact units where the front mask may be reduced to improve aerodynamical profile.

$CO_2$ units designed so far have better performance than R134a units, and they have lower total global warming potential. Heat exchangers are made from extruded aluminium microtubes, reducing the unit mass. There are problems in introducing this new technologies, which demand training new maintenance technicians. It is also required that the system works in a wide temperature range, in all climate conditions, from high summer temperatures to low winter temperatures.

### 5.2 Heat pumps in buildings

Prominent advantages of $CO_2$ heat pumps are their suitability to heat air up to 60 °C and lower air flow rates. Standard heat pumps use lower air temperatures and high air flow rates because the coefficient of performance of freon heat pumps is low at high temperature differences. $CO_2$ systems yield higher performance and more heat at low evaporating temperatures.

The $CO_2$ heat pump can produce temperature differences up to 50-60°C between the evaporator and the gas cooler at low volumetric flow rates and small pipe diameter. Figure 5.1 shows a system for water heating with a $CO_2$ transcritical heat pump.

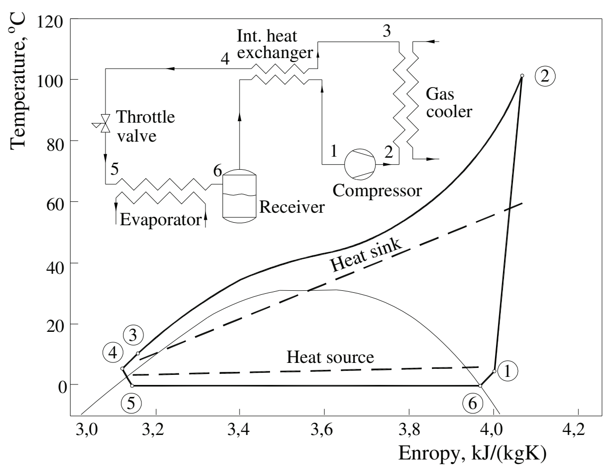



Figure 5.1 Domestic hot water heat pump

In order to achieve a high COP, it is essential that useful heat is rejected over a large temperature range, resulting with relatively low $CO_2$ outlet temperature from the gas cooler, and that after the pressure reduction there is a low gas phase amount in the refrigerant flow. The example of this concept is the 50 kW heat pump designed in SINTEF/NTNU laboratory [5]. The pump heats water to 60°C and the coefficient of performance depends on the evaporating temperature. Such systems may be used for major heating demands like in hotels, apartment complexes, hospitals, process industry, but also for small residential buildings. Heat pumps for family houses are usually under 10 kW, and are used for heating of domestic hot water at night, when the electric energy is cheaper, and the water is stored into insulated tanks. It is possible to achieve COPs up to 4 with water temperatures of 60°C. The performance drops by 15% when the water temperature is raised to 80 °C. Such high performances are achieved due to low compression ratios and favourable thermodynamic properties of $CO_2$.

Figure 5.2 represents a $CO_2$ transcritical heat pump system which is used for combined space heating and hot water heating [7].

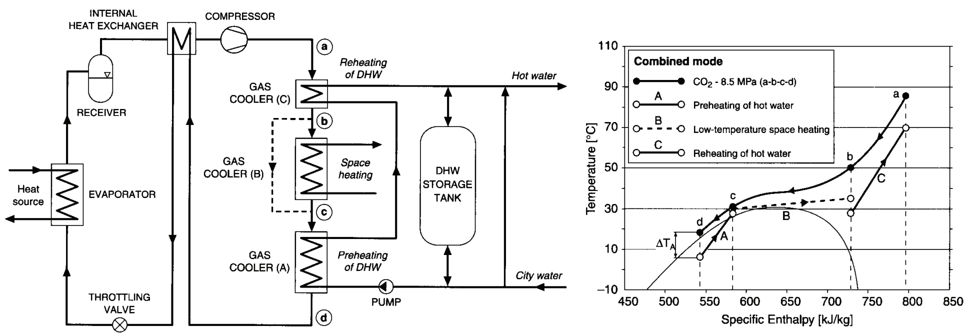



Figure 5.2 The principle of an integrated residential $CO_2$ heat pump system [7]

The high temperature part of the $CO_2$ curve is suitable for heating of domestic hot water (DHW), and low temperature part may be used for space heating. The gas cooler units A and C are connected to a standard DHW storage tank and an inverter controlled pump by means of a closed loop. The gas cooler unit B is connected to a low-temperature heat distribution system with floor heating, convectors or fan coils. An integrated $CO_2$ heat pump will be operating in three different modes: space heating only, hot water heating only and simultaneous space heating and DHW heating (combined mode). COP is in the range of 3 to 4.

### 5.3 Commercial cooling units

$CO_2$ cooling units can be also used within major cooling systems in supermarkets or central kitchens because those systems usually have high refrigerant leakages because of inadequate maintenance.

Cascade $CO_2/NH_3$ systems are used in the cold rooms and cabinets for deep cooling and storage. $CO_2$ is used in the lower cascade of the system in the temperature range from -10 to -50 °C where it is working in the subcritical range with vapour condensation, yielding high performance. The main advantage of this cooling concept is that ammonia is located outside the cooling area, so it may not get into contact with storage area workers, and that the amount of ammonia in the system is reduced ten times.

## 6. CONCLUSION

The paper deals with the application of refrigerant R744 in transcritical area. It is expected that in the near future obstacles related to high operating pressures of $CO_2$ as a refrigerant will be overcome with system components improvements. According to European directive 2006/40, dealing with vehicle air conditioning system leaks, after the year 2011 it will be prohibited to use refrigerants with a GWP>150 in new vehicle models. Basically, only natural refrigerants satisfy this requirement. It is necessary to develop new high pressure armature and transcritical heat exchangers with microchannels, which may be used in $CO_2$ heat pumps.

The specific $CO_2$ transcritical curve with a temperature glide is ideal in heating systems, and it is foreseen that transcritical heat pumps will gradually replace heating systems based on fossil fuels. Significant developments have been done in cascade systems using ammonia and carbon dioxide as refrigerants. The global warming issue caused a new approach in evaluation of refrigerants in refrigeration and air conditioning systems.

## NOMENCLATURE

| | | |
|---|---|---|
| COP | - coefficient of performance | [-] |
| $c_p$ | - specific heat | [kJ/(kgK)] |
| $m$ | - mass | [kg] |
| $p$ | - pressure | [Pa] |
| $s$ | - specific entropy | [kJ/(kgK)] |
| $q_o$ | - specific cooling capacity | [kJ/kg] |
| $q_{oV}$ | - volumetric refrigeration capacity | [kJ/m$^3$] |
| $T$ | - temperature | [K] |
| $v$ | - specific volume | [m$^3$/kg] |
| $V$ | - volume | [m$^3$] |
| $w$ | - specific work | [kJ/kg] |
| $\vartheta$ | - temperature | [$^o$C] |
| $\vartheta_e$ | - evaporating temperature | [$^o$C] |
| $\vartheta_c$ | - condensing temperature | [$^o$C] |
| $\vartheta_{ex}$ | - outlet temperature from gas cooler | [$^o$C] |
| $\rho$ | - density | [kg/m$^3$] |

**Abbreviation**

CFC - *chlorofluorocarbons,* completely halogenated derivatives of saturated hydrocarbons (R11, R12)

HCFC - *hydrochlorofluorocarbons,* partially halogenated derivatives of saturated hydrocarbons containing chlorine (R22)

HFC - *hydrofluorocarbons,* partially halogenated derivatives of saturated hydrocarbons without chlorine (R134a, R152, ...)

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# Numerical determination of temperature distribution at asphalt carrier ship structure

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## ABSTRACT

The paper presents numerical calculation of thermal insulation thickness for ship hull structure elements. Starting with simple model and continuing with more complex ones, the temperature distribution through thermal insulation of asphalt carrier ship structure was determined for various boundary conditions. Based on developed calculation procedure it is possible to determine the insulation thickness which will ensure the temperatures of ship construction elements in the safe range.

## INTRODUCTION

The liquid asphalt is transported by ship in specially designed tanks strengthen by stiffeners, thermally insulated and separated mutually by waterproof bulkheads (see fig.2). The average asphalt temperature is 250 ºC. The storage tanks for liquid asphalt must be constructed in such a way that all loads caused by gravity and inertia are transported from web frames to ship hull structure elements which are placed in the same horizontal and vertical planes as the web frames.

Figure 1. Asphalt carrier ship

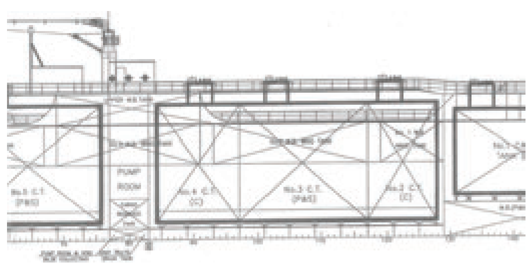

Figure 2. Asphalt tanks built in ship structure

The T profile and Holland profile frames are on the outer side of asphalt tanks (fig. 3). They are insulated by mineral wool (fig. 4) to protect the ship hull structure elements from overheating. The goal of the work was to develop the method for calculation the adequate thermal insulation thickness which will ensure the temperatures of flanges on web frames to remain below 70 ºC.

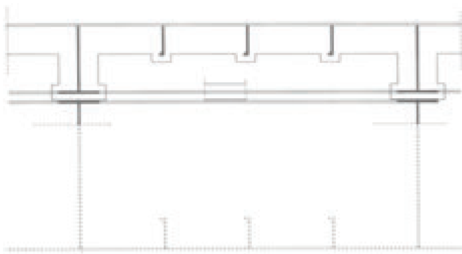

Figure 3. The plan of web frames

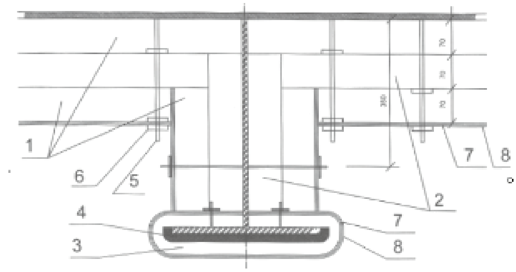


Figure 4. The web frame with insulation

For numerical calculation the characteristic cross section of tanks with stiffeners and insulation was taken into consideration (see fig. 5).

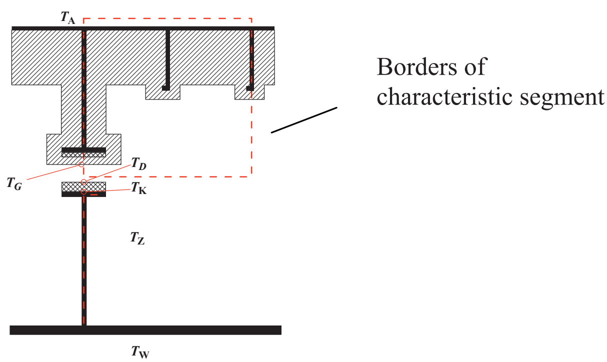


Figure 5. The cross section of characteristic segment

Where:

$T_A$ - asphalt temperature, K
$T_Z$ - temperature of the air between isolated tanks and hull ship, K
$T_W$ - temperature of the sea, K

**DESCRIPTION OF THE PROBLEM**

Different possibilities of stiffeners insulations were analyzed with the aim to ensure the flange temperature $T_K$ below 70 ºC. The analysis was performed by means of numerical method for given boundary conditions and material characteristic. Different insulating materials were applied in calculations.

**SOLVING THE PROBLEM**

The problem was solved in three steps:

1. Analysis of the main parameters for numerical model
2. Evaluation of numerical model
3. Numerical simulation

To obtain the basic parameters for numerical model the preliminary calculations were performed by means of classic thermodynamical approach taking into account radiation, convection and conduction. The goal was to get the approximate values of critical temperatures on the construction. Numerical model evaluated is based on the control volume method and is two dimensional steady state one and for known boundary conditions.

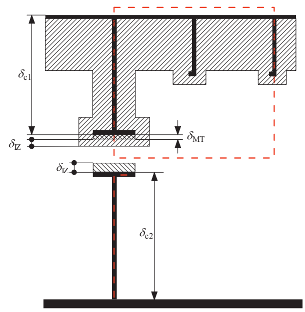


Figure 6. The geometry of the characteristic segment

**NUMERICAL MODEL**

The dimensions of control volumes were chosen to satisfy the needed accuracy. The model itself is based on 3D differential equation for head conduction:

$$\rho\, c\frac{\partial \vartheta}{\partial t}=\frac{\partial}{\partial x}\left(\lambda\frac{\partial \vartheta}{\partial x}\right)+\frac{\partial}{\partial y}\left(\lambda\frac{\partial \vartheta}{\partial y}\right)+\frac{\partial}{\partial z}\left(\lambda\frac{\partial \vartheta}{\partial z}\right)+\phi_v \tag{14}$$

where:

$\rho$ - density, $kg/m^3$
$c$ - specific heat capacity, J/(kgK)
$\lambda$ - heat conduction coefficient, W/(mK)
$\vartheta$ - temperature, ºC
$x, y, z$ - space coordinates, m
$\phi_v$ - heat source or sink, $W/m^3$
$a$ - thermal diffusivity, $m^2/s$

For numerical simulation the following presumptions were accepted:

- Temperature of asphalt tank's bottom is 250 ºC.
- Temperature of ship structure in contact with sea water is 15 ºC.
- Emissivity coefficient of all observed surfaces is $\varepsilon = 0{,}85$.
- Heat exchange on border surfaces was calculated as convection heat transfer with heat transfer coefficient of $\alpha_k = 4\ W/m^2K$ or combination of convection and radiation.
- For frames on hull ship (lower part) radiation was neglected.

The entire volume net is of higher density in the areas where higher temperature gradients are expected. Total number of control volumes is in average about 23 000.

## RESULTS

### Variant I

There is no insulation layer on flanges.

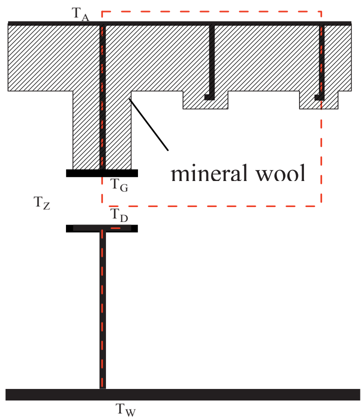


Figure 7. Variant I

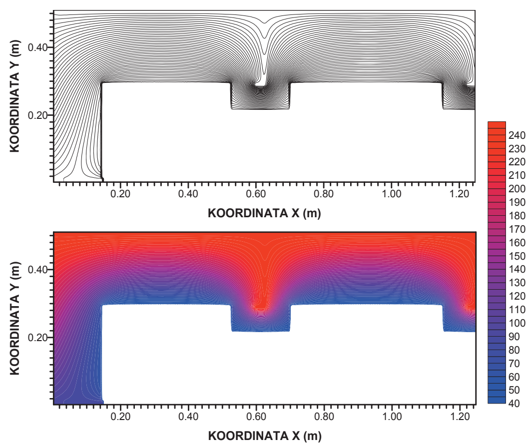


Figure 8. The surface temperature distribution across the characteristic segment of tank web frame

### Variant II

Insulation exists on both web frame flanges.

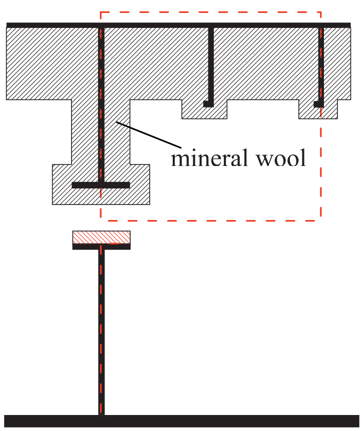


Figure 9. Variant II

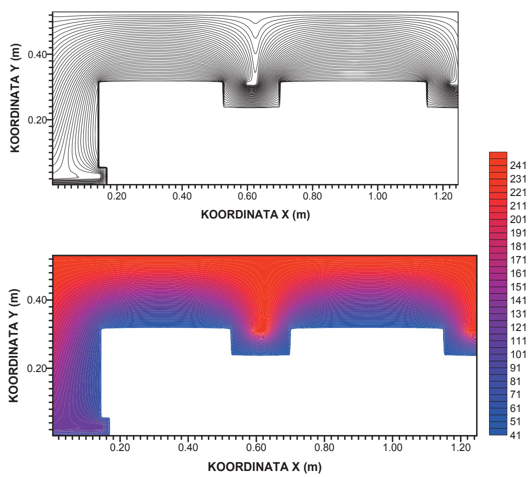


Figure 10. The surface temperature distribution across the characteristic segment of tank web frame

**Variant III**

Insulation exists only on upper web frame flange. It consists of high quality insulation layer ($\lambda = 0{,}02776$ W/mK) and layer of mineral wool.

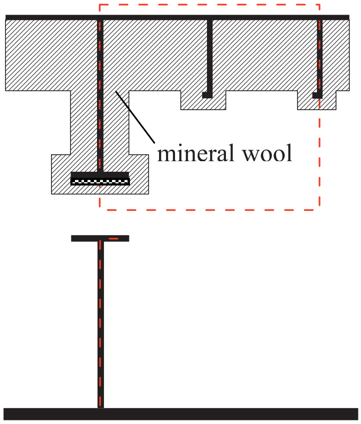


Figure 11. Variant III

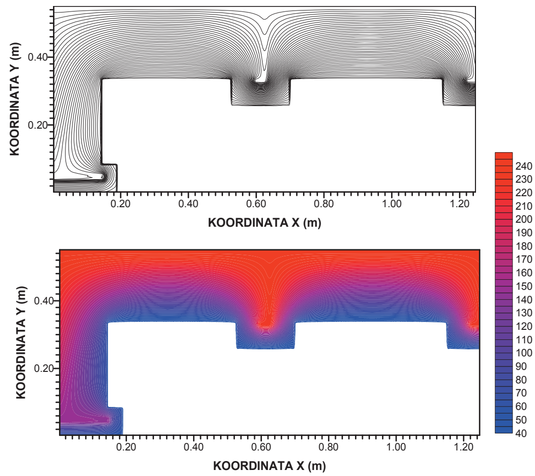


Figure 12. The surface temperature distribution across the characteristic segment of tank web frame

**Variant IV**

Insulation exists on both web frame flanges. Insulation of tank web frame flange is the same as in variant III and hull ship web frame flange is insulated with high quality insulation layer.

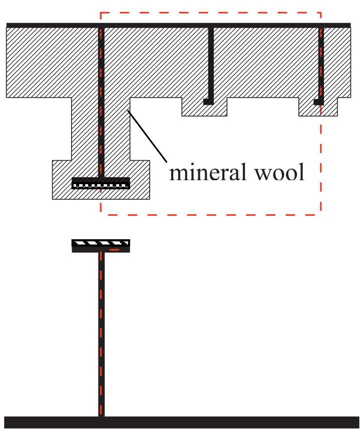


Figure 13. Variant IV

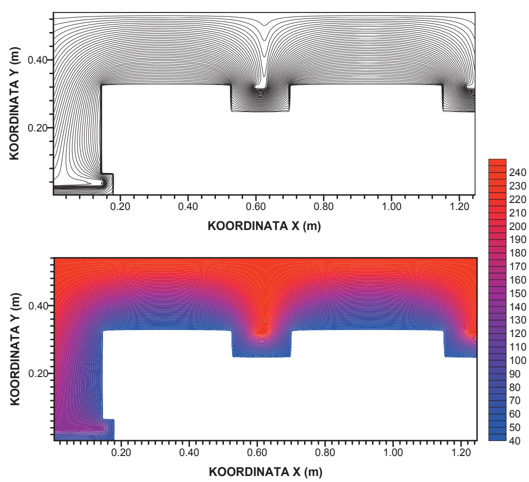


Figure 14. The surface temperature distribution across the characteristic segment of tank web frame

The temperature distributions along the web and flange of the hull ship for all four variants can be seen on figure 15.

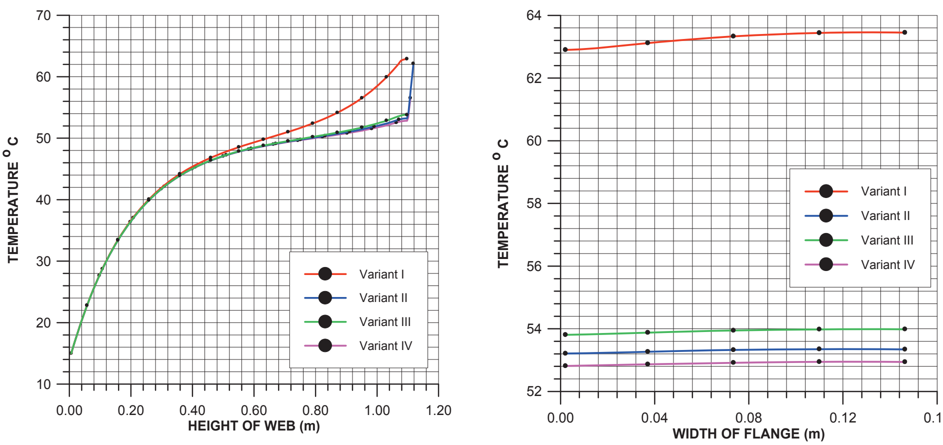


Figure 15. Temperature distributions along the web and flange (of hull ship)

## CONCLUSION

Based on the results of numerical simulations it can be concluded that variants II, III and IV satisfy the temperature conditions looked for. The lowest temperature of the flange (of hull ship) is obtained for variant IV ( 52,9 ºC).

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# 8. konferencija o termografiji

# 30 YEARS OF THERMOGRAPHY AT FACULTY OF MECHANICAL ENGINEERING AND NAVAL ARHITECTURE ZAGREB, LABORATORY FOR APPLIED THERMODYNAMICS

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## INTRODUCTION

The Laboratory for Applied Thermodynamics of the Faculty of Mechanical Engineering and Naval Architecture Zagreb started dealing with thermography in 1977. Since then new methods for application of the thermography are constantly developed.
Today, in the Laboratory two thermographic systems are used: the old AGA 680 and the new FLIR 2000SC. In the past 30 years many activities were done in the promotion of thermography for practical and scientific applications, and 30 generations of students also received elementary knowledge about this method. The thermographic methods developed are for qualitative and quantitative applications in the field of thermal and process engineering, energy production, material science, NDT methods, medicine, art, electro engineering etc.
However, 1977 was not the beginning of civil applications of infrared technology in Croatia. It started some 70 years ago with pioneering work of Croatian scientists in the field of IR photography. Figures 1 and 2 are two photographs made in the IR spectrum range, taken in around 1935.

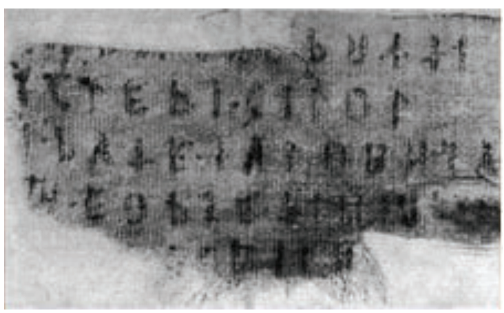

Fig. 1 Etruscan inscriptions on mummy bandages made visible by IR photography
Šplait and Weber, Zagreb, ca. 1935.

Fig. 2 Julian Alps by night taken with IR film from the mountain Sljeme,
Distance about 180 km, I.Plotnikov, Zagreb, ca 1935.

**BEGINNINGS**

The Laboratory for thermography has been established soon after the IR equipment was delivered in 1978, and the first measurements started as well as the promotion of IR thermography. It was recognized by daily newspapers with some bombastic titles, still actual today. Fig. 3 on the right: *ENERGY WASTERS: We build glass castles, we do not consider thermal insulation improvements, we do not use geothermal and waste water, and all this are means of energy savings.*

ENERGETSKA KRIZA

Rasipnici topline

● Gradimo staklene dvorce, popravljamo već otpisane instalacije, ne razmišljamo o dobroj izolaciji, ne iskorištavamo geotermalne i otpadne vode — a sve su to moguće mjere za uštedu toliko potrebne energije

Fig. 3 First laboratory for IR thermography and reflections in the press

**INTERNATIONAL COOPERATION**

International cooperation started in the early 80-ies, when researchers from the Faculty joined to scientific research projects in the field of IR thermography. The cooperation has been organized between the Technical University Prague, High Technical School Stockholm and Faculty of Mechanical Engineering and Naval Architecture Zagreb.

Fig. 4 In situ measurements (1982.) and IR measurements in laboratory (1984.)

In 1990 EUROTHERM has been established by the Committee of EU with the aim to promote Quantitative Infra Red Thermography in European Community. The Faculty has been involved in its activities from the first beginning. The cooperation results with International Conference QIRT 2002 in Dubrovnik organized by the Faculty.

## ACTIVITIES OF THE LABORATORY FOR APPLIED THERMODYNAMICS

The researchers of the Laboratory for applied thermodynamics of the Faculty utilized various IR techniques using first the AGA 680 thermographic systems and after 2004 the newly acquired FLIR 2000SC camera. They used different investigation methods for the development of new technologies, R&D, testing and inspection, restoration of antiquities, etc. Here are some exsamples:

### Development of new technologies

- Determination of subsurface defects in metal structures. It is a quantitative method combining IR thermography and numerical methods

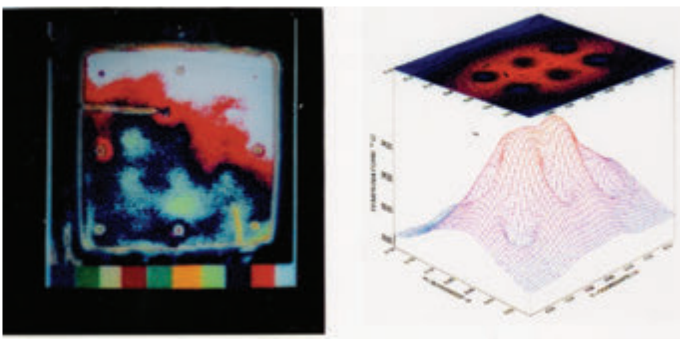

Fig. 5 Determination of subsurface defects

- Detection the buried objects. The method was developed with the goal to detect land mines. The laboratory simulations gave promising results, which have to be verified in field application.

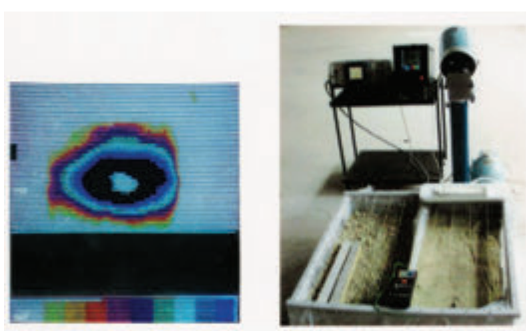

Fig.6 Detection of buried objects: thermogram and laboratory test rig

**Research and development**

- Determination of heat transfer coefficients on extended surfaces as a combination of IR thermography and numerical methods. This was used in many cases, mostly in cooperation with the industry where thermography was utilized to show functionality of new products.

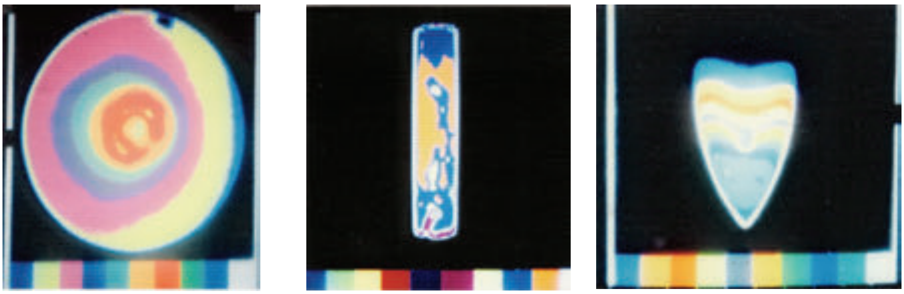

Fig. 7 Determination of heat transfer coefficients and temperature distribution

**Testing and inspection**

This is probably the widest application of thermofraphy. The Laboratory delivered a series of services to the industry and maintenance companies.

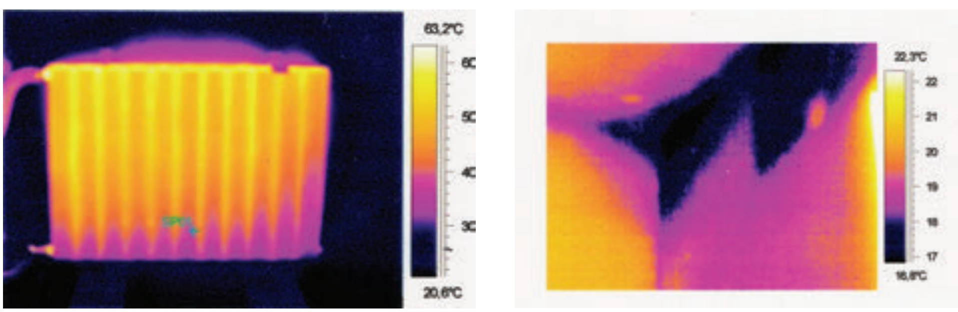

Fig. 8 Water distribution in radiator and wet zones on building wall

In 2004 the Laboratory acquired the new IR FLIR 2000SC camera, which opened new possibilities in research and development of IR techniques. One of the first activities was the analysis of the outside envelope of Jadrolinija building in Rijeka where new methods for the determination subsurface defects in façade were applied - Figure 9.

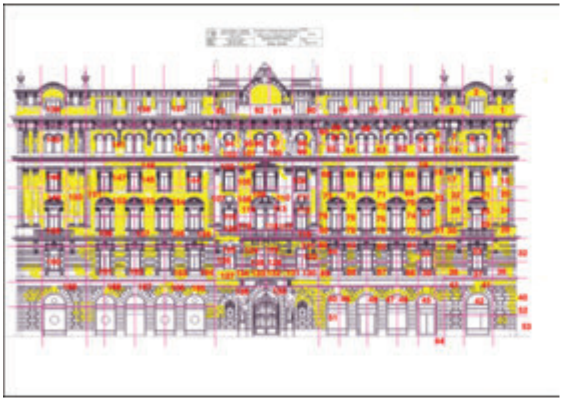

Fig. 9 The front façade of Jadrolinija building in Rijeka

Quantitative analysis of particular spots on thermograms taken on the building envelope was utilized in order to find subsurface defects as moisture penetration, air pockets etc. The results were used for the assessment of needed interventions.

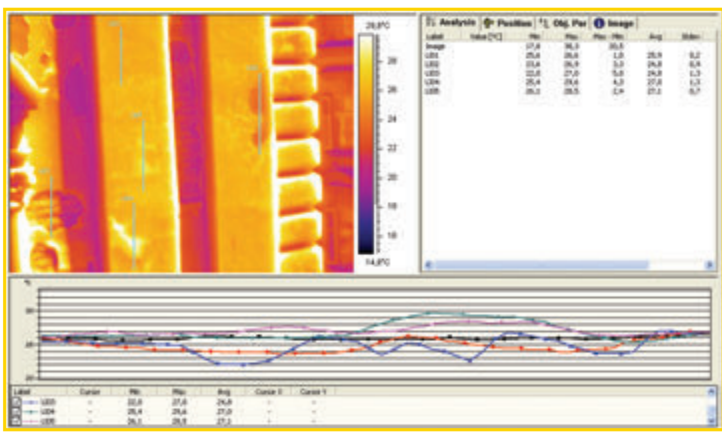

Fig. 10 Thermographic analysis of building façade

**Saving of national heritage**

A great deal of work was done in cooperation with the Croatian institute for restoration, where the Laboratory performed thermographic recording of artifacts and masonry. Thermograms of e.g. church walls revealed structural compositions behind the plaster, extent of capillary moisture penetration etc. Thermortaphy was also used in investigations of various kinds of artifacts, as wooden elements, paintings, sculptures etc.

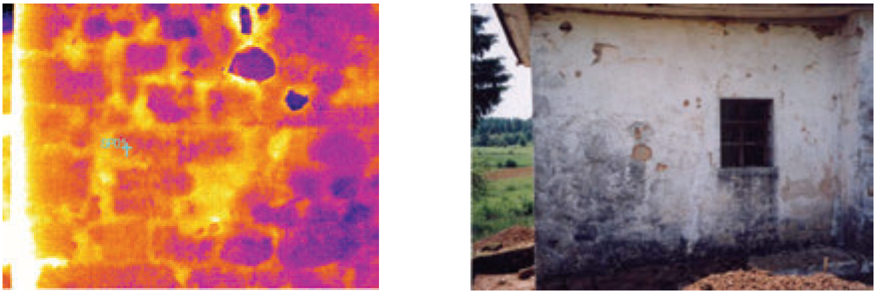

Fig. 11 Investigation of wall structure church in Mateško selo, Croatia

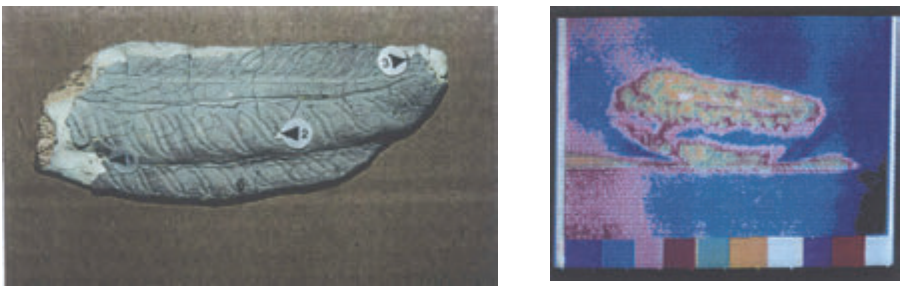

Fig. 12 Study of wooden altar element

**WHERE ARE WE STANDING TODAY**

The Laboratory has recently enlarged its premices with new space for thermographic measurements and lecturing. Preparations are made to start with Level I seminars for thermographers in June, as well as with special courses in quantitative IR thermography. The Laboratory experts are active in the Croatian Society for Thermography and have cooperation with Austrian Society for Thermography, European Society of Thermography, with the QIRT Working Group, and have contacts with Universities all over the world. They are active in presenting their research results at international symposiums and conferences like:

QIRT INTERNATIONAL CONFERENCE

BAM

V CORENDE, Patagonia, Argentina

Thermografie-Forum-Eugendorf, Austria

Thermo-MATE Budapest, Hungary

INTERKLIMA, Zagreb, Croatia

When exchanging knowledge and experience with others, the researchers of the Laboratory for applied thermodynamics of the Faculty of mechanical engineering and naval architecture of the University of Zagreb are trying to be and stay up to date in thermography.

**Fakultet strojarstva i brodogradnje Zagreb**
**Faculty of mechanical engineering and naval architecture**

**Laboratorij za toplinu i toplinske uređaje**
**Laboratory for applied thermodynamics**

# Energy audits of family houses to improve energy efficiency

# Energetski pregledi obiteljskih kuća u svrhu povećanja energetske učinkovitosti

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## ABSTRACT

Energy efficiency in buildings today is recognized as a field with the greatest potential for reducing total energy consumption, thus directly influencing a comfortable and quality living in a building, longer life of the building, as well as contributing to the environmental protection and reduced greenhouse gas emissions. Energy efficiency measures in buildings include a range of different possibilities of energy and heat savings, including the use of renewable energy sources where it is functionally and economically feasible. One of the features of a large part of family houses and residential buildings in Croatia is unreasonably high consumption of all energy forms, the heating energy in the first place, but lately also the energy for cooling purposes, which has recorded a growing trend.

Modern building energy management today includes a comprehensive evaluation of almost all energy systems within a facility. Energy auditing of buildings can range from a short walk-through of the facility to a detailed analysis with infrared thermography, analyses of the energy bills, and different measurements.

Energy Institute Hrvoje Požar conducted a large number of energy audits of family houses and residential buildings. For each house, according to specific location, infrastruction availability and specific demands, measures for efficient use of energy are calculated with preview of total investment and simple pay-back period. Analysing the results, it is clear that energy efficiency measures in overall investment for new and existing buildings are about 100 € per $m^2$, that is about 12% additional investment with pay back period of 6 to 12 years. Energy efficiency measures recommended in these houses include thermal insulation of entire building envelope, high quality windows frame and glazing, use of high efficient HVAC systems, heating and cooling regulation, high efficient lighting and use of renewable energy sources.

**KEYWORDS**
Energy efficiency in buildings, energy audit, reducing energy consumption, environmental protection

## INTRODUCTION

Energy audit is an analysis of thermal performance and energy systems of building with the purpose to determent its energy efficiency or non-efficiency. Energy audit also helps getting new conclusions and suggestions on how to increase the energy efficiency.

Main goal of energy audit is to access and process collected data, and to get as much accurate present energy performance of building, concerning construction characteristics in terms of thermal protection, quality and efficiency of heating, ventilation and cooling systems, quality and efficiency of lighting and household appliances and building management. After data has been processed, the most suitable economical and energy efficient measures will be suggested. Depending on the data level and its accuracy we distinct:

- Preliminary or walk through energy audit
- Detailed energy audit with feasibility study

Preliminary energy audit includes short input of energy condition in the building and its main objective is to determine its potential to increase the energy efficiency and to execute detailed energy audit. Visual observation of the buildings´ envelope and its energy systems with short analysis of collected data shows the key problems and gives recommendations for improving energy efficiency. After that, the decision if there is a necessity for detailed energy audit and investment study should be made.

Detailed energy audit consists of more detailed energy analysis of the building and identification of potential measures of energy efficiency, through conversation with its owners or management of the building and review into existing documents. Utility bills will be reviewed for at least 12 or optimal 36 month period to evaluate the facility's energy demand. Through detailed energy audit and execution of consumption measuring –heat and electricity, infrared thermography, air permeability of the building – the key problems are indicated and introduced to the owners or management of the building.

If the investment of increasing energy efficiency in the building shows energetically, economically and ecologically worthy, the project could apply for co-financing or financing according to conditions in some banks. Such application consists of business and financial plan with project description, its goal and final achievement, taking in consideration acceptable pay back period. Introducing classification, energy certification and grading according to energy consumption in buildings, energy audit of buildings should become unavoidable energy efficiency method, and data for future energy certification of the buildings. The following lines will describe experiences of the energy auditing and useful methods for implementing energy efficiency improvement of the new and existing buildings.

## ENERGY AUDITS OF BUILDINGS

Energy Institute Hrvoje Pozar (EIHP) has made a survey for gathering data important for execution of different energy audits in various buildings, which consists of:

- general data about the building – its type, purpose, year of the construction, year of the reconstruction, climate data, ownership
- constructions characteristic – total surface and heated area, window frame type and glazing, external wall, roof and floor type

- energy indicators –energy consumption for heating and cooling on a monthly basis, electricity consumption, characteristics of the heating, hot water and cooling systems, ventilation system and all other energy demand, including passive heat gains
- living comfort annotations from the owners or management of the building
- final conclusion and suggestion measures in at least two categories
  - smaller investment expenses and fast implementation
  - more expensive investment with obligation to conduct detailed energy audit and feasibility study

As a part of the UNDP project “Removing barriers for improving energy efficiency of the residential and service sectors” and financial support of the UNDP Croatia, EIHP conducted 10 energy audits of family houses all over the country. For each house the owners filled out a general data form concerning external envelope and construction condition, and other characteristic of the energy systems. The energy bills from 3 years ago were taken into account, and some interviews with the residents were done in order to get the input of their habits, needs and to get the building condition in general. After acknowledgment of the future energy issues, wishes and real energy needs, there was a set of executable variations on how to increase the energy efficiency, taking into consideration living comfort as well.
These variations are:

- to improve thermal performance of external envelope using thermal insulation
- to exchange or to improve energy efficiency of the heating system
- to exchange or to improve energy efficiency of the cooling and ventilation system
- to exchange or to improve hot water production system
- change of fuels if it is economically and ecologically exculpate
- implementation of the renewable energy sources (sun, geothermal, biomass,)
- increasing energy efficiency of the lighting and household appliances
- rationally water consumption
- building energy management

For each variation there was ascertain technical feasibility, total system compatibly, possible energy savings, financial evaluation and possible economical savings. Based on comparison of each variation and its investment worth, the evaluation report was made and it gave possible solution suggestions. As the energy obtained from fossil fuels decreases, it reduces harmful emissions for the environment, especially $CO_2$ emissions. Economical savings were calculated based on earlier estimated energy savings, taking into account net present value.

According to the analysis, in the north part of Croatia, natural gas for heating is used more than other fuels. There is some electric energy consumption, mostly for cooling purposes and hot water production and firewood for heating. Near the Croatian coast, for heating purposes people use more fuel oil for heating or electricity for heating and cooling. For illumination classical bulb is still in common use instead of energy saving bulbs. Conducted energy audits estimate reduction of heat energy consumption from 150-250 kWh/m$^2$ to 60-90 kWh/m$^2$. Recommended energy efficiency measures in analyzed houses include thermal insulation of entire building envelope, high quality windows frame and glazing, use of high efficient HVAC systems, heating and cooling regulation, high efficient lighting and use of renewable energy sources. Heat energy savings are higher, if low energy and passive buildings standard is executed. This means reducing the total amount of heat energy to 40 kWh/m$^2$ for low-energy and to 15 kWh/m$^2$ for passive houses standard. It is more convenient with new buildings, with recommendations based on the project documentation.

LOCATION
Velika Gorica
BUILDING DESCRIPTION
Family house with 2 appartments 3 storeys
TOTAL FLOOR AREA
253 m$^2$
TOTAL HEATED AREA
176 m$^2$
YEAR OF CONSTRUCTION
In 1978, refurbishment in 1996
CONSTRUCTION TYPOLOGY
Hollow brick walls 25 cm thick, prefabricated brick ceillings, wooden roof construction, wooden frame double windows with 2 glass panes
RECONSTRUCTION / ANNEX
Refurbishment of the existing house
YEAR OF RECONSTRUCION / ANNEX
Planned ground floor annex 77 m$^2$
CONSTRUCTION TYPOLOGY
Hollow brick walls 19 cm thick , wooden roof construction, wooden frame windows with insulation glass
ENERGY-GENERATING PRODUCT
Natural gas, electricity, fire wood, liquid petroleum gas for cooking
HEATING SYSTEM AND DHW
Boiler 35 kW for radiator central heating, electric heater 1 kW, fire wood stove, fireplace, split system for heating and cooling 2,5 kW, domestic hot water electric heathers localy
CURRENT HEAT ENERGY CONSUMPTION
185 kWh/m$^2$
RECOMENDED HEAT ENERGY SAVING MEASURES
Thermal insulation with mineral wool or polystyrene of external walls with 10 cm, ceilling toward attic with 16 cm, ground floor with 10 cm. New windows with new value U=1,4 W/m$^2$K. Solar collectors for domestic hot water, heating system control.
REDUCTION OF HEAT ENERGY CONSUMPTION
105 kWh/m$^2$
$CO_2$ EMISSION SAVINGS
8,5 t/y
EXPECTED HEAT ENERGY CONSUMPTION
80 kWh/m$^2$
YEARLY SAVINGS
cca 1.370 €
INVESTMENT
cca 10.800 €
SIMPLE PAY-BACK PERIOD
7,9 y

LOCATION
Krk
BUILDING DESCRIPTION
Family house with 3 appartments 2 storeys
TOTAL FLOOR AREA
141,20 m$^2$
TOTAL HEATED AREA
68,20 m$^2$
YEAR OF CONSTRUCTION
In construction
CONSTRUCTION TYPOLOGY
Porotherm block walls 25 cm thick, prefabricated brick ceilling and roof construction, thermal insulation with polystyrene of: external wall with 5 cm, roof with 5 cm, ground floor with 7 cm
ENERGY-GENERATING PRODUCT
Electricity
HEATING SYSTEM AND DHW
Electric heaters and split systems for heating and cooling, fireplace in the ground floor.
CURRENT HEAT ENERGY CONSUMPTION
90 kWh/m$^2$
RECOMENDED HEAT ENERGY SAVING MEASURES
Thermal insulation with mineral wool or polystyrene of external walls with 14 cm, roof with 16 cm, ground floor with 10 cm, floor to first floor with 8 cm.Windows with LOWe glass coating and new value U=1,1 W/m$^2$K. For ground floor appartment ventilation system with heat pump, dual split system 7 kW. For first floor appartments 2 mono split systems 3,5 kW, hot water production solar collectors 12 m$^2$ and 300 L tank.
REDUCTION OF HEAT ENERGY CONSUMPTION
68 kWh/m$^2$
$CO_2$ EMISSION SAIVNIGS
1,3 t/y
EXPECTED HEAT ENERGY CONSUMPTION
22 kWh/m$^2$
YEARLY SAVINGS
cca 930 €
INVESTMENT
cca 13.900 €
SIMPLE PAY-BACK PERIOD
14,9 y

Illustration 1: Results of energy audits for achieving standard insulated house compared to low-energy house /EIHP/

Concerning for home energy efficiency is correct approach to reduce total energy consumption. Analysing the results, it is clear that energy efficiency measures in overall investment for new and existing buildings are about 10-12% additional investment with pay back period from 6 to 12 years. Best results in energy savings are through improving thermal protection of buildings and using high efficient heating, ventilation and cooling systems.

## CERTIFICATION OF BUILDINGS – DOCUMENT ON ENERGY CHARACTERISTIC OF BUILDINGS

On 1st of July, 2005, new technical regulations on heat energy savings and thermal protection of buildings (Official Gazette, 79/2005) were accepted, representing a big improvement in the thermal protection of buildings in Croatia. The new technical regulations came into force on 1st of July, 2006 and have been observed as mandatory ever since. These regulations consist of:

- Technical demands on heat energy savings and thermal protection to achieve in design of new buildings and refurbishment and reconstruction of existing ones which are heated on space temperature higher than 12°C
- Project content regarding heat energy savings and thermal protection
- Statement of required heat energy for heating
- Building maintaining regarding heat energy savings and thermal protection
- Technical demands for building products
- Other technical demands on heat energy savings and thermal protection

The new aforementioned technical rule determines the maximum allowed annual heat demand per $m^2$ of the building $Q_h$ (expressed in $kWh/m^2a$), depending on the form factor of the building, i.e. the ratio between the area of the building's envelope (heated space) and the building's volume. The heat transfer coefficient for windows and balcony doors in buildings heated to the temperature of 18°C and above is limited to a maximum of $U=1.80\ W/m^2K$.

Building energy balance according to HRN EN 832:2000 + HRN EN 932/AC:2004 includes:

- Transmission and ventilation losses through windows from inside to outside area
- Transmission and ventilation losses through ventilation and heat gains from boarding zones
- Useful internal heat gains from internal heat sources
- Useful heat gains from sun
- Heating system losses
- Energy for heating

Based on calculations of the thermal performance of a building, a certificate on the required heat energy for heating will be made. This energy certificate includes notice of the required heat energy for heating stated by the designer in the main design on heat energy savings and thermal protection and verified by the contractor. The statement of the contractor confirms that the work performed in the building, or a part of the building, has been carried out in accordance with technical solutions and conditions of construction relating to the heat energy savings and thermal protection and with rules of the Technical regulation. The contractor's statement is signed by the head engineer of the building site. This energy statement is enclosed with technical documentation required for the technical inspection of a building, or a part of the building, and it makes an integral part of documentation on maintaining and improving the essential requirements on the building. It should be available to prospective buyers, tenants and other authorised customers of a building or its part. This legislation is a good beginning to future energy certification of buildings.

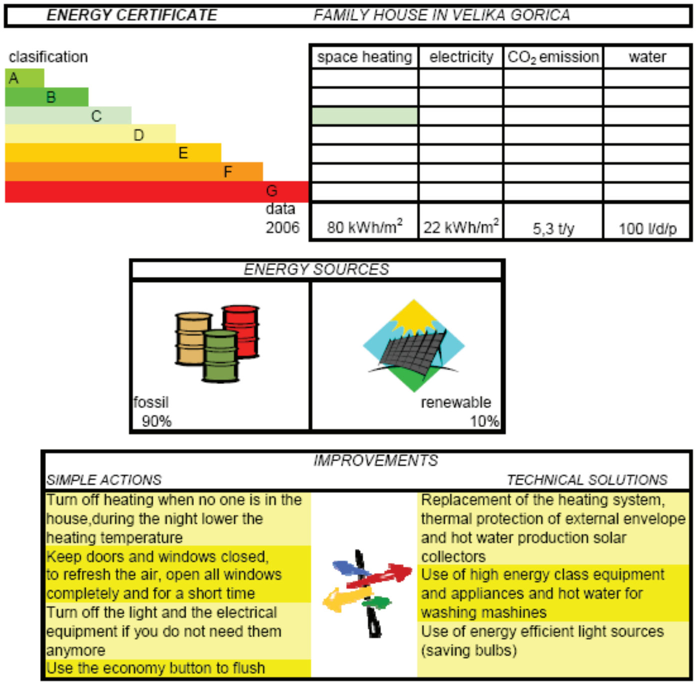


Illustration 2: Proposal appearance of energy certificate /EIHP/.

Next step in Croatia is establishing a frame for energy certification of buildings which includes preparation of the documented procedures and tools (software) and further general preparation of information campaign. Republic of Croatia, jurisdictional Ministry at the present is in the process of forming a Committee for implementation of Directive 2002/91/EC on Energy performance of Buildings in national legislation, with obligatory energy certification of buildings. Key issues of implementation are necessary law and regulation changes, preparing quality action plan for the implementation of the directive, forming building registry and education of energy experts trained for energy audits of buildings. The Directive on the energy performance of buildings is a major importance for the buildings sector and brings crucial changes for all participants in the process of design and construction.

The Directive asks for existing buildings with useful area larger than 1000 $m^2$ for which refurbishment is planed, improvement of minimum energy characteristics when ever is technically, functionally and economically feasible. Member states need to provide trained experts for building certification, boilers inspection, ventilation and air conditioning systems and drafting of recommendations for system improvements in respect to energy saving and limiting carbon dioxide emissions. The main goal of the Directive 2002/91/EC on the energy performance of buildings is to oblige member states to necessary reduction of final energy consumption in new and existing buildings. New buildings must be constructed to meet required minimum energy conditions. For new buildings, with useful area larger than 1000 $m^2$ technical, environmental and economic feasibility of alternative systems such as; decentralised energy supply systems based on renewable energy, cogeneration, district heating or cooling, heat pumps, etc. is considered and taken into account before construction starts. On going member states must offer the necessary measures to establish a regular inspection of boilers fired by non-renewable liquid or solid fuel, with output of 20 kW to 100 kW. For boilers larger than 100 kW inspections should be every 2 years. For gas boilers this period may be extended to four years. For heating installations with boilers older than 15 years measures to establish a one off inspection of the whole heating installation must be prepared. With regard to reducing energy consumption and limiting carbon dioxide emissions, necessary measures to establish a regular inspection of air conditioning systems with output of more than 12 kW must be created. Inspection shall include system efficiency classification. Member states must ensure that the certification of buildings and inspections shall be carried out in an independent manner by qualified experts.

When buildings are constructed, sold or rented out, an energy performance certificate will be available to the owner or by the owner to the prospective buyer or tenant. Energy certificates with data on yearly basis energy consumption will be displayed. Easy comparison of energy characteristics of buildings will enable the building industry to use this information as marketing tool. Introducing energy certification of buildings, properly insulated buildings with low energy consumption will be highly valuable on the market while for non insulated buildings value will decrease. This will progress the market to improving energy efficiency. Energy certificate can be based on measured or calculated values or use both indicator groups when available. Indicators assessed by calculation show true potential of a building while measured value adds building maintenance practice. Defining standard calculated energy indicators includes building data (insulation, HVAC system, lighting, etc.), which will be useful for drafting recommendations on energy characteristics improvement. For new buildings only calculated values are available. Measured indicators for existing buildings are easily available from electricity and heat energy bills. In public buildings measured indicator is maintenance practice evaluation and motivation to operator and users. Measured and calculated indicators don't give necessary the same results. Energy certificate must have easily understandable indicator of overall energy consumption. Overall consumption will be divided into energy classes which will determine quality of building according to energy, economy, ecology and indoor climate characteristics. In particular the use of renewable energy sources should be specified as the amount of produced energy. In energy auditing and building energy characteristics assessment, useful methods are infrared thermography and determination of air permeability of buildings.

## IMPORTANCE OF INFRARED THERMOGRAPHY IN FUTURE CERTIFICATION OF BUILDINGS

From conducted analyses, it is clear that the major part of overall energy consumption in buildings is a heat comfort assessment. Infrared (IR) termography is shown as especially useful method for visualisation of heat losses through constructional elements in improving energy efficiency of buildings survey. Termography inspection of buildings and expert interpretation of possible construction defects, are located and refurbishment actions directed to improve energy efficiency. Construction defect displayed by termography are non homogeneous wall material, incorrect or non existing thermal insulation, damp in construction, flat roof problems, thermal bridges, open air ducts and ventilation, slots, installations in walls and floors, etc. Wireless and distance temperature field scanning of the building has major advantages for common construction analysis. Introduction of IR termography in buildings is equally useful in energy auditing of existing buildings, historic buildings under protection, as in quality control of new buildings. Based on this, in developed countries IR termography is implemented as obligatory method in technical characteristics quality control, building maintenance and management of public buildings in particular. In numerous energy certificates as calculated analyses control an IR scan is attached to visualise quality or defects of a building. The aim to expand further use of IR termography in buildings is:

1. auditing and quality evaluation of new buildings through heat efficiency construction characteristics assessment
2. introducing IR thermography as a standard method for efficient maintenance of existing buildings
3. determination of building energy efficiency by measuring thermal losses through external envelope
4. improving building sector in the field of designing and constructing from the point of energy saving

Introducing energy certification of buildings in the future a significant use of IR termography is expected, as in energy certification of existing buildings and quality control of new buildings.

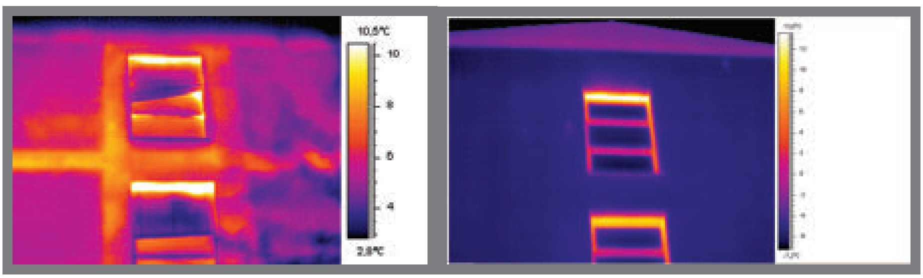


Illustration 3: Comparison of infrared thermography picture in energy auditing before reconstruction (left) and after reconstruction (right) according to energy efficiency measures /EIHP/.

## IMPORTANCE OF AIR PERMEABILITY OF BUILDINGS – FAN PRESSURIZATION METHOD

The fan pressurization method is intended to characterize the air permeability of the building envelope or parts of it. The system is composed of a large framed fan, temporarily fixed to doors with function to measure air permeability of a building and determining leakage areas. It can measure the air permeability of a building or part it for compliance with a design air tightness specification; to compare relative air permeability of several similar buildings or parts of buildings; to identify the leakage sources and to determine the air leakage reduction resulting from individual retrofit measures applied incrementally to an existing building or part of building. Today an average house has 1/3 of thermal losses through walls and ceilings, 1/3 through windows and doors and 1/3 through air leakage. This method doesn't measure the air infiltration rate of a building; the results of the fan pressurization test can be used to calculate the air infiltration. This method applies to measurements of air flow through the construction from outside to inside or vice versa. Survey is followed by a report with information:

- All details necessary to identify the building (or object) tested, purpose and test method
- A reference to standard EN 13829 and any deviation from it
- Test object:
  - Description of which parts of the building were subject to the test

- Net floor area, internal volume of the space and other required dimensions of the building
- Documentation of calculations – so that the stated results can be verified
- The status of all openings on the building envelope (latched, sealed, open, etc.)
- Detailed description of temporarily sealed openings, if any
- The type of HVAC system

- Equipment and technique employed

- Test data:
  - Zero flow pressure differences for pressurization and depressurization
  - Inside and outside temperatures
  - Wind speed, barometric pressure if it is part of the calculation
  - Table of induced pressure differences and corresponding air flow rates
  - Air leakage graph
  - The air flow coefficient, $C_{env}$, the air flow exponent, $n$ , and the air leakage coefficient, $C_L$ , for both pressurization and depressurization tests
  - Air change rate, $n_{50}$, at 50 Pa, for pressurization and/or depressurization and mean value
  - Derived quantity according to national regulation

There are two types of test method; the condition of the building envelope should represent its condition during the season in which heating or cooling systems are used or, any intentional opening in the building envelope will be closed or sealed. Fan pressurization method is useful in energy auditing for in a brief survey assumes condition of an external envelope. Measures to improve air permeability of buildings are simple; as windows, doors and partition sealing, including thermal protection and avoiding thermal bridges. All this contributes to improving internal climatic conditions, maintenance and durability of constructions and reducing energy consumption.

## CONCLUSION

There are many positive effects of the measures contributing to increased energy efficiency; they are successful if they meet 3 E system demands: energy, economy and ecology. Energy certification assumes the need for energy consumption measuring. This way the energy quality of constructions will be examined, as applied measures of energy efficiency: thermal insulation of entire building envelope, high quality windows frame and glazing, use of high efficient HVAC systems, heating and cooling regulation, high efficient lighting and use of renewable energy sources. With constant energy audits technical characteristics of new constructions and materials will be monitored and their value estimated according to indoor climatic conditions. Presented methods assume invasive inquiry of energy characteristics with reliable results. Analysing the results, it is clear that energy efficiency measures in overall investment for a new building or reconstruction are about 10-12% additional investment with pay back period from 6 to 12 years. In Croatia today, pay back period for the improvement of thermal protection and use of renewable energy sources could be too long. However, as energy prices rise and with the financial support of renewable energy sources usage, in the future economic analyses will be more convenient with shorter pay back period.

**SAŽETAK**

Energetska efikasnost u zgradarstvu prepoznata je danas kao područje koje ima najveći potencijal za smanjenje ukupne potrošnje energije, čime se direktno utječe na ugodniji i kvalitetniji boravak u zgradi, uz duži životni vijek zgrade, te doprinosi zaštiti okoliša i smanjenju emisija štetnih plinova u okoliš. Mjere energetske efikasnosti u zgradarstvu, uključuju cijeli niz različitih područja mogućnosti uštede toplinske i električne energije, uz primjenu obnovljivih izvora energije u zgradama, gdje god je to funkcionalno izvedivo i ekonomski opravdano. Jedna od karakteristika velikog dijela obiteljskih kuća i stambenih zgrada u Hrvatskoj je neracionalno velika potrošnja svih tipova energije, prvenstveno energije za grijanje, ali sve više i za hlađenje.

Suvremeno upravljanje energijom u zgradama uključuje široku analizu svih energetskih sustava zgrade. Energetski pregledi zgrada mogu biti kratki, preliminarni pregledi pa do detaljnih energetskih analiza uz korištenje infracrvene termografije, analizu računa za energiju i razna mjerenja.

Energetski institut Hrvoje Požar proveo je veliki broj energetskih pregleda obiteljskih kuća i stambenih zgrada. Za svaku kuću, prema karakteristikama lokacije, raspoloživosti infrastrukture i potrebama, predložene su mjere povećanja energetske efikasnosti uz pregled troškova i jednostavni period povrata ulaganja. Analizirajući rezultate možemo zaključiti da mjere energetske efikasnosti u odnosu na ukupne troškove gradnje ili rekonstrukcije iznose oko 100 eura po kvadratnom metru, ili oko 12 posto dodatnog ulaganja, uz period povrata od 6 do 12 godina. Preporučene mjere energetske efikasnosti u analiziranim kućama uključuju toplinsku zaštitu cijele vanjske ovojnice, visokokvalitetne prozorske profile i stakla, visokoefikasne sustave grijanja, ventilacije i klimatizacije, sustave regulacije grijanja i hlađenja, visokoefikasnu rasvjetu i korištenje obnovljivih izvora energije.



**KLJUČNE RIJEČI**
Energetska efikasnost u zgradama, energetski pregled, smanjenje potrošnje energije, zaštita okoliša


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# Quality Control of Wood-based and Composite Materials by Infrared Thermography

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ABSTRACT

The wood-based panel companies face their customer's increasing demand for high and constant quality of their products and for flexible production. At the same time, the cost pressure gets stronger. Quality assurance systems following standards such as ISO/DIN/EN 9000 become more and more important and require a careful monitoring of the production process and preventive measures to avoid production failures. In the view of high production speeds and the large number of different products, process integrated non-destructive testing methods are necessary for these tasks. Some non-destructive testing methods such as ultrasonic or optical techniques are already commercially available. However, these techniques do not always meet the high demands with respect to speed, reliability, and failure detection rate. Consequently, new techniques such as infrared thermography are attracting more and more interest.

## PASSIVE THERMOGRAPHY FOR QUALITY CONTROL OF WOOD-BASED PANELS

Since wood-based panels are mostly manufactured by hot-pressing, quality control can in many cases be achieved using passive heat flow thermography. In passive heat flow thermography the cooling of the objects to be tested is observed with an infrared camera typically installed directly behind the press (Figure 1).

Many defects differ in thermal capacity and/or conductivity from the good areas and thus become apparent by different surface temperatures during the cooling process. If for example there exists a delamination, the surface above this spot cools down faster because due to the lower heat conductivity of the defect heat diffuses more slowly from the interior of the panel to the surface than in the good parts of the panel (Figure 2).

Typical defects detectable this way are:

- Blisters in particle boards
- Delaminations between wood-based panels and decor paper
- Defective adhesive joints in plywood
- Fallen-off knots in the middle layers of plywood
- Improper adhesive quantity
- Surface damages
- etc.


Figure 1: Set-up for passive thermography behind the press

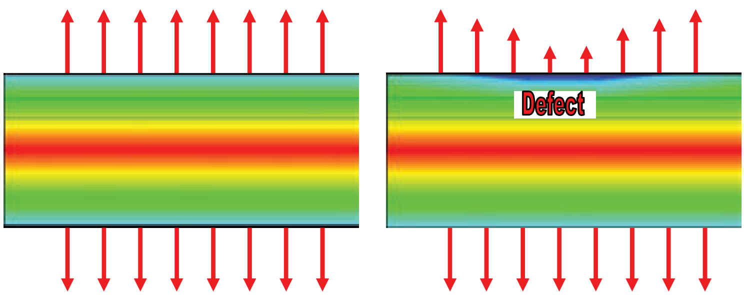



Figure 2: Principle of passive thermography

Thermography was used successfully for the examination of wood-based panels with thicknesses between 3 mm and 38 mm and at production speeds of up to 30 m/min both on continuous and on discontinuous presses. The surface temperatures of the wood-based panels can vary between 40 °C (coating with decor papers) and more than 100°C (OSB production). Figure 3 shows as examples infrared images of blisters in particle boards of 22 mm thickness which were taken after cladding with decorative papers.

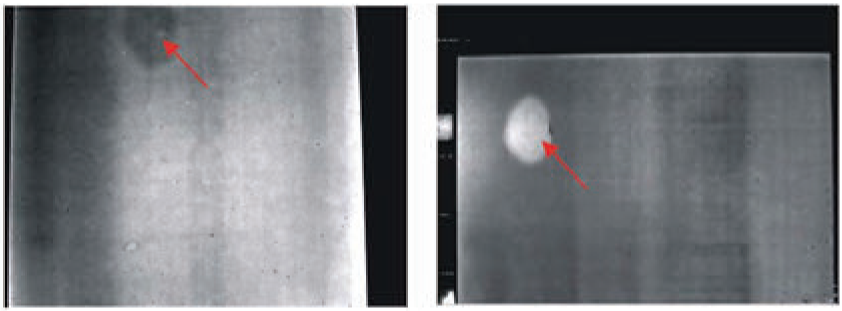


Figure 3: Infrared images of blisters in particle boards of 22 mm thickness. The dark (cool) spot in the left part shows a blister in the lower layer, while the light (warm) spot in the right part represents a blister in the upper layer.

Infrared images of particle boards can be easily evaluated automatically because of their homogeneous structure. In plywood the situation is more complicated because in the infrared image the natural wood structure is still visible after cladding with decorative papers. Nevertheless, fallen-off knots or gluing errors can be detected (Figure 4).

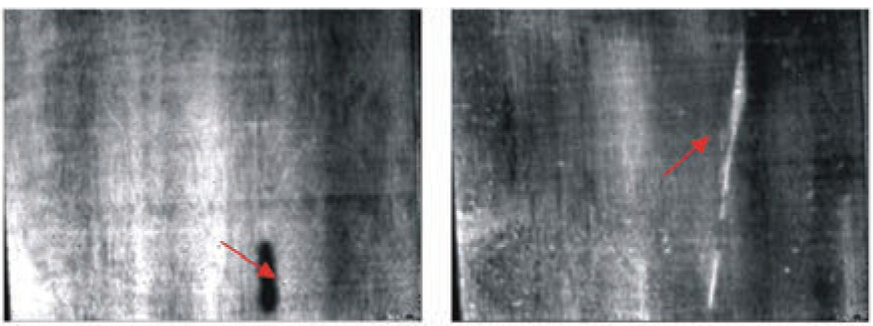


Figure 4: Infrared images of plywood boards of 5 mm thickness. The left image shows a fallen-off knot in the middle layer, while the right image shows imperfect gluing.

A further field of application of passive thermography in the wood-based panel industry is the detection of variations in density and moisture content in the manufacture of oriented strand boards (OSB). Such inhomogeneities involve respective differences in thermal capacity leading to a different cooling behaviour. Figure 5 shows a photography (left) and an infrared image (right) of a freshly pressed OSB laboratory board with intentionally inserted defects such as excessive moisture (upper right part of the board) and excess density (lower left part of the board). The infrared image was taken at a time where the thermal contrast between defect and good material was maximized.

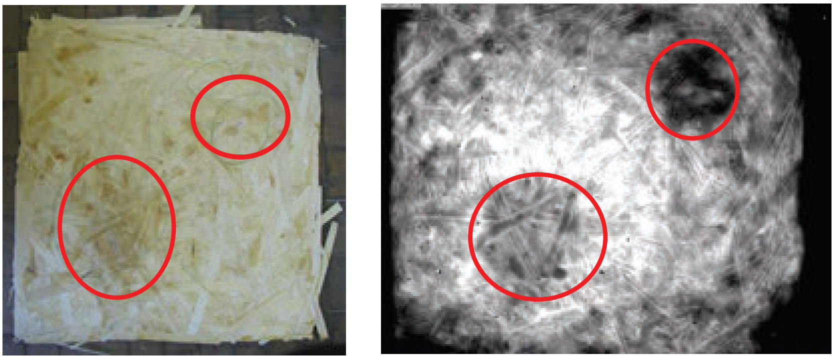


Figure 5: Freshly compressed OSB laboratory board with intentionally placed defects such as excessive moisture (upper right part of the board) and excess density (lower left part of the board)

After a suitable calibration the density measurement can be accomplished quantitatively. To this end a number of test boards is first characterized by thermography and then sawn into small pieces, the density of which is determined by weighing. The density distribution obtained this way corresponds both qualitatively (Figure 6) and quantitatively (Figure 7) very good with the grey values of the infrared image.

## ACTIVE THERMOGRAPHY FOR QUALITY CONTROL OF WOOD-BASED MATERIALS

If no process-related cooling processes can be made use of, the active variant of heat flow on-line thermography can be utilized (Figure 8). In this case the boards are transported on a conveyor belt and pass an infrared heater and subsequently the infrared camera. Important applications of this technique are the testing of plywood regarding gluing defects and fallen-off knots as well as of high-quality solid wood (wood for musical instruments or pencils) regarding density inhomogeneities. By means of ultrasonic excitation it is also possible to detect black knots in solid wood which cause problems because they can fall out easily. The ultrasonic excitation causes frictional heat in the black knots which can easily be detected by thermography. Good knots give no signal (Figure 9). The black knot appears as a light area in the infrared image.

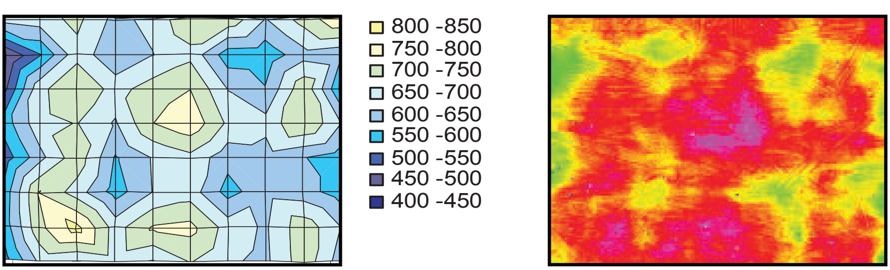



Figure 6: Comparison of gravimetrically determined density distribution (left) and infrared image (right) of an OSB- laboratory board. The density scale is kg/m$^3$

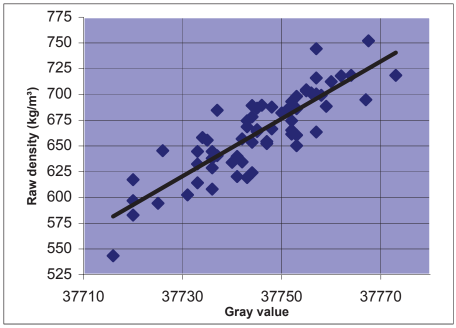



Figure 7: Quantitative comparison between density and gray value

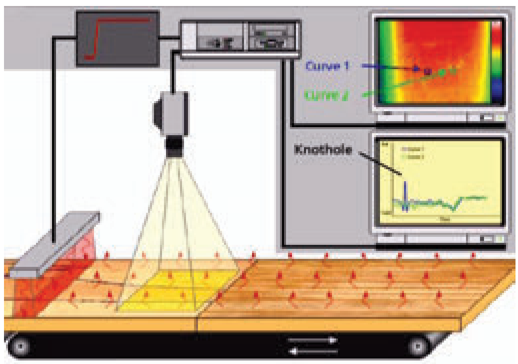



Figure 8: Principle of active online heat-flow thermography

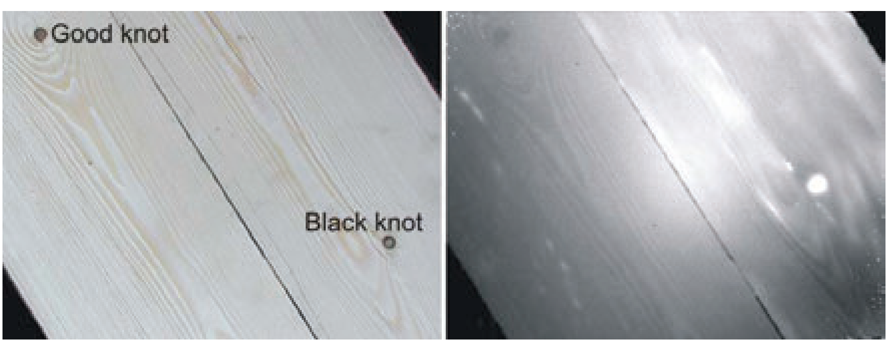



Figure 9: Solid wood piece with good (left top) and black knot (right above). The black knot appears as a light area in the infrared image

QUALITY CONTROL ON ROTOR BLADES OF WIND TURBINES

Different composite materials such as glass-fibre reinforced plastics (GRP) can be examined in the same way as wood. An important example for this material group and at the same time for very large test pieces are the rotor blades of wind turbines, which consist predominantly of GRP, but partially also of wood.

Rotor blades of wind turbines are highly stressed by frequently changing loads such as wind forces, weight forces, centrifugal forces and forces of inertia. They are also subjected to erosion by air particles and to lightning. These environmental influences result in different damages of the rotor blades. Possible defects in rotor blades are defective adhesive joints between shell and spar, air inclusions, delaminations, and the penetration of humidity or hydraulic oil. The damage spectrum reaches from losses in yield to the total loss of the turbine (Figure 10). Consequently, rotor blades must be checked carefully after manufacturing regarding production defects. They also have to be examined regularly on-site. In view of the offshore plants with problems in access this is of special importance.

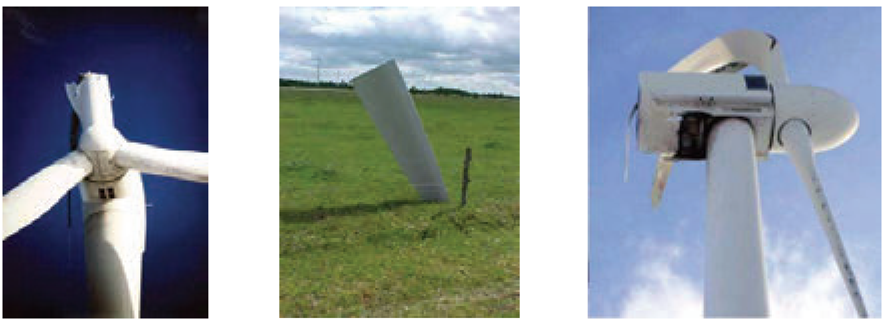


Figure 10: Damages due to defective rotor blades

In contrast to the strong requirements on the blades the existing testing methods are so far limited to visual inspections and the so-called tapping test. By means of infrared thermography the examination of rotor blades can be substantially simplified and improved.

Here one also has to differentiate between passive and active procedures. Passive procedures can use process-related cooling processes in a similar way as for wood-based materials, but in the case of rotor blades additionally the heat of reaction during hardening of adhesive joints can be utilized. This is illustrated by a look into a freshly manufactured rotor blade (Figure 11). In such a way defects located deeply in the material can be detected. This is especially true for defective joints at the leading edge, which are otherwise difficult to locate (Figure 12). A further possibility of passive inspection of wind turbines is given by the fact that in the turning rotor blades frictional heat is formed at defective spots, which can easily seen in the infrared camera (Figure 13).

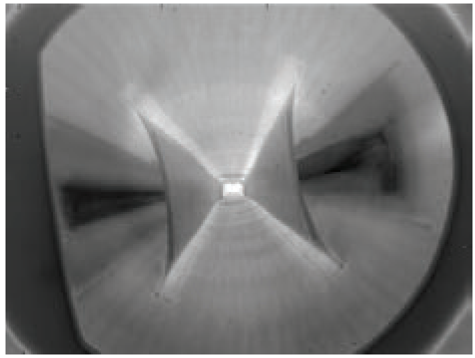
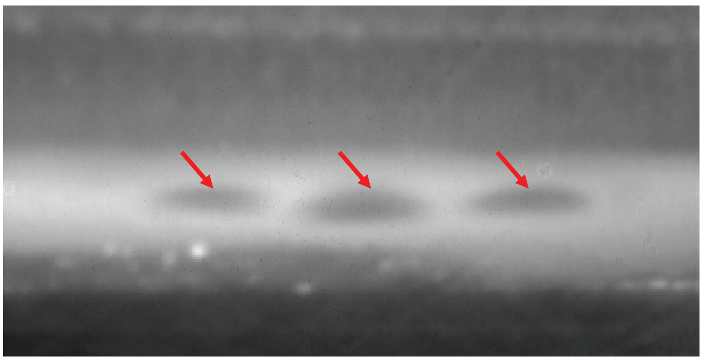


Figure 11: Thermography at the inside of a freshly manufactured rotor blade. The infrared image (right) shows the heat of reaction of the adhesive



Figured 12: Infrared image of a defective joint on the leading edge of a freshly manufactured rotor blade

The utilization of the temperature variation in the course of the day represents a limiting case between passive and active thermography. In principle it is a matter of active thermography with a particularly long heat impulse, but the examiner does not intervene in the system itself and observes the rotor blade passively, similarly to building thermography. In this way an insight deep into the structure can be obtained. This is demonstrated in Figure 14 which shows a rotor blade stored outside. The image was taken shortly after sunset on a cold, but sunny winter day. The internal structure of the rotor blade is clearly recognizable as well as an ice formation in the inside, which is due to penetrated water which could not drain off in consequence of wrongly placed drainage drillings.

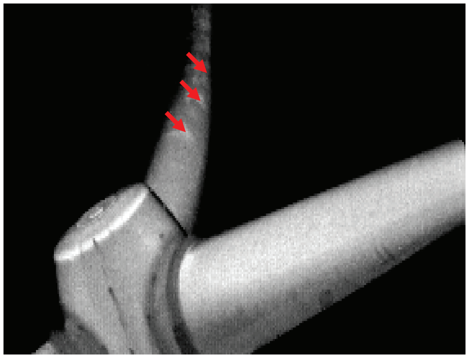


Figure 13: Thermography on a turning rotor blade with defects on the front edge

Figure 14: Thermography image of a rotor blade stored outside taken shortly after sunset.

The active variant of online thermography can be transferred without any problem to very large test pieces such as rotor blades as well. In this case the objects stand still while the measurement setup moves alongside same. The WKI has developed a test system allowing a substantially faster and more reliably testing than with the knocking method used so far. The system consists of a carriage, which moves on a rail system along the rotor blade measuring it completely (Figure 15). An infrared heater and an infrared camera are mounted on the carriage in such a way that a given part of the blade is first heated and subsequently imaged by the camera (see Figure 8 for comparison). The driving speed ranges from 1 m/min to 10 m/min depending on the desired penetration depth.

A typical defect found with this setup is shown in Figure 16. The bright areas show air inclusions, which developed during the vacuum injection process used for soaking of the glass fibre mats.

For in-situ inspection of rotor blades by means of online thermography the use of a work lift is still necessary, however work on the development of a suitable robot system is under way at the WKI in co-operation with Idaswind Ingenieursgesellschaft mbH.

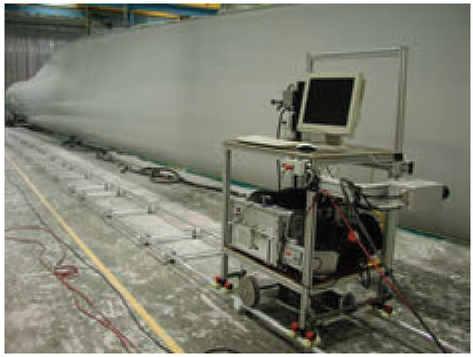


Figure 15: Test setup used by the WKI for testing of rotor blades at the production site.

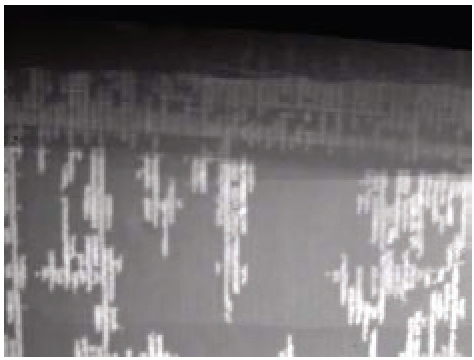


Figure 16: Rotor blade section with air inclusion due to problems in the vacuum injection process

SUMMARY

Thermography is a widely applicable testing method allowing the detection of a large number of defects in wood, wood-based materials and other composite materials such as GRP. One has to differentiate between passive and active procedures. Online thermography is particularly well suited for integration into production processes or also for the inspection of very large test pieces. By suitable image processing algorithms many test procedures can be automated.

# Testing pilots in a vacuum-chamber for emergency cases

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## ABSTRACT

The imagery of the thermal radiation of various objects in a defined range of the infra-red (IR) spectrum opens new vistas to researchers and practical specialists in the field of engineering and other sciences (medicine, biology, etc.). The paper presents an example on the possibilities of ergonomic-biologic examinations (the effect of changing ambient conditions) mainly from the aspect of thermal engineering.

*Keywords*: IR-image processing, human testing in vacuum-chamber, human adjustment to hypobaric-anoxia, thermal testing of pilots

## INTRODUCTION

The devices creating IR-images, thermographs and thermogrammeters which convert the variation on the surface of objects' thermal radiation to visible image, i.e. thermogram, work in this principle.
The examination of the human organisms as a thermal system leads to various conclusions in the field of biology and medicine. The comparison of the IR-images to the results of medico-diagnostic examinations permits further conclusions or makes it possible to apply them as screening test[1].

## TOPICS

*Ergonomic personality test by means of IR-images, pilot's test.* The different personality traits e.g. the way the individual reacts to various effect of the environment are well-known from psychology. However, this is not easy to determine such reactions without disturbing person by the use of instruments. In this cognitive process we can also avail ourselves of the information provided by IR-images.
In case of emergency or simply when emergency is possible to occur, or in an less dramatic situation, e.g. when a question is put to someone in an examination, the reaction of individuals manifests itself in the change in the thermal state of the head, namely that of the face, forehead and/or in the eye-nook, as well as on the hands. The character and degree of the change in temperature is in connection with the reaction brought about in the course of solving the problem, or only by the state of preparedness, and it differs according to the individual, but it is characteristic of the person. However, the same symptoms are effected by the change of ambient temperature and pressure [2].

## VACUUM-CHAMBER FOR TESTING PILOTS

In the case of an airliner's decompression the pilots have to tolerate some lack of oxygen (anoxia, hypobaric-anoxia) at 0.5 bar pressure for approximately 15 minutes without any oxygen respiration. The temperature of the eye-nook and the forehead are adequate indicators of the human thermophysical condition. The pilots' reactions were tested in a vacuum-chamber.

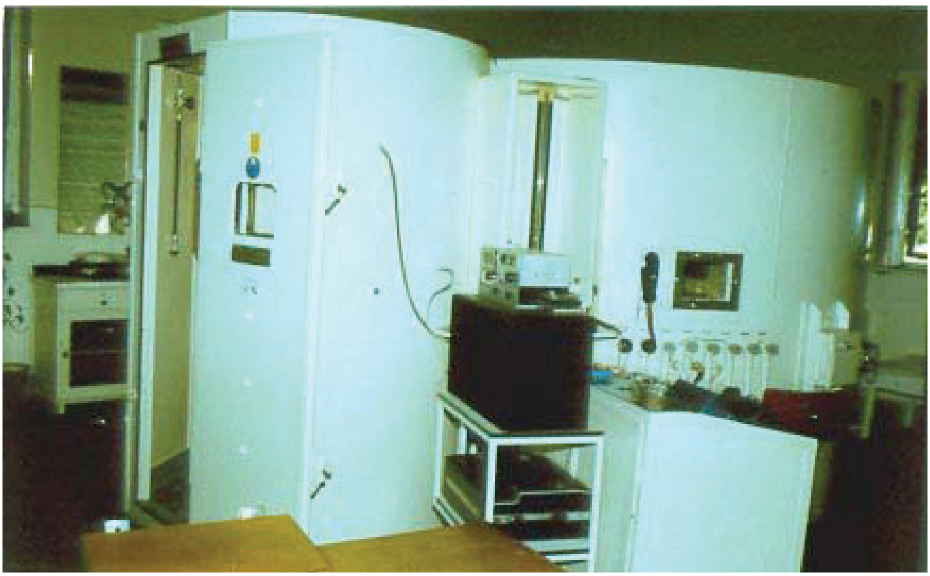

**Figure 1.** Outside view of an experimental vacuum-chamber equipped with medical measurement devices

## EXPERIMENTAL CONDITIONS

Process of testing of pilots and stewardess. Regular trainings are established during 15 min for stewardess on 4000 m and for pilots on 5000 m 3,[3].
A figure shows the pressure changes in the chamber vs. elapsed time from 'taking-off' to 'landing' without pressure equalizing (Fig. 4).

**Figure 2.** Lock-chamber of the main vacuum-chamber for sluicing/locking successive pressure equalising

**Figure 3.** Inside view in the vacuum-chamber with oxygen respiration equipment

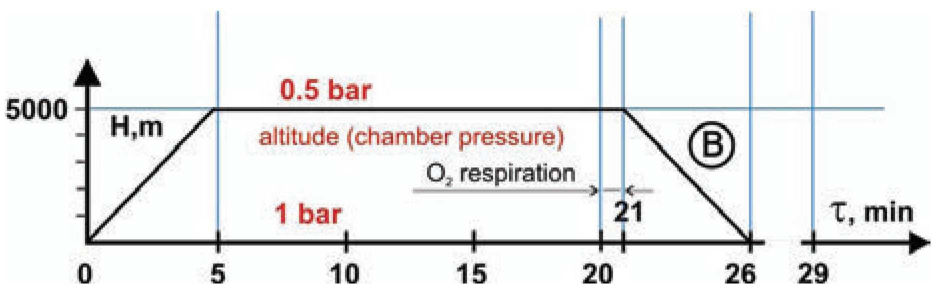


**Figure 4.** A model of the decompression in an airliner's emergency cases. Pressure-course in vacuum-chamber vs. process time without pressure equalizing

The model of decompression (Fig. 5) of an airliner's emergency case can be seen in function of altitude (chamber pressure) vs. time.

Pilot's thermal data (temperature of eye-nook and forehead) vs. process time serve as indicators of the pilot's adjustment to changing ambient parameters (Fig.6).

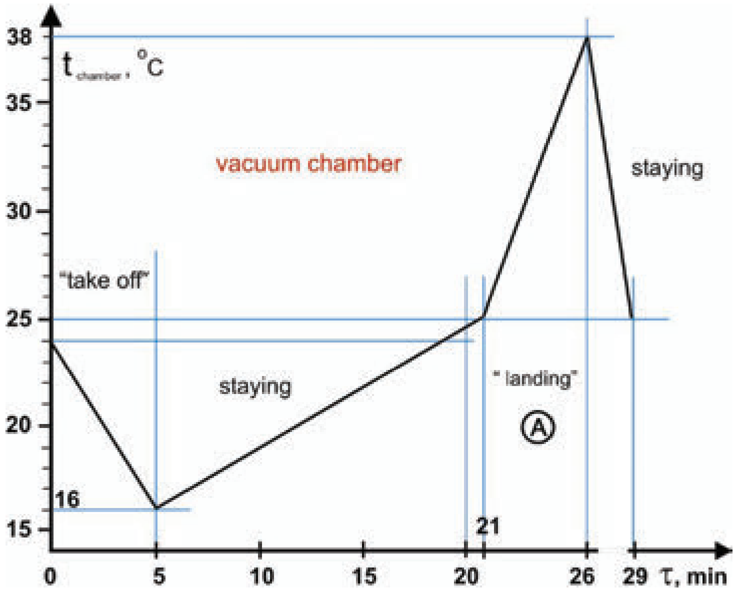


**Figure 5.** Temperature changes in a vacuum-chamber during the 'emergency' process (at the end of 'take-off' the fog/mist will be appear)

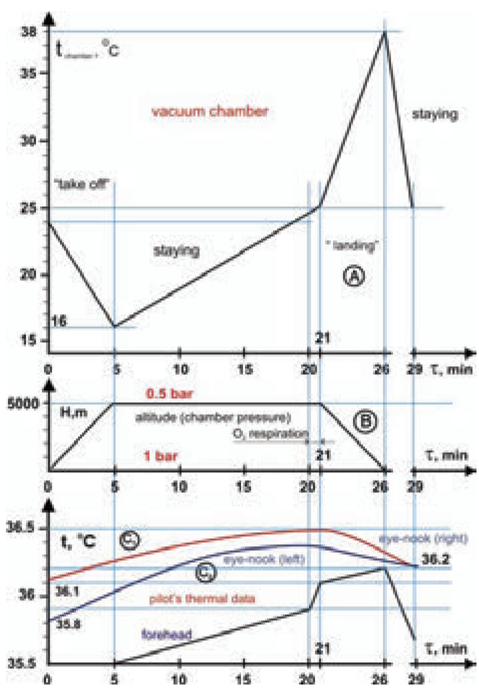


**Figure 6.** Pilot's thermal data vs. process time as indicators of the pilot's adjustment to changing ambient parameters

## HUMAN EXPERIMENT IN A VACUUM-CHAMBER THROUGH IR- IMAGING

The paper presents two series of IR-pictures taken of two pilots (beginner and experienced) with different capability for adjusting to decompression (Fig.7, $C_1$ – eye-nook left. $C_2$ –eye-nook right of a beginner pilot).

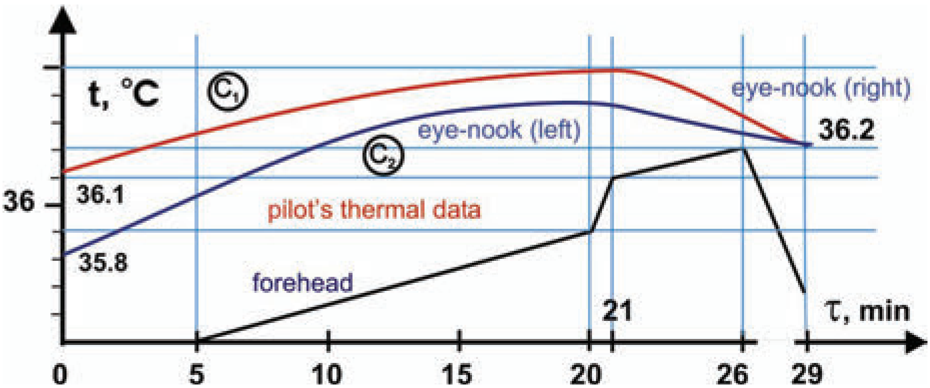


**Figure 7.** Pilot's eye-nook ($C_1$ and $C_2$) and forehead temperature vs. process time of a beginner pilot

## THERMAL METHOD OF ANALYSIS OF IR-IMAGES

### Thermal testing of a beginner pilot

Young, beginner pilot at zero minute (initial stage) before the load in vacuum-chamber on $24^0$C :

- IR-image of forehead (Fig. 8),
- IR-image of face and right- eye (Fig9),
- IR-image of shoulder and left-eye (Fig.10).

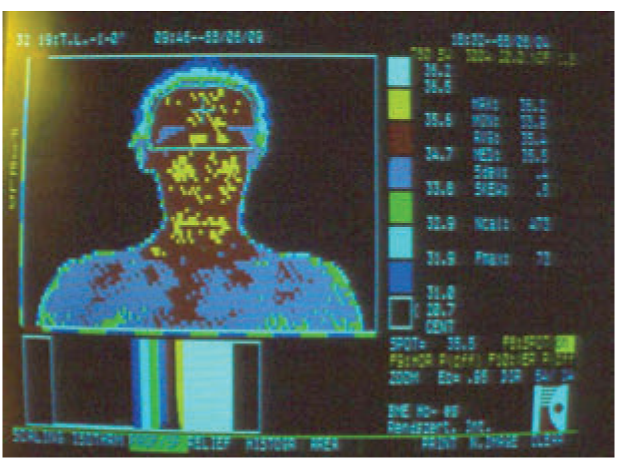

**Fig.8** A young pilot (initial stage)

0 min, 24 °C
forehead (f-h)

avg=35.4 °C (f-h)
spot=35.5 °C (f-h)

white: 36.6–36.5 °C
yellow: 36.5-35.6 °C
red: 35.6-34.7 °C

**Fig.9** A young pilot
(initial stage)

0 min, 24 °C face

avg=35.5 °C (face)

spot=35.5 °C (r.eye)

white: 36.6–36.5 °C

yellow: 36.5-35.6 °C

red: 35.6-34.7 °C

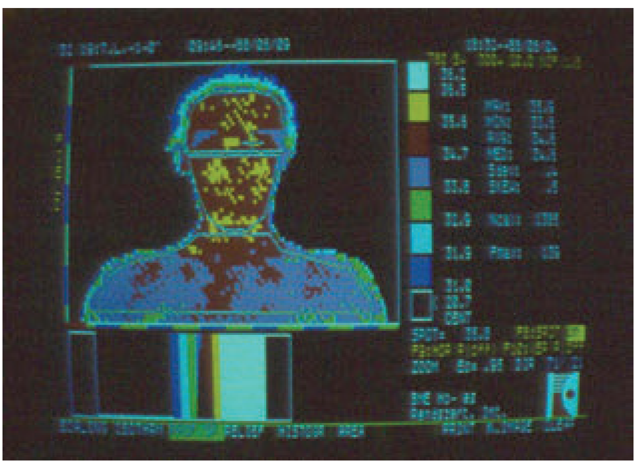

**Fig.10** A young pilot
(initial stage)

0 min, 24 °C
shoulder (s-r)

avg=34.6 °C (s-r)

spot=35.8 °C (l.eye)

white: 36.6–36.5 °C

yellow: 36.5-35.6 °C

red: 35.6-34.7 °C

Young, beginner pilot at the process:

- 'A' (take-off) on 16$^0$C, forehead, after 5 min of load (Fig.11),
- 'B' (staying) on 24$^0$C, on 0.5 bar at the end of 15. min (Fig.12).

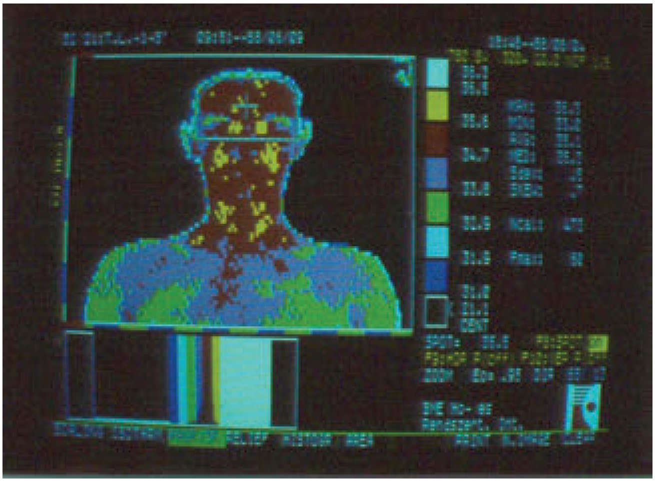

**Fig.11** A young pilot (process A)
5 min, 16 °C,
forehead

avg=35.1 °C (f-h)

spot=35.5 °C (f-h)

white: 36.6–36.5 °C

yellow: 36.5-35.6 °C

red: 35.6-34.7 °C

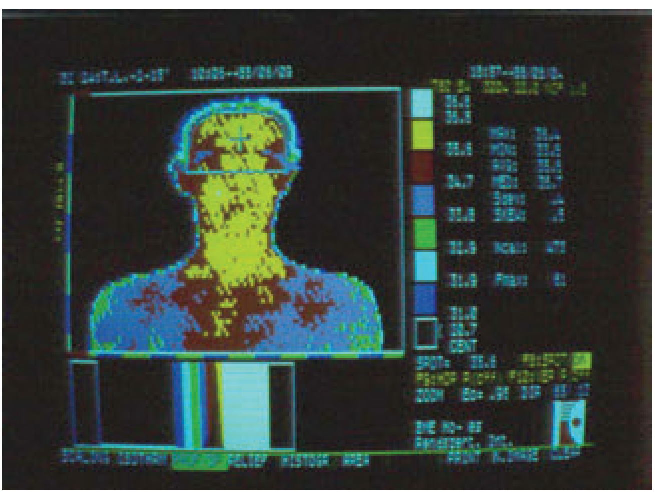

**Fig.12** A young pilot (process B)
15 min, 24 °C,
forehead

avg=38.6 °C (f-h)

spot=35.9 °C (f-h)

white: 36.6–36.5 °C

yellow: 36.5-35.6 °C

red: 35.6-34.7 °C

**Thermal testing of an experienced pilot**

Experienced pilot at zero minute (initial stage) before the load in vacuum-chamber on $24^0$C :

- IR-image of forehead (Fig. 13),

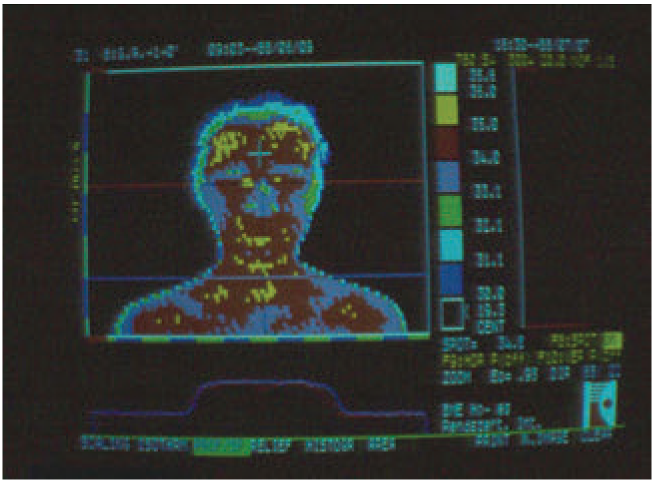

**Fig.13** Experienced pilot (initial stage)
0 min, 24 °C,
forehead

line-thermogram

spot=34.9 °C (f-h)

white: 36.4 – 36 °C

yellow: 36 - 35 °C

red: 35 - 34 °C

Experienced pilot at the process:

- ‘A’ (take-off) on $15.5^0$C, forehead, after 5 min of load (Fig.14),

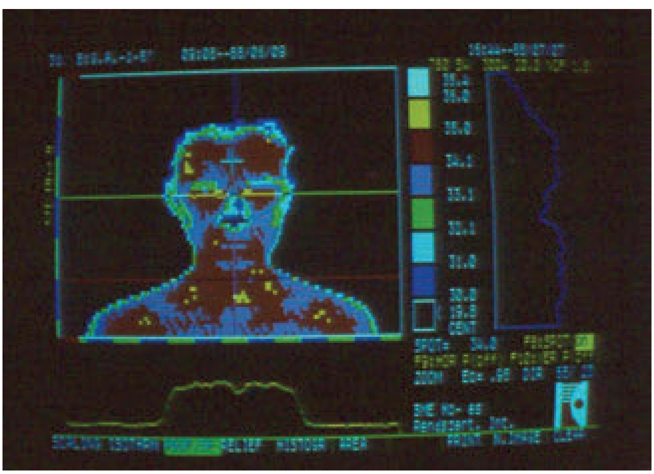

**Fig.14** Experienced pilot (process A)
5 min, 15.5 °C,
forehead

line-thermogram

spot=34 °C (f-h)

white: 36.4 – 36 °C

yellow: 36 - 35 °C

red: 35 - 34 °C

- ‘B’ (staying) on 24$^0$C, on 0.5 bar at the end of 15. min (Fig.15).

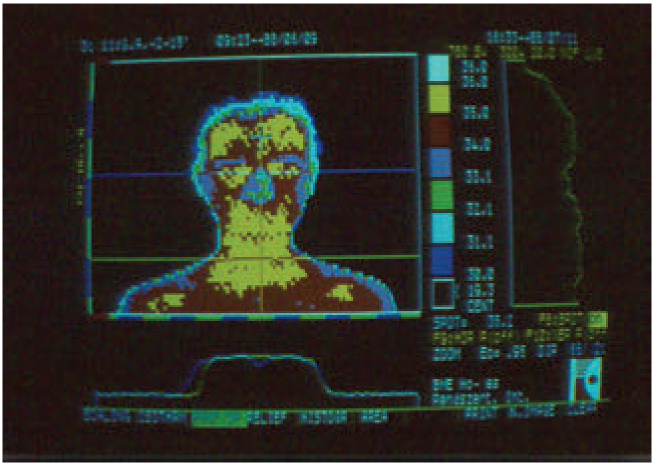

**Fig.15** Experienced pilot (process B)
15 min, 24 °C,
forehead

line-thermogram

spot=35.2 °C (f-h)

white: 36.4 – 36 °C

yellow: 36 - 35 °C

red: 35 - 34 °C

**SUMMARY OF ACCOMODATION OF DIFFERENT PILOTS**

Comparison of results of pilot’s thermal testing are presenting in Table 1A. and Table 1B. and Fig.7 .

Table 1. Accomodation of a beginner and an experienced pilot during the process in the vacuum-chamber

Table 1A. Thermal data of a beginner pilot

**Young pilot**

| | Fig. No. | $t_{amb}$, °C | time, min | histogram | avg, °C | spot, °C | Note | colours, °C |
|---|---|---|---|---|---|---|---|---|
| initial stage | 8. | 24 | 0 | forehead (f-h) | 35.4 | 35.5 | f-h | white: 36.6-36.5 |
| | 9. | 24 | 0 | face | 35.5 | 36.1 | r.eye | yellow: 36.5-35.6 |
| | 10. | 24 | 0 | shoulder | 34.6 | 35.8 | l.eye | red: 35.6-34.7 |
| process A | 11. | 16 | 5 | f-h | 35.1 | 35.5 | f-h | |
| process B. | 12. | 24 | 15 | f-h | 38.6 | 35.9 | f-h | |

Table 1B. Thermal data of an experienced pilot

**Experienced pilot**

| | Fig. No. | $t_{amb}$, °C | time, min | line-thermogram | avg, °C | spot, °C | Note | colours, °C |
|---|---|---|---|---|---|---|---|---|
| init. stage | 13. | 24 | 0 | forehead (f-h) | no | 34.9 | f-h | white: 36.4-36 |
| process A | 14. | 15.5 | 5 | f-h | no | 34 | f-h | yellow: 36-35 |
| process B. | 15. | 24 | 15 | f-h | no | 35.2 | f-h | red: 35-34 |

## CONCLUSION

The presented method and results allow to study the capability of pilots for adjusting to decompression.

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# Limitations of 1D Heat Flow Thermography on Compounds using Flash Sources

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## ABSTRACT

One of our current research projects deals with the inspection of compound materials for automotive lightweight construction. A hollow piece of light metal is filled with plastic to reduce weight and improve strength. The connection between these materials has to be measured. A classical method for measuring the bond strength of two layers is heat flow thermography. The surface of the specimen is heated with a flash pulse and the temperature sequence over time on the same side is observed. A bad connection between the layers transports less heat and causes a higher surface temperature after a certain time. The main problem for the mentioned project is that the metal layer has a much higher thermal conductivity than the plastic layer. Practical results that verify the principal feasibility are presented. We set up rules for the influence of material parameters on the sensitivity of the measurement.

## INTRODUCTION

Main objective of this work is to measure the quality of an interface between two materials. Only the top layers surface is accessible for exposure and temperature detection in the future application, so a one sided method is applied, even if the measured samples are made of simple plates.

The sample, to be inspected has a first layer of magnesia AZ91C, followed by a second layer with a material of lower thermal conductance. To get a good heat transmission, a heat conductive paste is used. Hence there are actually 3 layers, like shown in figure 1, called first, interface and second.

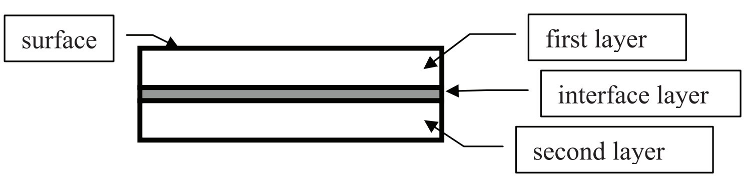


Figure 1. Composition of the samples

As second layer materials carbon steel, NiCr-steel, PVC and Teflon are used to get data from very poor conductors as well as for median conductors. For preventing measurement errors caused by lateral flux, the layers were made of small plates (25 x 25 x 1.5mm).

We describe the interface quality generally as an effective conductivity. Transmission between the three model layers is thought to be perfect. Interface quality in this context includes the conduction of heat through the interface material as well as the transmission from the first layer to the interface and from interface to the second layer.

The higher the conductance, the better is the interface quality. To be able to quantify the quality, a reference is neccassary. One reference point can be derived with a model, which has only the first layer. An interface with extremely low conductance would act in the same way. Practically it means that there is air behind the layer. We will use the temperature progress in this one-layer composition as lowest interface quality. To get another extreme point called optimal interface quality, the thickness of the first layer could be made larger, if the second layers material is the same as for the first layer. If we have different materials a simulation of the temperature progress with different thermal properties of the interface is used to give some referece values for possible interface qualities.

The procedure for evaluating the property of an interface is to look for the best fit of the simulated temperature progress to the measured one. Read out of the parameters, used for the best fitting simulation result then shows the thermal properties of the interface layer. To express the quality by only one single value, the heat conductivity, we keep the thickness of the interface layer at 0.1mm. Further we set the heat capacity and the density constant. To avoid differences that are caused by wrong thermal diffusivity values for the first and the second layer, we measured the diffusivity values.

**DIFFUSIVITY MEASUREMENT**

Thermal diffusivity is measured for all materials by a double-sided method, since it is believed to be the acuratest and robustest method. In this technique, that was derived by W. J. Parker (1961), the 50% transition $t_{50\%}$ of the temperature raise at the back side, caused by an short exposure at the front side is measured. With known thickness z the diffusivity $\kappa$ can be calculated by equation 1, in case of one dimensional heat flux.

$$\kappa = \frac{1.38 z^2}{\pi^2 t_{50\%}} \tag{1}$$

For better supression of the noise in the temperature signal a fit of the calculated temperature signal has been done. An equation for the temperature sequence after a dirac-pulse on the backside is given by Carslaw/Jaeger [1]. With the dimensionless time $\tau$ and the start time $\tau_0$ we get the temperature $T$:

$$\frac{T(\tau)}{T_{\max}} = 1 + 2\sum_{n=1}^{\infty} (-1)^n e^{-n^2(\tau-\tau_0)} \qquad \tau := \frac{\pi^2 \kappa}{z^2} t \tag{2}$$

To consider a finite flash duration $\tau_F$, we assume a rectangular flash function $F(\tau)$ which is 1 for $\tau < \tau_F$ and 0 otherwise. Integration gives

$$\frac{T_F(\tau)}{T_{F,\max}} = \frac{1}{\tau_F}\int_0^{\tau} F(\tau)\frac{T(\tau)}{T_{\max}}\,d\tau_0 =$$

$$= \begin{cases} \tau \ge \tau_F: & 1 + \dfrac{2}{\tau_F}\displaystyle\sum_{n=1}^{\infty}\frac{(-1)^n}{n^2}\left(e^{-n^2(\tau-\tau_F)} - e^{-n^2\tau}\right) \\ \tau \le \tau_F: & \dfrac{1}{\tau_F}\left(\tau - \dfrac{\pi^2}{6} - 2\displaystyle\sum_{n=1}^{\infty}\frac{(-1)^n}{n^2}e^{-n^2\tau}\right) \end{cases} \tag{3}$$

The values of $T_{F,\max}$ and $\kappa$ are varied in a fitting loop until an accurate coincidence has been reached. An example is shown in figure 2. The flash pulse is assumed to emit a constant amount of energy for the whole duration of 2.1ms. Since the pulse is cut off electrically, the exposure is close to fulfill this criterion.

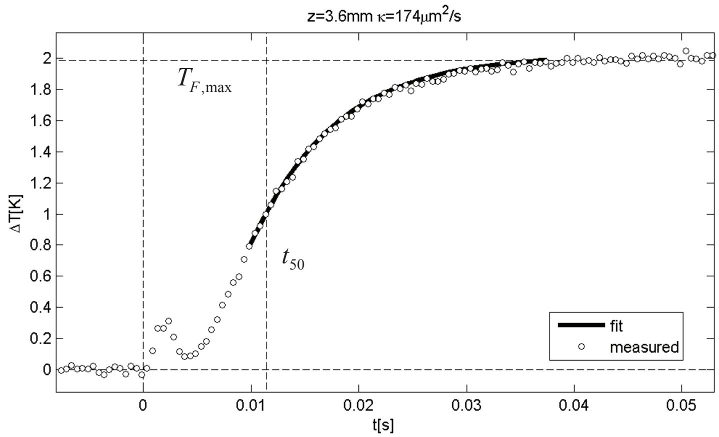


Figure 2. diffusivity measurement

The measured diffusivities are used in the simulations later in this work. By measuring the diffusivities, we want to avoid deviations of the temperature progress not caused by the properties of the interface.

| used in layer | conductivity W/m/K | density kg/m³ | capacity J/kg/K | diffusivity m²/s *1e-6 | effusivity $s^{0.5}Wm^{-2}K^{-1}$ | name |
|---|---|---|---|---|---|---|
| first | 50.2 | 1800 | 1030 | 27.2 | 9620 | Mg AZ91C |
| interface | 0.5 | 2000 | 1000 | 0.25 | 1000 | WLP0 |
| interface | 1 | 2000 | 1000 | 0.5 | 1410 | WLP01 |
| interface | 2 | 2000 | 1000 | 1 | 2000 | WLP02 |
| interface | 4 | 2000 | 1000 | 2 | 2830 | WLP03 |
| second | 66 | 7910 | 448 | 18.6 | 15300 | steel |
| second | 16.2 | 8000 | 500 | 4.05 | 8050 | steel (NiCr) |
| second | 0.2 | 945 | 1700 | 0.124 | 567 | PVC |

Table 1. thermal properties

Starting with the measured diffusivities, we searched for a material with diffusivity as close as possible to the measured one. So for instance we knew that we had magnesia AZ91 as first

layers material. Comparing the diffusivities led us to the more accurate result, that the material is AZ91C.
The different values in Table 1 for the interface materials WLP0..WLP03 are assumed. They are used in the simulation, to get references when measured data are compared.

**SIMULATION**

The temperature progress for different second layers materials is calculated by a finite differences model (FDM). It has been done in the one dimensional model with a spatial quantisation of 0.1mm. Time steps were set to 147µs to keep the calculation stable. Time is set to 0 for the 50%-time of the flash duration. The interface is modeled as a 0.1mm layer of WLP0 material.

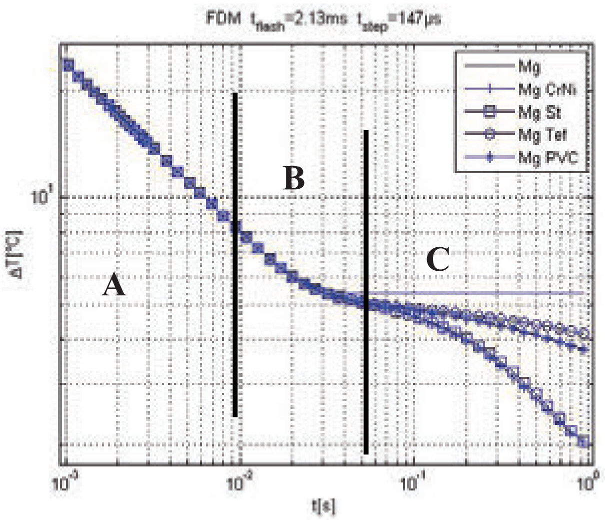


Figure 3. simulated temperature signal

As proposed by Krapez [2] the representation in logarithmic scale, as displayed in figure 3, is far more informative. Especially the derivatives with respect to the logarithm of time are very sensitive to subsurface features. Steven M. Shepard [3] divides the signal into distinct time regimes for better observation. For times, smaller than 10ms, there is almost no influence of the interface to the surface temperature progress. In the logarithmic domain the temperature has a gradient, close to -0.5, like it is expected for a semiinfinite solid. In figure 4, that shows the first derivative, it can be seen, that the temperature progress does not perfectly fit to this prediction.

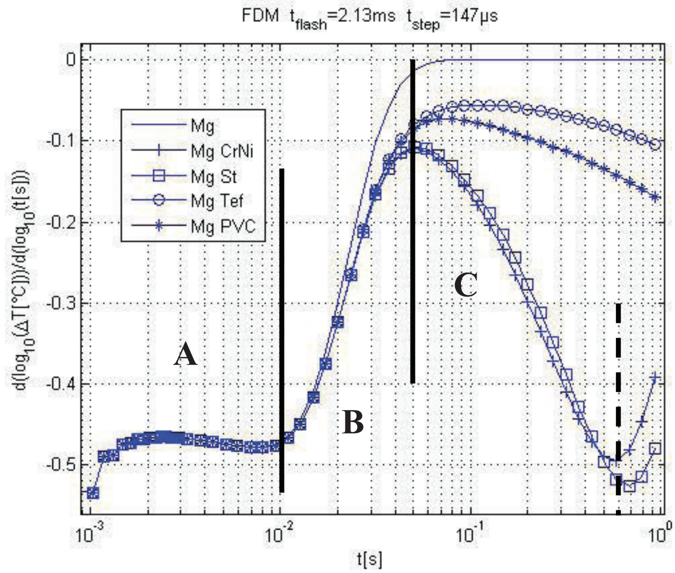


Figure 4. 1st derivative of simulated temperature signal with respect to the logarithm of time

The assumption, made for the semiinfinite solids temperature progress caused by an exposure of negligible duration is not given sufficiently enough. This becomes clearer if we look at the relation between flash duration $\tau_h$ =2.1ms and the time regime for zone A with t<10ms in figures 3 and 4. While the flash is heating up the surface, a significant flux of heat into the first layer causes a spatial temperature distribution, which differs from initial conditions. Since we want to find out the time regime, in which temperature is determined mainly by the interface, the influence of the flash duration plays a secondary role as long as the zones A and B are seperatable.

Zone B is mainly determined by the response of the interface between the first and the second layer. It could be seen as a kind of interconnection of temperatures behaviour caused by the layers itself. We define everything not caused by the first and the second layer to belong to the interface, which is modelled as a thin layer. As explained before, the interface quality will be expressed as an effective conductivity in a layer with a thickness of 0.1mm.

The response of heat conduction in the second layer is mainly determining the temperature progress in zone C, starting at aproximately 40ms. The end of zone C should be set to a time when an influence, caused by the backside of the second layer occures. Since we used different materials for the second layer, the time is different for “fast” and “slow” diffusivities. In figure 4 we highlight it for “Mg St” and “Mg NiCr” with a dashed line at aproximately 600ms.

## MEASUREMENT

All measurements were done with the Silver 450 thermocamera and a Graphit A4 flashdevice in the same composition. The flash was adjusted to 1600J in 2.1ms. Since it is switched off electrically, the energy pulse looks much like a rectangular pulse. To avoid reflections from the heated up lamp, an IR-cutoff filter was used. The camera captures the thermal images, exposed for 450µs in 500µs intervalls together with the signal from a function generator, which is triggered by a light-sensor. Extrapolation of a saw tooth signal from a function generator makes it possible, to calculate an accurate flash start time. The raising slope of the sawtooth is simply extended by a straight line. The intersection with the base-line then gives the accurate start time. Figure 5 shows a sketch of the measurement setup.

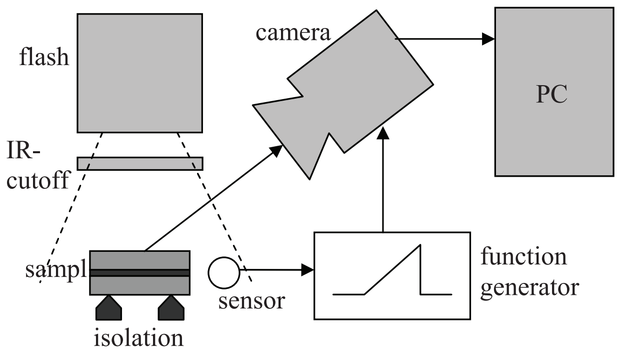


Figure 5. measurement setup

The captured thermal signals were evaluated by a PC, starting with calculation of the starting temperature of the samples surface. All temperatures in this paper are related to the starting temperature, which is believed to be close to the ambient temperature. A mean value from an area of 100 pixels was taken generally as well as a mean value of adjecent images was calculated in higher time regimes.

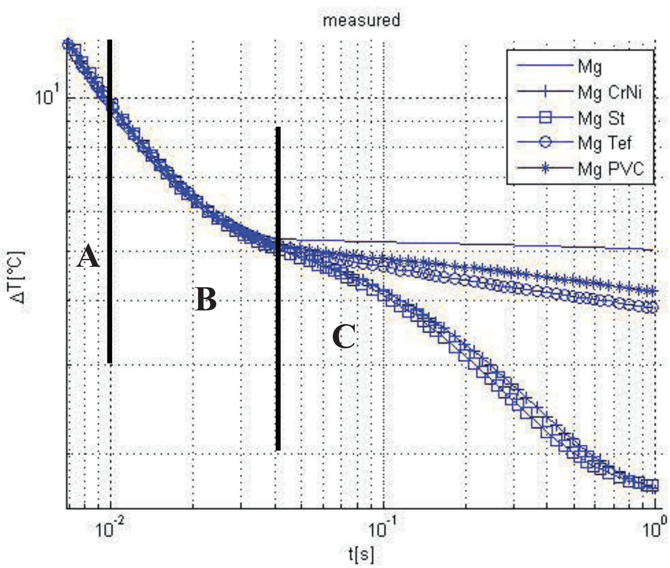


Figure 6. measured temperature signal

The measured curves in figure 6 look very similar to the simulated ones with three exceptions. In zone C the heat loss due to convection is visible in the "Mg" curve, which shows a cooling

down process, which of course influences all curves. The temperature difference does not change too much in this short time, so it is just an offset in figure 7 which shows the derivatives. Since it is small in relation to the signal amplitude, it should not influence our considerations for interface quality.
The gradiant in zone A, and partly still in zone B is another exception, because it is smaller than in the simulation. This is caused by the reflection of the cooling down flash lamp, which superposes the emission of the samples surface. In the comparisson between simulated and measured curves like displayed in figure 8, it can be seen, that this reflective influence tends to be very small at times, later than aproximately 25ms.

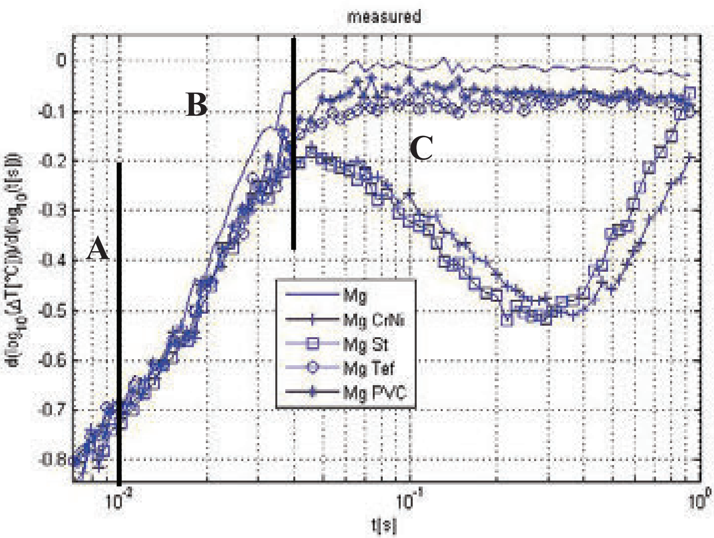


Figure 7. 1st derivative of measured temperature signal with respect to the logarithm of time

The third exception is located in zone C. Minimum of temperature derivative curve in figure 7 happens earlier than in simulation. Since the properties of the first and the second layer are the same in simulation and measurement, this effect must be caused by the properties of the interface.

**INTERFACE**

Variation of the interface properties in the simulation for the same geometry and the same first layer are displayed together with the measured data in figure 8.

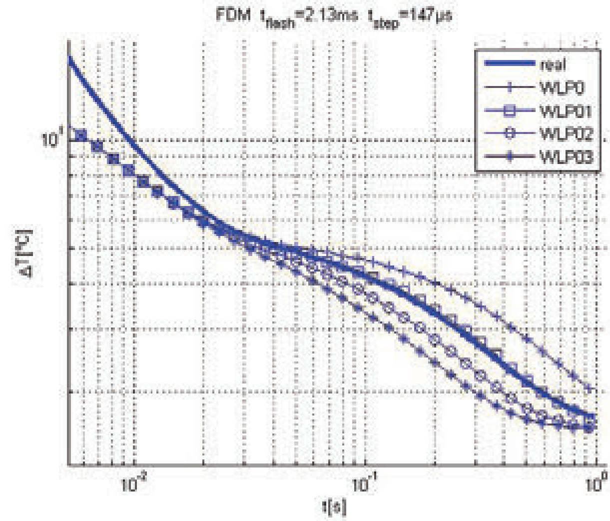


Figure 8. measured and simulated temperature signal for NiCr-steel

The measured temperature progress fits very well to the simulated one for WLP01 as interface. It is better observable in the first derivative, shown in figure 9.

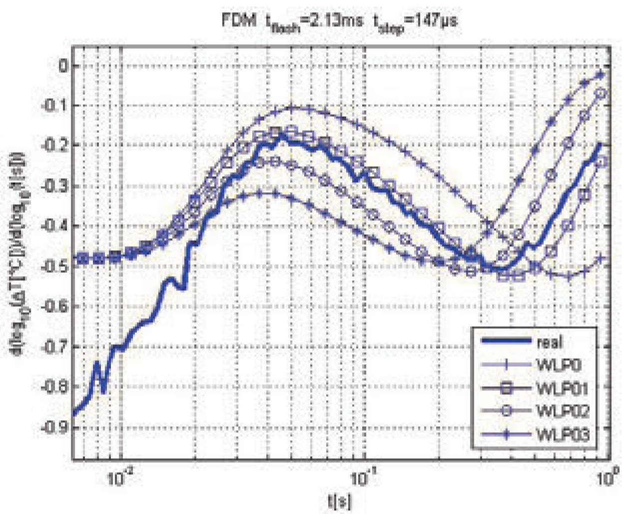


Figure 9. 1st derivative of measured and simulated signal for NiCr-steel

Not only the amplitude later than t=25ms, but also the time-shift of the minimum position fits quite well. Further the statements related to the differences between simulation and measurement, that were made in the precceeding chapture are shown to hold. The influence of reflection tends to become zero at aproximately 25ms, and the time when there is a minimum in the gradiant signal depends on the interface properties. If someone wants to find out, which of the simulated curves fits the best to the measured one, it is a preferrable situation. In case of NiCr-steel behind a magnesium layer, the interface quality can be measured with an excellent resolution. The effective conductivity in the measured sample is equivalent to WLP1, which means a $\lambda$ of 1W/m/K.

Figure 10 shows the temperature progresses for a second layers material of PVC. Again the reflection tends to zero at aproximately 25ms.

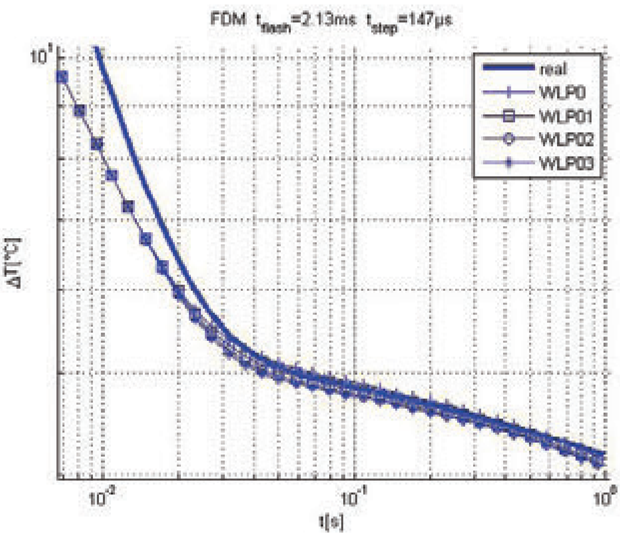


Figure 10. measured and simulated temperature signal for PVC

In opposition to the curves for NiCr-steel as second layer, there is no notable amplitude difference depending on the different interface properties. Also the gradients in figure 11 do not show a markable difference.

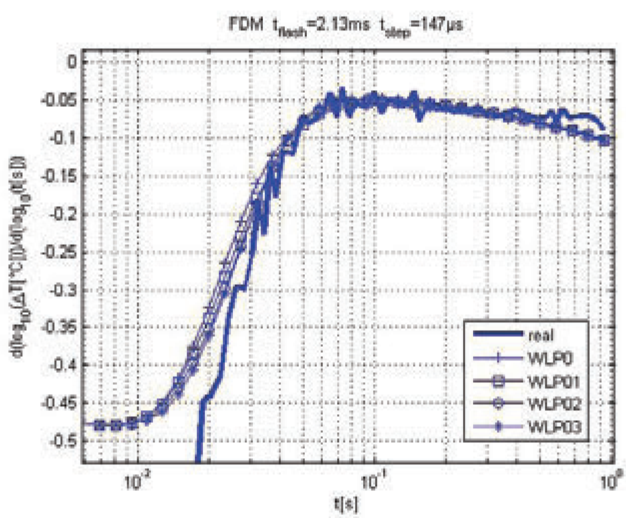


Figure 11. 1st derivative of measured and simulated signal for PVC

Since the signal amplitude of the surface temperature (4..5°C) is much higher than noise in the signal, it is still possible to evaluate the interface quality, but the resolution is very low now. In an extreme situation no seperation for quality can be done.

## CONCLUSIO

By comparing simulated temperature progresses for different interface qualities and their derivatives with measurements, absolute values for the interface property can be found when the properties of the first and the second layer are known. Especially when materials of the second layer with high or median conductivities are used, the resolution for the interface quality value is excellent. With low conductive materials the resolution becomes poor, but their existance behind the first layer is still detectable.

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# The application of infrared thermography in thermal analysis of open plate heat exchanger

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## ABSTRACT

The goal of the research was to find the relation for determining the heat transfer coefficient between a free streaming water film and metal corrugated surface. The relation between relevant parameters which determine the heat transfer coefficient has been obtained by means of Wilson-plot method using data from the measurement.

## INTRODUCTION

The open plate heat exchanger (OPHE) consists of two plates welded together so that one stream is flowing between plates and another one over the outer surface of upper plate. The aim of such heat exchanger design is to enable the heat exchange between outer stream which could be waste dirty water and inner stream. The OPHE are often used for heat recovery purposes in laundries, paper industries etc. The main problem in thermodynamical calculation of such type of heat exchanger is determination of the heat transfer coefficient on outer side.
Because the plates are always inclined to the horizontal plane and outer surface is corrugated to ensure better heat transfer the standard relations for flat plate could not be used. The paper presents the results of the research done on the one plate of OPHE with the aim to find the relation for heat transfer coefficient between streaming film and plate. The IR thermography was used for measurements of temperature distribution on water film streaming over the outer plate surface.

## THE PROBLEM

The open plate heat exchanger is aimed for heat exchange between hot waste water stream and cold fresh water stream. Waste water flows over the outer (upper) side of the OPHE plate in free stream and fresh water flows inside, between the plates. OPHE are used in the case when waste water contents particles or coagulants. The advantage of such design is easy cleaning of the plates and module design which enables enlargement of the unit. The upper side of OPHE plate is corrugated to improve the heat exchange between the free stream of waste water and the plate. The OPHE plates are inclined (5 º) to horizontal plane to ensure the free streaming by means of gravity. While the equations for calculating the heat transfer coefficient inside the plates could be found in literature, the relation for the calculation of heat transfer coefficient on the outer side became a problem. The goal of the research was to find the relation for the heat transfer coefficient on the outer side of the OPHE plate.

The single plate of OPHE used in experiment is shown in figure 1 and the detail of corrugated upper surface in figure 2.

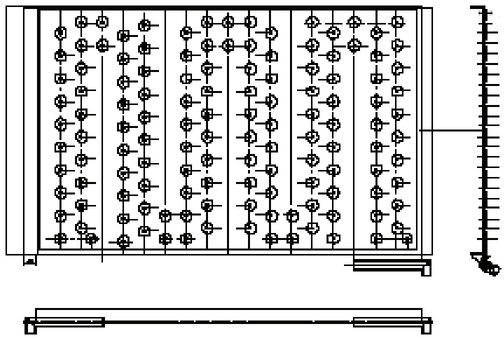


Figure 1. The plate of the heat exchanger

Figure 2. The features of upper plate

**EXPERIMENTAL RIG**

The experimental rig consists of single OPHE plate placed on a frame with upstream and downstream container (see figure 3). The hot waste water enters the upstream container and overflow on the plate. After passing the plate the water enters the downstream container. The fresh water streams in counter flow between the plates.

Figure 3. Experimental rig

The mass flow of waste (hot) and fresh (cold) water are measured by weighing while the temperatures are measured by thermocouples (see figure 4). Beside the thermocouples the surface temperature distribution of hot water streaming over the plate was recorded by IR camera. All measurements were done in stationary state.

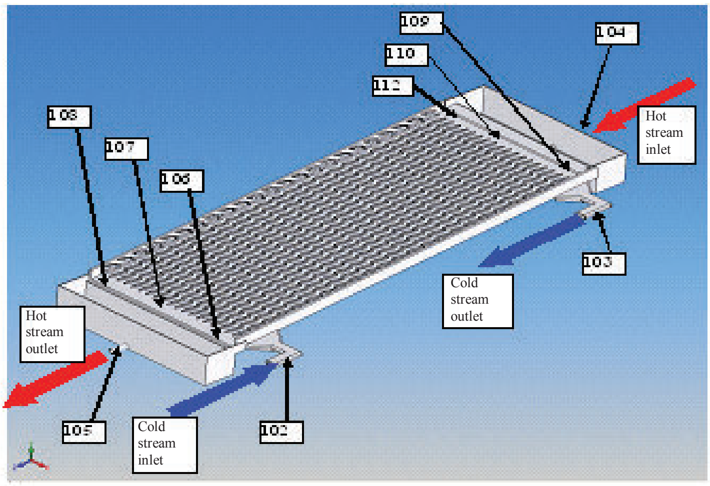



Figure 4. Positions of thermocouples and directions of hot and cold water stream

**MEASUREMENTS**

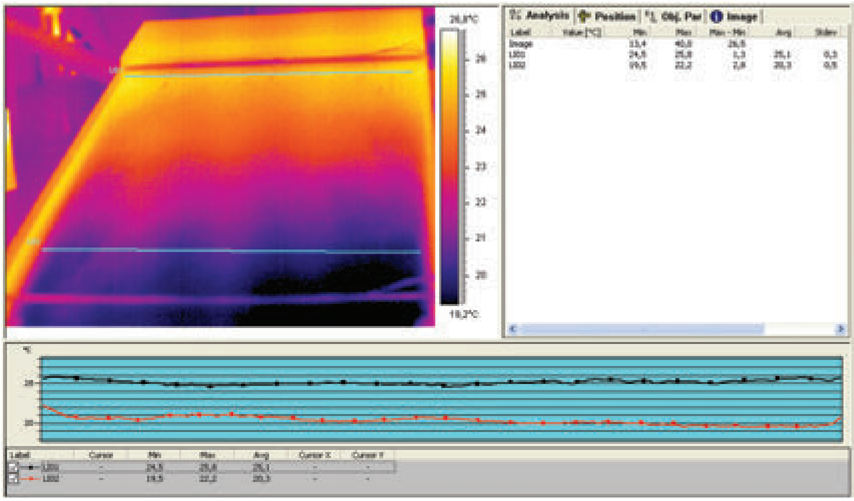
The measurements were carried out with mass flow of 0,1; 0,2; 0,3 and 0,4 kg/s for hot stream and 0,1 and 0,2 kg/s for cold stream. Some of the results are presented in table 1 and the related thermograms are shown in figure 5.

*The results of measurement using thermocouples*

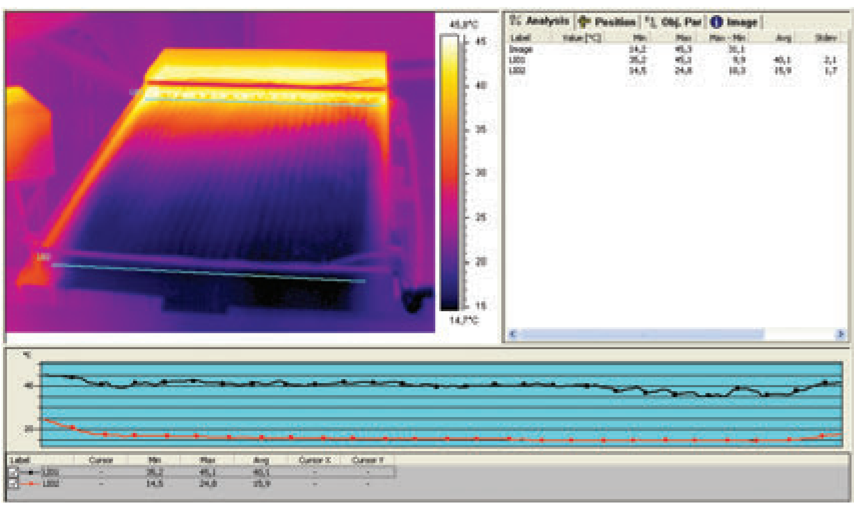
Table 1. The results of measurement using thermocouples

| Measurement No. | | Hot stream | Cold stream |
|---|---|---|---|
| 1 | Mass flow [kg/min] | 12,58 | 6,31 |
| | Inlet temperature [°C] | 25,67 | 12,57 |
| | Outlet temperature [°C] | 19,59 | 21,93 |
| 2 | Mass flow [kg/min] | 5,03 | 12,48 |
| | Inlet temperature [°C] | 43,54 | 13,47 |
| | Outlet temperature [°C] | 15,31 | 23,65 |
| 3 | Mass flow [kg/min] | 12 | 12,14 |
| | Inlet temperature [°C] | 27,29 | 13,72 |
| | Outlet temperature [°C] | 18,23 | 21,21 |
| 4 | Mass flow [kg/min] | 24,5 | 12,25 |
| | Inlet temperature [°C] | 21,90 | 15,11 |
| | Outlet temperature [°C] | 19,07 | 19,61 |

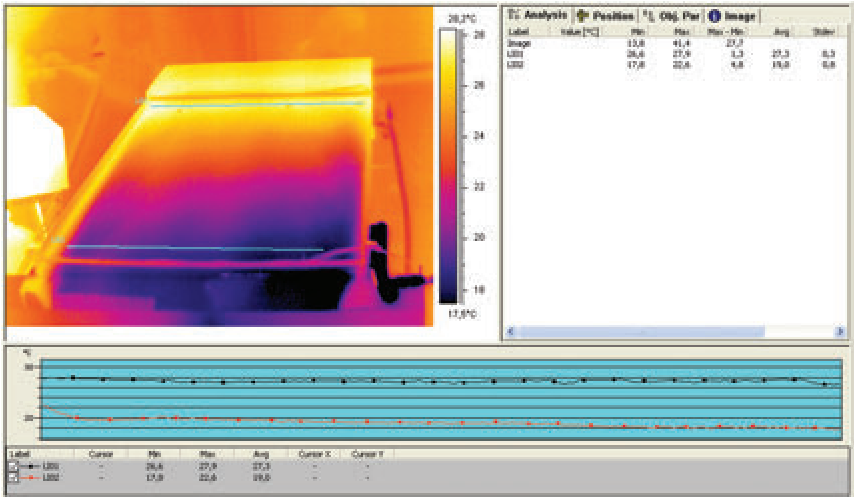
*The results of measurement using infrared camera*



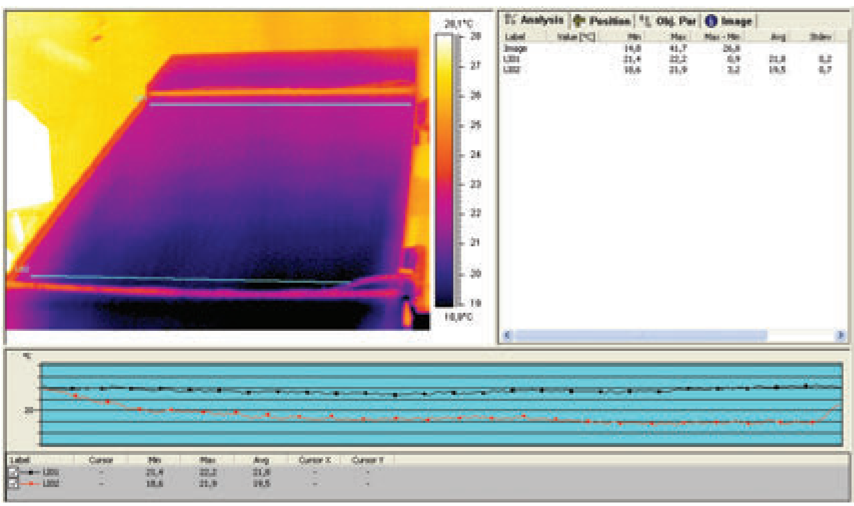
a) Measurement No.1



b) Measurement No. 2



c) Measurement No.3



d) Measurement No.4

Figure 5. Thermograms and referent temperatures

Table 2. The results of measurement; the thermocouples were used for cold stream and infrared camera for hot stream

| Measurement No. | | Hot stream | Cold stream |
|---|---|---|---|
| 1 | Mass flow [kg/min] | 12,58 | 6,31 |
| | Inlet temperature [°C] | 25,1 | 12,57 |
| | Outlet temperature [°C] | 20,3 | 21,93 |
| 2 | Mass flow [kg/min] | 5,03 | 12,48 |
| | Inlet temperature [°C] | 40,1 | 13,47 |
| | Outlet temperature [°C] | 15,9 | 23,65 |
| 3 | Mass flow [kg/min] | 12 | 12,14 |
| | Inlet temperature [°C] | 27,3 | 13,72 |
| | Outlet temperature [°C] | 19,0 | 21,21 |
| 4 | Mass flow [kg/min] | 24,5 | 12,25 |
| | Inlet temperature [°C] | 21,8 | 15,11 |
| | Outlet temperature [°C] | 19,5 | 19,61 |

The heat exchanged between the streams is calculated using relations:
- for hot stream

$$\Phi_h = q_h \cdot c_h \cdot (\vartheta_{ih} - \vartheta_{oh}) \text{ kW} \quad (1)$$

- for cold stream

$$\Phi_h = q_c \cdot c_c \cdot (\vartheta_{oc} - \vartheta_{ic}) \text{ kW} \quad (2)$$

Table 3. The results of calculation of the heat fluxes

| Measurement No. | $\Phi_c$ kW | $\Phi_h$ thermocouples kW | $\Phi_{h\ IR}$ kW |
|---|---|---|---|
| | cold stream | hot stream | |
| 1 | 4,119 | 5,327 | 4,21 |
| 2 | 8,861 | 9,883 | 8,49 |
| 3 | 6,343 | 7,580 | 6,68 |
| 4 | 3,845 | 4,843 | 3,92 |

It can be seen that, using recordings by infrared camera, better accuracy was obtained for hot water stream in comparison with those obtained by thermocouples.

**RESULTS**

By means of overall heat transfer coefficients, calculated from measurement, and heat transfer coefficient for cold stream using Eq. (3), the heat transfer coefficient for hot stream has been determined using Eq. (4).

$$\alpha_c = 0{,}022 \cdot \mathrm{Re}^{0,8} \cdot \mathrm{Pr}^{0,4} \cdot \frac{\lambda}{d_{ekv}} \tag{3}$$

$$\frac{1}{\alpha_h} = \frac{1}{k} - \frac{\delta_{wall}}{\lambda_{wall}} - \frac{1}{\alpha_c} \tag{4}$$

Using Wilson-plot method the relation Eq. (5) for determination of heat transfer coefficient of outer stream can be found using data from measurements:

$$\alpha_h = 0{,}524 \cdot \mathrm{Re}^{0,74} \cdot \mathrm{Pr}^{1/3} \cdot \frac{\lambda}{L} \tag{5}$$

Calculating the heat transfer coefficient using Eq. (5) for hot stream and comparing them with those obtained from measurement it can be seen that good mutual correspondence is obtained (see table 4).

Table 4. The comparison of heat transfer coefficients for hot stream, measurement and Eq.(5)

| Measurement No. | $\alpha_{from\ measurement}$ W/m$^2$K | $\alpha_{calculated\ Eq.(5)}$ W/m$^2$K |
|---|---|---|
| 1 | 2892,7 | 2866,3 |
| 2 | 2337,4 | 2286,5 |
| 3 | 2914,5 | 2832,0 |
| 4 | 3232,8 | 3376,5 |

## CONCLUSION

In the paper the application of IR thermography in determination of heat transfer intensity has been described. The specific profile of heat exchanger's surface required an investigation for new relations for calculation of heat transfer coefficients. The results obtained using Eq.(5) for calculation heat transfer coefficients on hot water side indicated a good correspodence with the measurement's result.

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[3] Gerald C. Holst, Common sense approach to thermal imaging, *SPIE Optical Engeenering Press*, Bellingham, Washington, USA, 2000

# Application of thermography for determination of thermal conductivity of prefabricated insulating products

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## ABSTRACT

The paper presents the possibilities of using thermography in measurement of thermal conductivity of prefabricated insulating sandwich panels and pipe/in/pipe assemblies according to EN and ISO standards. From data presented it can be seen that thermography could be successfully used in measurements of outer surface temperatures of the samples as well as thermocouples or other standard sensors. Thermography gives not only values of temperature on certain point, but also average surface temperature or average line temperature and some additional information of sample structure. Using thermography as additional tool in such measurements more useful information can be obtained in comparison with classical methods.

## INTRODUCTION

Determination of steady state thermal transmission properties for thermal insulation and prefabricated insulating products, such as sandwich insulating panels or pipe/in/pipe assemblies, is of great importance. In conformity assessment procedures of such construction products the value of its thermal conductivity must be given to enable the designer to perform correct and efficient calculations and design. The following technical specifications lay down a test method and apparatus for determination of steady state thermal resistance and related properties:

a. HRN ISO 8302:1998. Thermal insulation-Determination of steady state thermal resistance and related properties-Guarded hot plate apparatus.
b. HRN EN ISO 8497:1988. Thermal insulation-Determination of steady state thermal transmission properties of thermal insulation for circular pipes.
c. EN 253:2003, District heating pipes-Preinsulated bonded pipe systems for directly buried hot water networks-Pipe assembly of steel service pipe, polyurethane thermal insulation and outer casing of polypropylene.

The apparatus proposed in technical specifications have been built in the Laboratory for Applied Thermodynamics, Faculty for Mechanical Engineering and Naval Architecture Zagreb and measurements of relevant parameters needed for the calculation thermal conductivity performed.

**INSULATING PANELS (HRN ISO 8302:1998.)**

**Apparatus principle**

The guarded hot plate apparatus is intended to establish within specimens in the form of uniform slabs having flat parallel faces, a unidirectional uniform density of heat flow rate at steady state conditions.

**Apparatus types**

From the basic principle two types of guarded hot plate apparatus were derived:

a. with two specimens (and central heating unit)
b. with single specimen

The type with two specimens has been designed and built in the Laboratory, (see figure 1.).
In two specimen apparatus a central flat plate consist of a heater and metal surface plates and this heating unit is sandwiched between two nearly identical specimens. The heat flow rate is transferred through the specimen to separate isothermal flat assemblies called the cooling units or to outer samples surfaces.

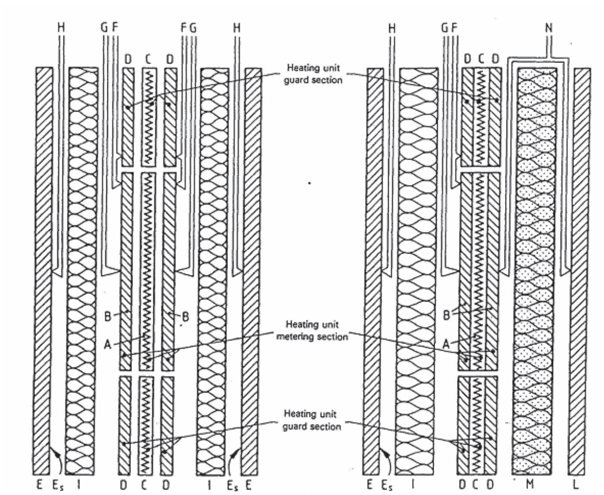


Figure 1. Schematic of test configuration for prefabricated insulating panels (see HRN ISO 8302)

**Heating and cooling units**

The heating unit consists of separate metering section, where the unidirectional uniform and constant density heat flow rate can be established, surrounded by guard sections separated by a narrow gap. The cooling units may consist of continuous flat plate assembly.

**Edge insulation and auxiliary guarded sections**

Additional edge insulation and/or auxiliary guard sections are required, especially when operating above or below room temperature.

**Experimental rig – two specimen apparatus**

The guarded hot plate apparatus shown in figure 2 and 3 is designed and constructed in the Laboratory for Applied Thermodynamics according to specification in HRN EN 8302:1998. It consists of cooper heating unit and heating guard sections.
Test specimens are placed on both sides of the heater and thermocouples placed on the required positions. The flat plates of main heater and guard heaters are made of 7 mm cooper sheet of and electrical wire form the heater between them. For temperature measurement 8 thermocouples type T were used and temperatures indicating the heat equilibrium between main and guard plates was measured with 16 thermopiles. The data were collected by an A/D converter type AGILEND and processed later.

The outer samples surface temperatures were measured by thermocouples and IR camera type ThermaCAM 2000 SC. The samples were heated to stationary state before the measurements start. The heating lasts about 35 hours

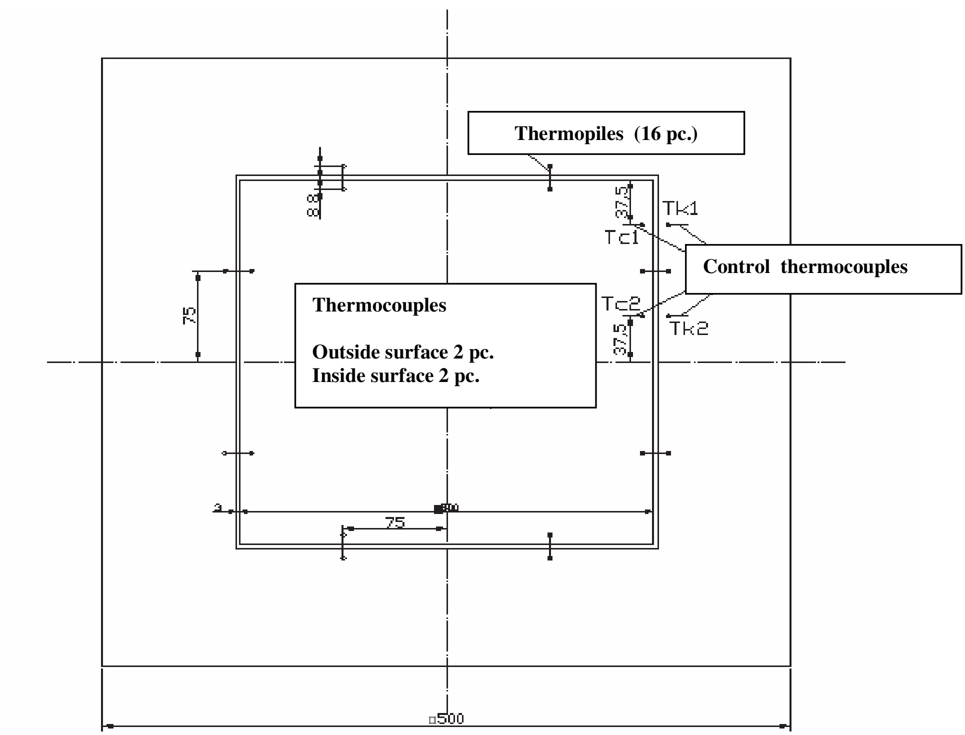


Figure 2. Position of thermocouples and thermopiles in testing apparatus

Figure 3. The test rig for prefabricated panel

**Samples and thermographic measurements**

The samples were flat polyurethane panels having dimensions of 300 x 300 mm and 60 and 80 mm thickness. The sample density was 46,1 kg/m$^3$ for 50 mm thickness and 52,3 kg/m$^3$ for 80 mm thickness.
Thermographic measurements were performed in stationary state in a conditioned room. All together 25 thermograms were recorded for each sample with following parameters:

*Sample I (thickness 60 mm)*
- emissivity: 0,95
- ambient temperature: 25,5 °C
- relative humidity: 50 %
- recording distance: 1,4 m

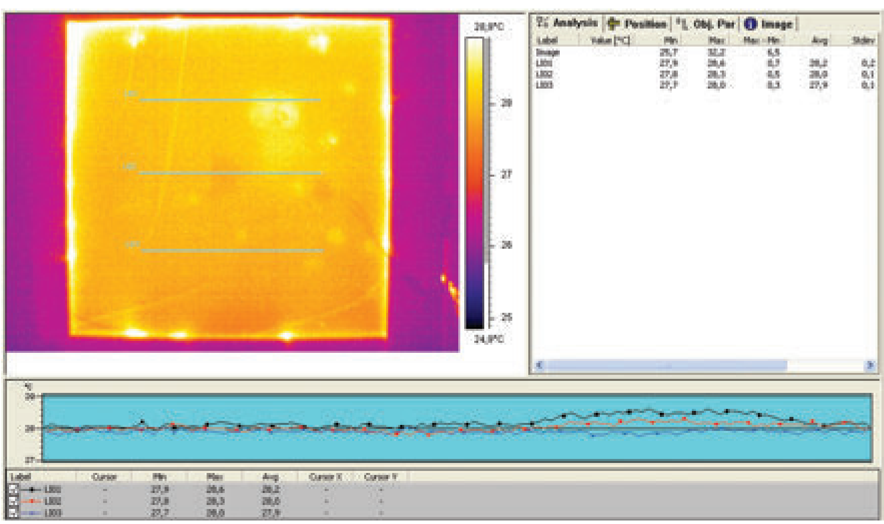


Figure 4. Thermogram of sample surface, side A
(Average surface temperature 28,03 °C)

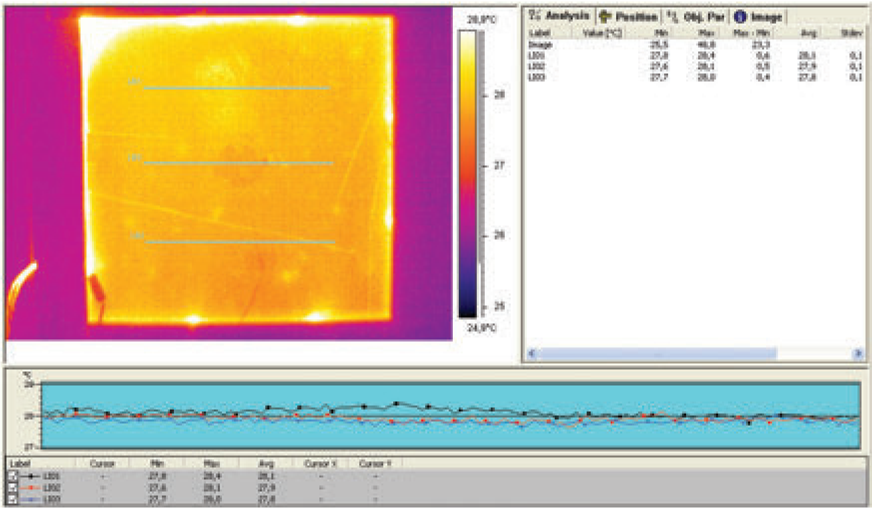

Figure 5. Thermogram of sample surface, side B
(Average temperature 27,95 ºC)

*Sample II (thickness 80 mm)*

- emissivity: 0,95
- ambient temperature: 27,4 ºC
- relative humidity: 50 %
- recording distance: 1,4 m

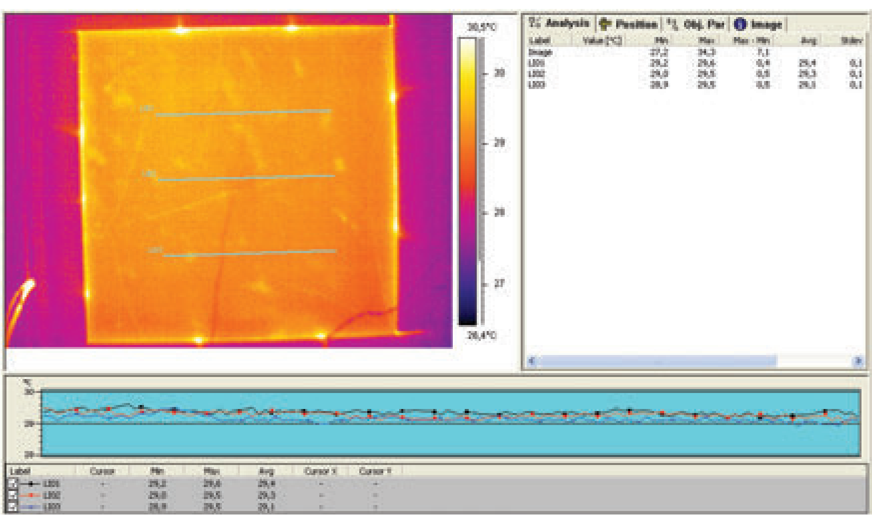

Figure 6. Thermogram of sample surface, side A
(Average surface temperature 29,16 ºC )

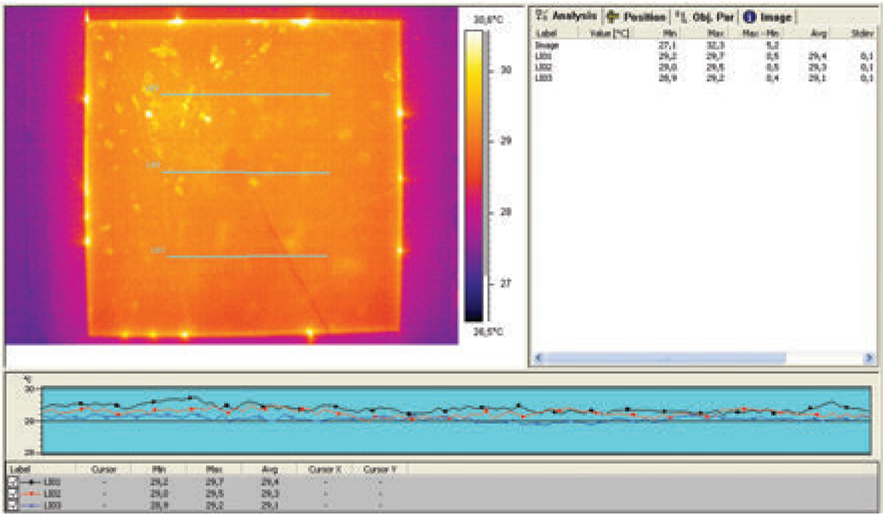

Figure 7. Thermogram of sample surface, side B
(Average temperature 29,26 ºC )

Thermograms show good uniformity of temperature distribution on samples outer surfaces. Some debiation from average temperatures can be seen in the areas between guards and outer insulation.

**Thermal conductivity of measured samples**

Thermal conductivity of the sample is calculated as:

$$\lambda = \frac{\Phi \cdot d}{A \cdot (T_1 - T_2)} \quad \mathrm{W/(mK)} \qquad (1)$$

Were:
$\Phi$ – average power of central heating unit, W
$T_1$ – average temperature of inner sample surface, K
$T_2$ – average temperature of outer sample surface, K
$A$ – surface area of central section according to standard, m$^2$
$d$ – average sample thickness, m

*Sample I*
$T_v$ – temperatures measured by thermocouples, outer sample side
$T_u$ – temperatures measured by thermocouples, inside sample surface
$T_c$ – control thermocouples, heating unit
$T_k$ – control thermocouples, guard heating units

Table 1. Results obtained for thickness 60 mm

| | | |
|---|---|---|
| $T_{u1}$ | °C | 64,0 |
| $T_{u2}$ | °C | 64,0 |
| $T_{v1}$ | °C | 27,6 |
| $T_{v2}$ | °C | 27,7 |
| $T_{ok}$ | °C | 25,6 |
| $T_{k1}$ | °C | 62,7 |
| $T_{c1}$ | °C | 63,8 |
| $T_{k2}$ | °C | 62,9 |
| $T_{c2}$ | °C | 64,0 |
| $U_k$ | V | 10,51 |
| $U_c$ | V | 8,614 |
| $P_c$ | W | 3,484 |
| $\lambda$ | W/mK | **0,030484** |

*Sample II*

Table 2. Results obtained for thickness 80 mm

| | | |
|---|---|---|
| $T_{u1}$ | °C | 70,8 |
| $T_{u2}$ | °C | 70,9 |
| $T_{v1}$ | °C | 28,9 |
| $T_{v2}$ | °C | 29,0 |
| $T_{ok}$ | °C | 27,4 |
| $T_{k1}$ | °C | 69,0 |
| $T_{c1}$ | °C | 70,6 |
| $T_{k2}$ | °C | 69,3 |
| $T_{c2}$ | °C | 70,8 |
| $U_k$ | V | 10,463 |
| $U_c$ | V | 8,571 |
| $P_c$ | W | 3,276 |
| $\lambda$ | W/mK | **0,034432** |

The differences between thermal conductivities obtained from thermographic measurements and by means of thermocouples were within 2 % which satisfy the requirements of applied standards.

*Results:*

Table 3. Measurements results

| Plate thickness | Density | Average thermal conductivity |
|---|---|---|
| 60 mm | $\rho = 46{,}1\ kg/m^3$ | $\lambda = 0{,}030484\ W/(mK)$ |
| 80 mm | $\rho = 52{,}3\ kg/m^3$ | $\lambda = 0{,}034432\ W/(mK)$ |

**DETERMINATION OF THE THERMAL CONDUCTIVITY OF PREINSULATED PIPES (HRN EN ISO 8947:1994.)**

**Testing apparatus**

Thermal conductivity can be measured by means of two types of apparatus defined in Standard EN ISO 8947:1994.
Guarded end apparatus uses separately heated pipe sections called "guards" located at each end of the metered test section having the length of $L \geq 1$ m. The guards shall be maintained at the test section temperature to eliminate axial heat flow in the apparatus, and to aid in achieving uniform temperatures so that all heat flow in the specimen test section will be in radial direction.
Calibrated end apparatus for which the heat loss through the end caps must be determined. This shall be obtained by making measurements on specimens with different length taken from the same pipe sample. Length of the test section shall be not less than $L \geq 2$ m.

**Testing conditions**

The test specimen shall be conditioned at room temperature for one week. Dimensions given in figure 8 must be measured with the following accuracy:
Casing diameter ± 0,5 mm
Service pipe diameter ± 0,2 mm
Thickness of the casing ± 0,2 mm

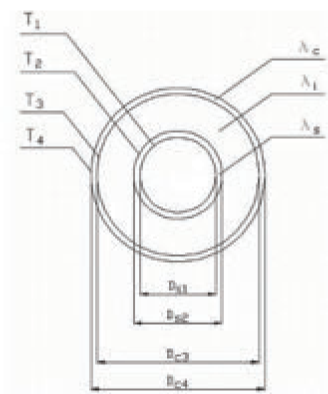

Figure 8. Symbols

Sensors for measuring the temperature of the specimen shall be attached on the service pipe inner surface and casing outer surface. The test section length shall be divided into at least four equal parts and at least one temperature sensor at the steel pipe and at the casing shall be longitudinally located at the centre of each part. The sensors shall be circumferentially equally spaced.
Ambient temperature of still air in the testing room must be 23 ± 2 °C may vary during the test by more than ± 1°C.
Test pipe temperature interval 70 °C to 90 °C measured at the inside surface of the service pipe. Tests are to be run at a minimum of three service pipe temperatures with accuracy ± 0,3 °C.

**Thermal conductivity**

Thermal conductivity at the mean temperature in the insulation material shall be calculated by linear regression using results obtained at different pipe temperatures. The result is reported as $\lambda$ at the mean temperature $T_m$.

| | | |
|---|---|---|
| Heat flow rate | $\Phi$ | W |
| Test section length | $L$ | m |
| Temperature of service pipe inner surface | $T_1$ | °C |
| Temperature of insulation inner surface | $T_2$ | °C |
| Temperature of insulation outer surface | $T_3$ | °C |
| Temperature of casing outer surface | $T_4$ | °C |
| Mean temperature of insulation | $T_m$ | °C |
| Service pipe inner diameter | $D_{s1}$ | m |
| Inner diameter of insulation material | $D_{s2}$ | m |
| Outer diameter of insulation material | $D_{c3}$ | m |
| Outer diameter of casing | $D_{c4}$ | m |
| Thickness of casing | $t$ | m |
| Thermal conductivity of insulation material | $\lambda_i$ | W/(mK) |
| Thermal conductivity of casing | $\lambda_c$ | W/(mK) |
| Thermal conductivity of service pipe | $\lambda_s$ | W/(mK) |

$$\lambda_i = \frac{\ln\left(\frac{D_{c3}}{D_{s2}}\right)}{\frac{2\cdot\pi\cdot(T_1-T_4)\cdot L}{\Phi} - \frac{1}{\lambda_c}\cdot\ln\left(\frac{D_{c4}}{D_{c3}}\right) - \frac{1}{\lambda_s}\cdot\ln\left(\frac{D_{s2}}{D_{s1}}\right)} \qquad (2)$$

$$T_3 = T_4 + \frac{\Phi}{2\cdot\pi\cdot L\cdot\lambda_c}\cdot\ln\left(\frac{D_{c4}}{D_{c3}}\right) \qquad (3)$$

$$T_m = \frac{(T_3 - T_2)}{2} \qquad (4)$$

$$T_2 = T_1 - \frac{\Phi}{2\cdot\pi\cdot L\cdot\lambda_s}\cdot\ln\left(\frac{D_{s2}}{D_{s1}}\right) \qquad (5)$$

**Application of thermography for surface temperature measuring**

Thermography can be successfully used for measuring the surface casing temperature instead of using standard sensors. It gives not only the possibility of recording the temperature of certain point but also on a chosen line or surface. Thermogram gives more accurate temperature recording than point by point measurement. Simultaneously, some non uniformities in the pipe structure can be seen, like position and influence of distancers.
In figure 9 and 11 the thermogram recorded during the measurement of pipe insulation characteristic is shown. Beside the temperature distribution on pipe surface the position of distancers can be clearly seen on thermogram because of different coefficients of thermal conductivity of distant rings and insulation.

*Testing pipe I*

Inside temperature of service pipe: 94,7 °C (stationary state)
Average temperature on outer casing surface: 22,5 °C (measured by thermography)
Ambient temperature: 21 °C
Input power: 57,81 W
Casing diameters: $D_{C4}$ = 308 mm; $D_{C3}$ = 302 mm
Service pipe diameter $D_{S2}$ = 220 mm; $D_{S1}$ = 213 mm
Thermal conductivity: $\lambda_c$ = 0,4 W(/mK), $\lambda_s$ = 50 W/(mK)
Length: $L$ = 1,61 m

Using equations (2) to (5) we obtain:
$\lambda_i$ = 0,0252 W/(mK)
$T_3$ = 22,68 °C
$T_2$ = 94,696 °C
$T_m$ = 58,68 °C

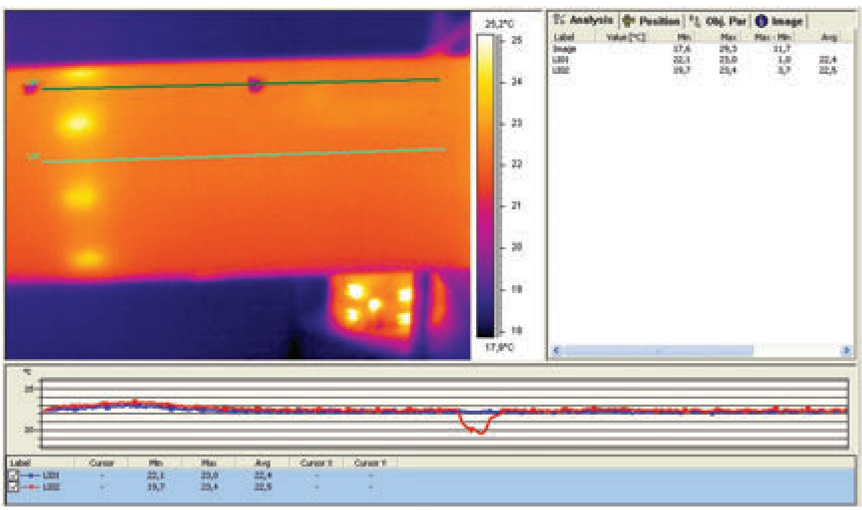

Figure 9. Thermogram of testing pipe I

Figure 10. Photo of testing pipe I

*Testing pipe II*

Inside temperature of service pipe: 84,4 °C (stationary state)
Average temperature on outer casing surface: 24,25 °C (measured by thermography)
Ambient temperature: 22,8 °C
Input power: 45,02 W
Casing diameters: $D_{C4}$ = 492,4 mm; $D_{C3}$ = 484,4 mm
Service pipe diameter $D_{S2}$ = 324,4 mm; $D_{S1}$ = 314,4 mm
Thermal conductivity: $\lambda_c$ =0,4 W/(mK), $\lambda_s$ = 50 W/(mK)
Length: $L$ = 2,11 m
Using equations (1) to (4) we obtain
$\lambda_i$ = 0,0231 W/(mK)
$T_3$ = 24,39 °C
$T_2$ = 84,39 °C
$T_m$ = 54,3 °C

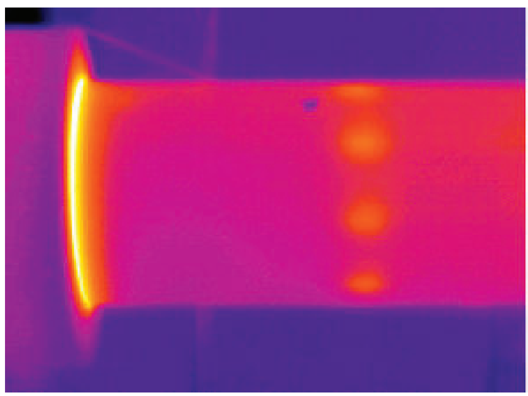

Figure 11. Thermogram of testing pipe II

Figure 12. Photo of testing pipe II

## CONCLUSION

Thermography can be successfully used not only for determining the thermal conductivity of insulating sandwich panels and prefabricated insulated pipe-in-pipe assemblies according to the requirements of existing standard but also for determination of non-uniformities between insulation and casing as well as for testing the quality of panels and pipes during production and on situ before being built in or buried.

## REFERENCES

1. HRN ISO 8302:1998. Thermal insulation-Determination of steady state thermal resistance and related properties-Guarded hot plate apparatus.
2. HRN EN ISO 8497:1988. Thermal insulation-Determination of steady state thermal transmission properties of thermal insulation for circular pipes.
3. EN 253:2003, District heating pipes-Preinsulated bonded pipe systems for directly buried hot water networks-Pipe assembly of steel service pipe, polyurethane thermal insulation and outer casing of polypropylene.
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# Stručni dio

# Pravilnik o označavanju energetske učinkovitosti kućanskih uređaja

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**SAŽETAK**

Svjetska i europska iskustva pokazuju da je označavanje energetske učinkovitosti kućanskih uređaja djelotvoran način u cilju razvitka energetski racionalnog nacionalnog tržišta i smanjenja potrošnje električne energije na nacionalnoj razini. U cilju usuglašavanja hrvatske energetske regulative s EU legislativom s jedne i uspostavi zakonodavnog okvira za energetski osviješteno tržište kućanskih uređaja s druge strane 17. studenog 2005. godine donesen je *Pravilnik o označavanju energetske učinkovitosti kućanskih uređaja (*Narodne novine 133/05). Pravilnik je u punoj primjeni od 1. svibnja 2006. godine.
Glavni razlozi za donošenje Pravilnika su:

- Usklađivanje sa zakonodavstvom Europske unije;
- Smanjenje potrošnje električne energije na nacionalnoj razini;
- Podizanje razine osviještenosti građana Hrvatske po pitanjima energetske učinkovitosti;
- Provedba odrednica iz strategijskih i razvojnih dokumenata na području učinkovitog korištenja energije u Republici Hrvatskoj;
- Potreba uređenja i utvrđivanja zadaća svih sudionika u procesu proizvodnje, uvoza, nabave, prodaje i korištenja kućanskih uređaja u Republici Hrvatskoj.

## 1. UVODNA RAZMATRANJA

U skladu sa Zakonom o energiji (NN 68/1) [1], temeljnom Direktivom o označavanju energetske učinkovitosti kućanskih uređaja 92/75/EEC i implementacijskim direktivama, 17. studenog 2005. godine stupio je na snagu Pravilnik o označavanju energetske učinkovitosti kućanskih uređaja (NN 133/05) (u primjeni od 1. svibnja 2006.) koji propisuje izgled i sadržaj oznaka energetske učinkovitosti za slijedeće grupe kućanskih uređaja: hladnjake, ledenice i njihove kombinacije, perilice rublja, sušilice rublja, kombinirane perilice-sušilice rublja, perilice posuđa, električne pećnice, klimatizacijske uređaje i izvore svjetlosti direktno napajane iz mreže [2].
Kao pomoć uspostavi energetski učinkovitijeg tržišta kućanskih uređaja u Hrvatskoj, u srpnju 2005. godine pokrenut je UNDP/GEF PDF B projekt: *Jačanje institucionalnog kapaciteta za uklanjanje prepreka implementaciji standarda i oznaka energetske učinkovitosti kućanskih uređaja u zemljama kandidatkinjama za Europsku uniju (S@L EUCC projekt).* Projekt je regionalnog karaktera, a zemlje sudionice uz Hrvatsku su Bugarska, Rumunjska i Turska.

## 2. PREGLED HRVATSKOG TRŽIŠTA KUĆANSKIH UREĐAJA

U sklopu UNDP/GEF S@L EUCC projekta, GfK Hrvatska je proveo istraživanja i analize o zastupljenosti pojedinih kućanskih uređaja u hrvatskim kućanstvima [3], prosječnoj starosti uređaja [4], zastupljenosti kućanskih uređaja prema energetskom razredu [5], te zastupljenosti kućanskih uređaja u ovisnosti o proizvođaču [6].
U tablici 1. prikazan je popis važnijih proizvođača, uvoznika i distributera kućanskih uređaja u Hrvatskoj.

Tablica 1. Popis proizvođača, uvoznika i distributera kućanskih uređaja u Hrvatskoj.

| Proizvođači i uvoznici s udjelom na hrvatskom tržištu većim od 5% | Distributeri kućanskih uređaja |
|---|---|
| Končar – Kućanski aparati | KONČAR |
| BOSCH | BRODOMERKUR |
| CANDY | ELEKTROPROMET |
| ELECTROLUX | ECOS TRGOVINA |
| GORENJE | E plus |
| INDESIT | EUROPATRADE |
| LG | ELEKTROMATERIJAL |
| WHIRPOOL | ELIPSO |
| ZANUSSI | GORENJE |
| ARISTON | IZZI komerc |
| Mielle | KONIKOM |
| QUATRO | KRALJ Appliances Shopping Centar |
| | SPARTAK |
| | MERCATONE |
| | Electron |
| | Frigo &CO |
| | GETRO |
| | METRO |
| | PEVEC |
| | Robot Commerce |
| | Merkur international |
| | Electron |

Tablica 2. Ukupan broj trgovina kućanskih uređaja

| **Distributer** | **Ukupan broj trgovina** |
|---|---|
| KONČAR | 12 |
| BRODOMERKUR | 52 |
| ELEKTROPROMET | 33 |
| ECOS TRGOVINA | 33 |
| E plus | 37 |
| EUROPATRADE | 61 |
| ELEKTROMATERIJAL - Euronics | 59 |
| ELIPSO | 7 |
| GORENJE | 12 |
| IZZI komerc | 10 |
| KONIKOM | 17 |
| KRALJ Appliances Shopping Centar | 6 |
| SPARTAK | 14 |
| MERCATONE | 3 |
| Electron | 8 |
| Frigo &CO | 7 |
| GETRO | 9 |
| METRO | 3 |

| | |
|---|---|
| PEVEC | 11 |
| Robot Commerce | 8 |
| Merkur international | 6 |

Tablica 3. Ukupan broj uvoznika pojedinog kućanskog uređaja

| Vrsta uređaja | Ukupan broj uvoznika | Najzastupnjeniji marke uređaja |
|---|---|---|
| Hladnjaci i ledenice | 147 | Gorenje, Electrolux, Konikom (Ariston, Indesit), Whirpool |
| Perilice rublja | 97 | Gorenje, Electrolux |
| Perilice posuđa | 105 | Gorenje, Electrolux,Whirpool |
| Električne pećnice | 47 | Gorenje, Electrolux, Whirpool |
| Klimatizacijski uređaji | 85 | LG, Samsung |

Prema istraživanju provedenom na representativnom uzorku od 1000 hrvatskih kućanstava u rujnu 2005. godine [3] zastupljenost pojedinih kućanskih uređaja je slijedeća:

- hladnjaci - 100%
- perilice rublja - 98%
- mikrovalne pećnice - 34%
- perilice posuđa - 29%
- sušilice rublja - 5%
- klimatizacijski uređaji - 23%
- pećnice - 94%

U veljači 2006. godine GfK Hrvatska je proveo OMNIBUS anketu na reprezentativnom uzorku od 1000 kućanstava o mišljenjima i stavovima hrvatskih građana vezano uz korištenje i štednju električne energije u kućanstvima [7]. Prema rezultatima ankete 64% hrvatskih građana vodi računa o potrošnji električne energije i nastoji se ponašati racionalno, 31% ih je svjesno potrebe racionaliziranja potrošnje energije ali ih iz raznih razloga ne realiziraju, dok 5% uopće ne vodi računa o potrošnji energije. Regionalno gledano, najviše energetski osviještenih građana ima u Slavoniji (74%), a najmanje u Istri s Primorjem (48%).
Nadalje, anketa pokazuje da 74% kućanstava ne posjeduje niti jedan kućanski uređaj s oznakom energetske učinkovitosti, 21% kućanstava posjeduje barem jedan takav uređaj, dok 8% ispitanih ne zna odgovor.

Na pitanje da li su spremni investirati više novca za kupnju energetski učinkovitog kućanskog uređaja odgovori su bili slijedeći:

- svakako da - 35%
- vjerojatno da - 47%
- vjerojatno ne - 9%
- sigurno ne - 3%
- ne zna - 6%.

Spremnost na kupnju energetski učinkovitih uređaja najveća je u Slavoniji (42%) i u kućanstvima s visokim mjesečnim prihodima (56%). Vjerojatnost takve kupnje iskazana je najviše duž cijele obale (51%) i u kućanstvima sa srednjom razinom mjesečnih primanja.

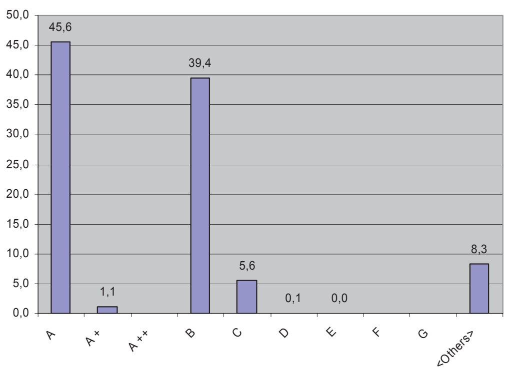
Odgovori na pitanje da li bi se odlučili na kupnju energetski učinkovitog kućanskog uređaja pokazuju slijedeće:

- svakako da - 52%
- vjerojatno da - 39%

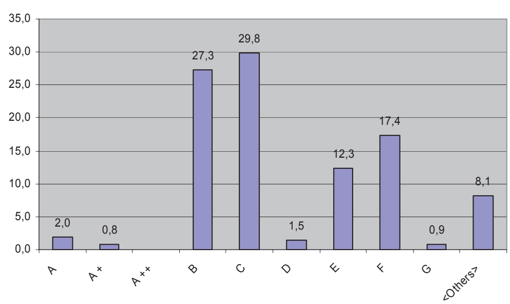
- vjerojatno ne - 4%
- sigurno ne - 2%
- ne zna - 4%.

Rezultati provedene ankete pokazuju da su hrvatski građani svjesni potrebe racionalnog korištenja električne energije i da ih je potrebno kontinuirano informirati o načinima povećanja energetske učinkovitosti u kućanstvima što će u konačnici rezultirati smanjenjem potrošnje električne energije na nacionalnoj razini.

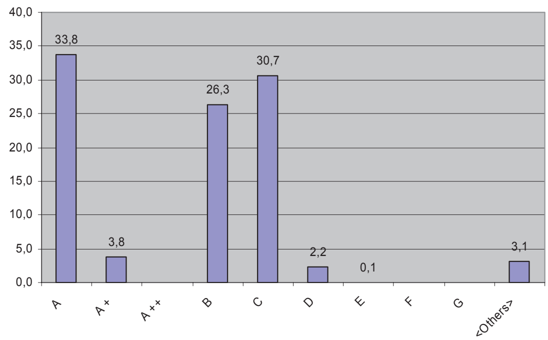
Zastupljenost kućanskih uređaja na hrvatskom tržištu prema razredu energetske učinkovitosti [5] prikazana je na slikama 1-4.



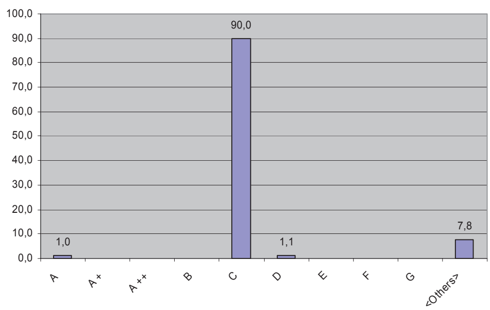

Slika 1: Zastupljenost hladnjaka na hrvatskom tržištu prema razredu energetske učinkovitosti




Slika 2: Zastupljenost ledenica na hrvatskom tržištu prema razredu energetske učinkovitosti




Slika 3: Zastupljenost perilica rublja na hrvatskom tržištu prema razredu energetske učinkovitosti




Slika 4: Zastupljenost sušilica rublja na hrvatskom tržištu prema razredu energetske učinkovitosti

## 3. PRAVILNIK O OZNAČAVANJU ENERGETSKE UČINKOVITOSTI KUĆANSKIH UREĐAJA

### 3.1 Ciljevi i razlozi za donošenje Pravilnika

Jedan od glavnih ciljeva Pravilnika je poticanje racionalnog korištenja energije i energetski osviještenog tržišta kućanskih uređaja što treba rezultirati znatnim smanjenjem potrošnje energije i onečišćujućih tvari u atmosferu na nacionalnoj razini.
Utjecaj Pravilnika u gospodarskom i socijalnom smislu očitovati će se, između ostalog, u:

- Pružanju boljeg uvida u isplativost kupnje određenog tipa kućanskih uređaja;
- Smanjenju potrošnje električne energije u kućanstvima;
- Uklanjanju energetski neučinkovitih kućanskih uređaja s hrvatskog tržišta.

### 3.2. Zakonska podloga za donošenje Pravilnika

Zakonska podloga za donošenje Pravilnika je Zakon o energiji od 19. srpnja 2001. (NN 68/01) [1] i Zakon o izmjenama i dopunama Zakona o energiji od 3. prosinca 2004. (NN 177/04) .
Članak 13. Zakona o energiji glasi: Svi uređaji koji koriste energiju trebaju biti opremljeni oznakom energetske učinkovitosti. Sadržaj i izgled oznake energetske učinkovitosti regulirani

su podzakonskim aktom koji donosi ministar resornog Ministarstva gospodarstva, rada i poduzetništva.

### 3.3 Usklađivanje Pravilnika sa zakonodavstvom Europske unije

Pravilnik o označavanju energetske učinkovitosti kućanskih uređaja usuglašen je sa sljedećim Direktivama Europske komisije:

- Direktivom Europske komisije 92/75/EEC od 22. rujna 1992. godine – temeljna direktiva
- Direktivom Europske komisije 94/2/EC od 21. siječnja 1994. godine o obaveznom energetskom označavanju kućanskih električnih hladnjaka, ledenica i njihovih kombinacija
- Direktivom Europske komisije 95/12/EC od 23. svibnja 1995. godine o obaveznom energetskom označavanju kućanskih električnih perilica rublja
- Direktivom Europske komisije 95/13/EC od 23. svibnja 1995. godine o obaveznom energetskom označavanju kućanskih električnih sušilica rublja
- Direktivom Europske komisije 96/60/EC od 19. rujna 1996. godine o obaveznom energetskom označavanju kućanskih električnih kombiniranih perilica/sušilica rublja
- Direktivom Europske komisije 2002/40/EC od 8. svibnja 2002. godine o obaveznom energetskom označavanju kućanskih električnih pećnica
- Direktivom Europske komisije 97/17/EC od 16. travnja 1997. godine o obaveznom energetskom označavanju kućanskih električnih perilica posuđa
- Direktivom Europske komisije 2002/31/EC od 22. svibnja 2002. godine o obaveznom energetskom označavanju kućanskih klimatizacijskih uređaja
- Direktivom Europske komisije 98/11/EC od 27. siječnja 1998. godine o obaveznom energetskom označavanju izvora svjetlosti u kućanstvima

Pravilnik ustvari objedinjuje gore navedenih 10 direktiva Europske komisije, s time da su tehničke odredbe Pravilnika u potpunosti usuglašene sa spomenutim direktivama, a ulaskom Hrvatske u Europsku uniju biti će usuglašene i opće odredbe.

### 3.4. Najvažnije odredbe Pravilnika o označavanju energetske učinkovitosti kućanskih uređaja

Pravilnikom se određuju kućanski uređaji koji moraju biti označeni oznakom energetske učinkovitosti te propisuje oblik i sadržaj oznake o energetskoj učinkovitosti kućanskih uređaja.

Odredbe Pravilnika primjenjuju se na sljedeće skupine kućanskih uređaja:

- hladnjake i ledenice, te njihove kombinacije;
- perilice rublja;
- bubnjaste sušilice rublja,
- kombinirane perilice -sušilice rublja;
- perilice posuđa;
- električne pećnice;
- klimatizacijske uređaje;
- električne izvore svjetla napajane direktno iz mreže.

Gore navedenim skupinama uređaja mogu biti dodane i daljnje skupine kućanskih uređaja u skladu s njihovom rasprostranjenošću na hrvatskom tržištu i utjecajem na potrošnju energije.

Članak 3. Pravilnika kaže da svi gore navedeni uređaji koji za pogon koriste električnu energiju, a proizvode se ili uvoze u Republiku Hrvatsku radi stavljanja na hrvatsko tržište, moraju biti označeni oznakom energetske učinkovitosti.
Podaci o energetskoj učinkovitosti kućanskih uređaja na oznaci energetske učinkovitosti moraju biti označeni u skladu s odredbama ovoga Pravilnika. Uvođenjem oznake energetske učinkovitosti na jednostavan se i jasan način kupcima kućanskih uređaja daje informacija o energetskoj učinkovitosti uređaja i omogućuje usporedba raznih modela s obzirom na njihove energetske karakteristike.
Pravilnik se ne primjenjuje na:

1. kućanske uređaje koji koriste autonomne izvore energije,
2. kućanske uređaje čija je proizvodnja prestala prije stupanja na snagu Pravilnika,
3. kućanske uređaje koji su stavljeni na tržište prije stupanja na snagu Pravilnika
4. rabljene kućanske uređaje.

Nadalje, odredbe Pravilnika ne primjenjuju se na:

1. perilice rublja:
   - bez mogućnosti centrifugiranja,
   - s odvojenim bubnjevima za pranje i sušenje;
2. prijenosne električne pećnice mase manje od 18 kg;
3. kućanske klimatizacijske uređaje čija je izlazna snaga hlađenja veća od 12 kW;
4. izvore svjetlosti:
   - čiji je svjetlosni tok (luminacijski fluks) veći od 6 500 lumena,
   - čija je instalirana snaga manja od 4 W,
   - reflektorskog tipa:
     - koji su deklarirani i primarno se prodaju kao trošila drugih izvora energije (npr. baterija),
     - koji nisu deklarirani za proizvodnju svjetlosti u vidljivom spektru (400 do 800 nm),
   - koji su deklarirani kao dio proizvoda čija osnovna namjena nije rasvjetna.

Karakteristično za izvore svjetlosti, odredbe Pravilnika primjenjuju se na:

1. izvore svjetlosti u kućanstvima:
   - klasične (volframove) žarulje,
   - kompaktne fluorescentne cijevi (štedne žarulje),
   - fluorescentne cijevi;
2. fluorescentne cijevi koje su deklarirane za korištenje izvan kućanstva (uslužni sektor, uredske zgrade i dr.).

U slučaju električnih pećnica, ukoliko pećnica ima više od jednog prostora za pečenje svaki od njih treba imati vlastitu oznaku energetske učinkovitosti.

Dobavljač (proizvođači, trgovci na veliko, uvoznici, predstavnici ili zastupnici proizvođača koji nemaju sjedište u Republici Hrvatskoj, te ovlašteni predstavnici proizvođača u Republici Hrvatskoj) je dužan uz kućanski uređaj koji isporučuje distributoru (trgovci na malo koji kućanske uređaje stavljaju na tržište ili omogućuju njihovu dostavu krajnjim kupcima), bez posebne naknade, dostaviti oznaku energetske učinkovitosti te tehničku dokumentaciju, koja potvrđuje detaljnija objašnjenja o podacima na oznaci.

Spomenuta tehnička dokumentacija treba sadržavati:

- naziv (tvrtku) i adresu dobavljača kućanskih uređaja,
- opći opis kućanskog uređaja dovoljan za identifikaciju modela,
- podatke o karakteristikama modela, posebno onih koji utječu na potrošnju energije,
- izvještaje o provedenim ispitivanjima i testiranjima proizvoda od mjerodavnih institucija,
- upute za uporabu.

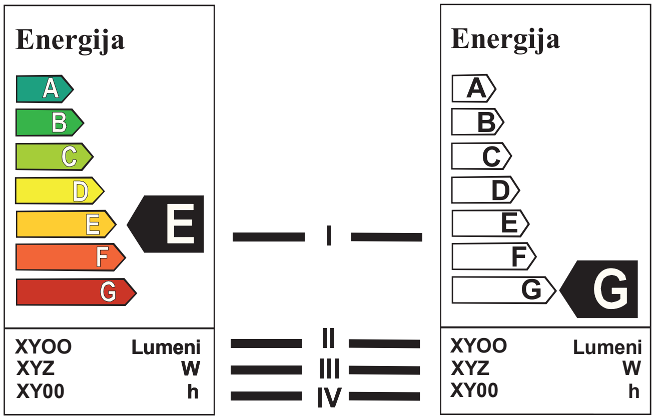
Za točnost podataka koje sadrži oznaka energetske učinkovitosti odgovoran je dobavljač koji je ujedno dužan čuvati tehničku dokumentaciju uređaja najmanje 5 godina.

Dobavljač sam određuje način dostave oznake energetske učinkovitosti:

1. Oznaka energetske učinkovitosti može se dostaviti kao jedinstvena oznaka koja sadrži podatke o proizvođaču kućanskog uređaja, vrsti, tipu i modelu kućanskog uređaja, razredu učinkovitosti te potrošnji energije kao vrijednosti specifične za pojedinu vrstu, tip i model kućanskog uređaja.

2. Oznaka energetske učinkovitosti može se dostaviti kao temeljna oznaka koja s lijeve strane sadrži podatke o proizvođaču kućanskog uređaja, vrsti, tipu i modelu kućanskog uređaja, razredu učinkovitosti te potrošnji energije, ali ne sadrži vrijednosti specifične za pojedinu vrstu, tip i model kućanskog uređaja. Vrijednosti specifične za pojedinu vrstu, tip i model kućanskog uređaja dostavljaju se na dodatnoj oznaci (traci s podacima) koju distributor postavlja na predviđeno mjesto na temeljnoj oznaci. Time se omogućava korištenje jedinstvene temeljne oznake za jedan tip kućanskih uređaja, dok se traka s podacima može dostaviti izravno iz države podrijetla kućanskog uređaja bez potrebe za prevođenjem.

Distributor je dužan na kućanski uređaj, koji je u prodavaonicama izložen radi prodaje vidljivo istaknuti oznaku učinkovitosti propisanu ovim Pravilnikom.
Pravilnik nadalje propisuje, da ukoliko se kućanski uređaj prodaje putom poštanske dostave, kataloške prodaje, Interneta ili na drugi način koji ne omogućuje kupcu uvid u tehničku dokumentaciju uređaja, potrebno je osigurati da se mogući kupac prije kupnje proizvoda upozna s podacima s energetske oznake.
Detaljni sadržaj oznake energetske učinkovitosti utvrđen je posebno za svaku skupinu kućanskih uređaja a kao primjer je na slici 5. prikazana oznaka energetske učinkovitosti izvora svjetlosti napajanih iz mreže (žarulja) u boji i u crno-bijeloj verziji .




Slika 5. Oznaka energetske učinkovitosti žarulja

Detalji oznake prema slici 5 imaju sljedeće značenje:

I. Razred energetske učinkovitosti izvora svjetlosti
II. Svjetlosni tok izvora svjetlosti izražen u lumenima (lm)
III. Nazivna snaga izvora svjetlosti izražena u W
IV. Nazivni životni vijek izvora svjetlosti izražen u satima. Ako na ambalaži izvora svjetlosti nije naveden nikakav podatak o životnom vijeku, može biti izostavljen i s oznake.

Mjerni postupci za dobivanje podataka koje sadrži oznaka energetske učinkovitosti provode se u skladu s hrvatskim normama navedenima u tablici 4.

Tablica 4. Hrvatske norme za skupine kućanskih uređaja obuhvaćene Pravilnikom

| Norma | Skupina uređaja |
|---|---|
| HRN EN 153 | Hladnjaci i ledenice, kombinirani hladnjaci |
| HRN EN 60456 | Perilice rublja |
| HRN EN 50229 | Kombinirane perilice-sušilice rublja |
| HRN EN 61121 | Bubnjaste sušilice rublja |
| HRN EN 50242 | Perilice posuđa |
| HRN EN 50304 | Električne pećnice |
| HRN EN 814-1<br>HRN EN 814-2<br>HRN EN 814-3 | Klimatizacijski uređaji |

Inspekcijski nadzor nad provedbom Pravilnika obavljaju Državni inspektorat i drugi nadležni inspektori prema posebnim propisima, sukladno odredbama članka 32. stavka 2. Zakona o energiji (Narodne novine 68/2001 i 177/2004).

Nadležno inspekcijsko tijelo u provedbi inspekcijskog nadzora može:

- zahtijevati da se proizvodi opskrbe propisanim oznakama, odnosno da se uklone nedopuštene oznake,
- zabraniti stavljanje na tržište, ograničiti stavljanje na tržište ili odrediti povlačenje nesukladnih proizvoda s tržišta i provesti dodatne mjere koje će osigurati da se ta zabrana poštuje.

Za uspješnu provedbu Pravilnika od velike je važnosti u što većoj mjeri informirati korisnike kućanskih uređaja s jedne, te iznaći što djelotvornije načine kontrole i nadzora njegove provedbe s druge strane.

## 5. ZAKLJUČAK

Obaveza označavanja energetske opreme postavlja jasne zahtjeve pred proizvođače, uvoznike i prodavače uređaja i na veoma djelotvoran način im onemogućuje da svjesni relativno niskog životnog standarda prosječnog potrošača, pretvaraju Hrvatsku u veliko tržište neefikasne energetske opreme. Uvođenje oznaka energetske učinkovitosti, između ostalog, omogućuje kupcima uređaja jednostavnu usporedbu energetskih karakteristika različitih modela dostupnih na tržištu. Uspješna provedba Pravilnika o označavanju energetske učinkovitosti kućanskih uređaja rezultirati će uspostavom energetski racionalnog tržišta kućanskih uređaja na kojem će svi akteri voditi računa o energetskoj potrošnji što će se u konačnici odraziti na značajno smanjenje potrošnje električne energije na nacionalnoj razini.

## L I T E R T U R A

[1] Zakon o energiji, Narodne novine 68/01

[2] Pravilnik o označavanju energetske učinkovitosti kućanskih uređaja, Narodne novine 133/05

[3] GfK Hrvatska, Istraživanje o zastupljenosti kućanskih uređaja u hrvatskim kućanstvima, rujan 2005.

[4] Vlasta Kroflin Fišer, GfK Hrvatska, Istraživanje o starosti uređaja i planovima za nabavku novih uređaja u hrvatskim kućanstvima, prosinac 2005

[5] Vlasta Kroflin Fišer, GfK Hrvatska, Istraživanje o zastupljenosti razreda energetske učinkovitosti kućanskih uređaja, prosinac 2005.

[6] GfK Hrvatska, Istraživanje o zastupljenosti kućanskih uređaja pojedinih proizvođača na hrvatskom tržištu, ožujak 2006

[7] GfK Hrvatska, OMNIBUS anketa o potrošnji električne energije u kućanstvima, veljača 2006.

# TOPLINSKA BILANCA LJUDSKOG TIJELA U VODI

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SAŽETAK :
Ovaj članak je povezan sa prethodnim člankom „Termička ravnoteža ljudskog tijela kod kupanja" . Tema u ovom članku je da se kvantitativno determinira toplinski tok sa ljudskog tijela kada je uronjeno u vodu. Na osnovu provedenih proračuna dobiveni su konačni traženi rezultati. Iz rezultata potvrđeno je da toplinski gubitak ljudskog tijela u vodi može biti značajan u različitim kombinacijama parametara.
U članku su prezentirani i podaci za morsku vodu. U konačnici rezultati pokazuju neznatnu razliku u odnosu na čistu vodu.

SUMMARY:

Zdravko Poša ,B.Sc.(Eng)
THERMAL BALANCE OF HUMAN BODY IN THE WATER .

This article is connected to previous article „Thermal balance of human body at bathing". Topic in this article is to determine heat flux from human body when submersed in the water . Basic calculatins are done and final results are obtained . From results we can conclude that heat loss from human body in the water exists and can be extreme at various combinations of parameters.
In article also data for sea water are presented and it shows some slight difference regarding to the water.

KLJUČNE RIJEČI :

| | |
|---|---|
| ljudsko tijelo | human body |
| termička bilanca | thermal balance |
| voda | water |
| more | sea water |

UVOD
Na 18. Međunarodnom simpoziju o hlađenju i klimatizaciji Interklima 2005 – Zagreb prezentiran je članak pod naslovom TOPLINSKA BILANCA LJUDSKOG TIJELA KOD KUPANJA . Potaknut pozitivnim reakcijama na navedeni članak pristupio sam kompletiranju razmatranja toplinskih procesa ljudskih tijela u okruženju vode tj kod kupanja . Stoga u ovom radu ,za razliku od mojeg prethodnog , objekt posmatranja je ljudsko tijelo uronjeno u vodi i njegova termička bilanca .

Ljudski organizam može se promatrati kao toplinski uređaj sa svim svojim zakonitostima . Za obavljanje osnovnih funkcija ljudskog organizma odnosna njegovih internih organa (srca,mozga,jetre itd) proizvodnja energije u organizmu je neprekidna .Ta energija se razvija u organizmu putem oksidacije hrane. Taj neprekidni proces koji održava temperaturu ljudskog organizma približno konstantnom (oko 37$^{o}$ C) kao i rad unutarnjih organa nazvan je bazalni metabolizam ($Q_{mb}$). On iznosi oko 1,2 do 1,5 W /kg tjelesne težine čovjeka. U uvjetima povećane aktivnosti ta proizvodnja toplinske energije u tijelu tj. metabolizam može se povećati i do deset puta. Povećana proizvodnja energije u tijelu mora biti povezana i sa adekvatnim i efikasnim načinom odvođenja. Naime podržavanje normalne temperature ljudskog tijela (37 $^{0}$C) je vrlo bitno jer gornja granica,i to fatalna je 43 $^{0}$C . Ista taka postoji i donja granična vrijednost temperature ljudskog organizma . Kod temperature središnjice tijela od 25 $^{0}$C život također nestaje.

Pod pretpostavkom da je uspostavljena termoregulacija odnosno ravnoteža između proizvedene i odavane energije može se reći da odavana energija H (W) sastoji se od dvije komponente :

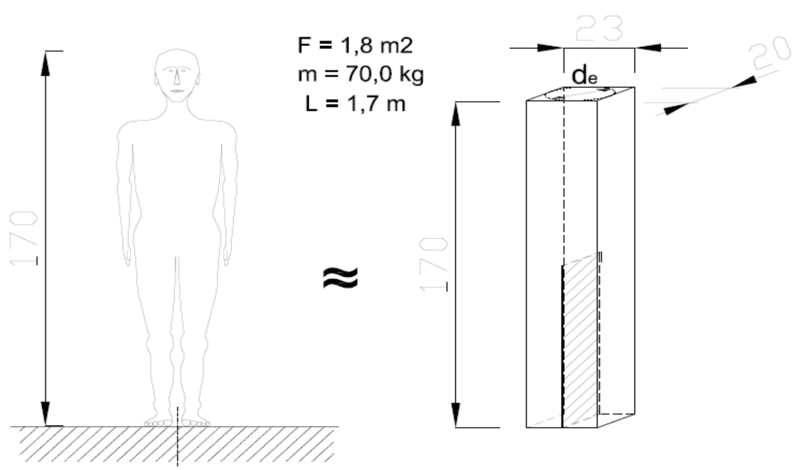
- osjetne($H_o$ )
- latentne ($H_l$ )

$$H = H_o + H_l \text{ (W)} \tag{1}$$

Odnos odavane topline (H) i udjela komponenti odavanja ($H_o$,$H_l$) regulirana je čovječjim mehanizmom termoregulacije. Dvije vrste nervnih perceptora, za hladno i toplo na ljudskoj koži primaju utjecaj okoline i stvaraju impulse koji nervni sistem prenosi i utiče na:

- metabolizmom povećane proizvodnje topline kod osjećaja hladnoće
- povećanje odavanja topline kod osjećaja toplote

Ljudsko tijelo nije pravilno geometrijsko tijelo. To predstavlja poteškoću u određivanju toplinskih procesa . Radi matematičkog računanja, u proračunima što slijede, može se zamijeniti kao bliska aproksimacija ljudsko tijelo sa prizmatičnim tijelom približno iste površine i volumena kako je prikazano i na slici 1 a što je korišteno i u spomenutom radu [lit. 2/] .




Slika1. – Model zamjene ljudskog tijela sa prizmatičnim tijelom

Ljudsko tijelo u vodi se mora održavati na površini sa gibanjem ekstremiteta .Dakle prisutan je ljudski rad za koju postoji zakonitost

$Q_m$ = L+H (W) …………………………………………………………..(2)

U izrazu Qm(W) je toplina koja se metabolizmon stvara u ljudskom tijelu a L(W) je je korisni izvršeni rad a H(W) je toplina koju tijelo ,kao i svaki toplinski uređaj ,odbacuje u okolinu . Ta toplina se može izraziti i sa stepenom korisnosti mehaničkog rada ŋ . Za laganu fizičku aktivnost numerička vrijednost ŋ je približno jednaka 0. Inače granična odnosno maksimalna vrijednost iskoristivosti za ljudsko tijelo je 0,2.

$$H = Q_m . (1-ŋ) \;\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3)$$

U literaturi postoje navedene vrijednosti $Q_m$, ŋ . Usvaja se iz literature /7/ kao najbliža vrijednost 160(W) a što je dano u tablici 1. Vrijednosti u tablici su važeća za ljudsko tijelo odrasle osobe u okruženju zraka temperature $24^0$C .

| Opis aktivnosti | $Q_m(W/m^2)$ | $Q_m(W)$ |
|---|---|---|
| Sjedenje (kazalište) | 63,8 | 115 |
| Sjedenje (ured) | 72,8 | 130 |
| Stajanje,lagano hodanje (trgovine) | 88,8 | 160 |
| Hodanje ,stajanje (trgovine) | 88,8 | 160 |
| Rad za stolom (tvornice) | 130,5 | 235 |
| Umjereno plesanje | 147,2 | 265 |

Tablica 1.- Vrijednosti odavanja topline ljudskog tijela

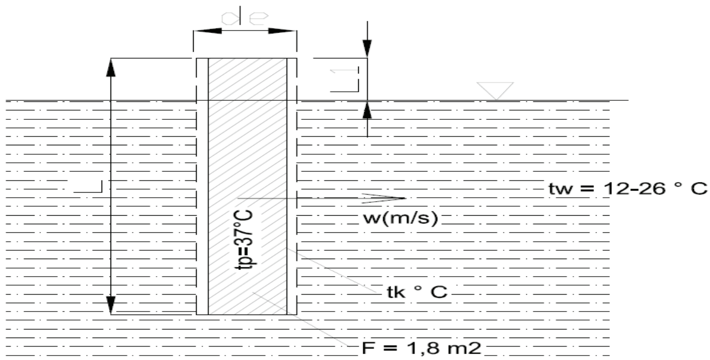
Naznačena toplina odavanja u okruženju zraka odnosno na kopnu se odvodi prema poznatim postupcima termoregulacije sa procesima konvekcije, kondukcije, zračenjem i isparivanjem . U vodi se situacija mijenja zbog značajnog gubitka topline sa površine ljudske kože . Kad kažemo gubitak time podrazumijevamo da je temperatura vode odnosno tekućine manja od temperature tijela. Toplinska bilanca ljudskog tijela uronjeno u vodi može biti negativnog ili pozitvnog predznaka a iznosi :

$$\Sigma Q = Q_m - H_k - (Ho+Hi)\text{“} \;\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(4)$$

Drugi član u izrazu je kovektivni gubitak u vodi a izraz (Ho+Hi)“ je zbir osjetnih i latentnih gubitaka tijela u okruženju zraka . Prema podacima iz literature [2] i [7] možemo kao dozvoljenu aproksimaciju odrediti izraz (Ho+Hi)“ kao 15% od metaboličke topline. Te u konačnici imamo :

$$\Sigma Q = Q_m .(1-0,15) - H_k = 160 .0,85 - H_k = 136,0 - H_k \text{ (W)}$$

Ljudsko tijelo u vodi može biti u približno horizontalnom ili vertikalnom položaju. Horizontalni položaj tijela je karakterističan za intenzivnije plivanje tj gibanje. Vertikalni položaj pak je vezan uz laganije plivanje ili plutanje na vodi . Brzina plivanja ili relativno kretanje tijela naspram tekućini može biti slabo ili intezivno . Raspon brzina je od 0 m/s ,za plutanje ili slabo plivanje pa sve do 2,0m/s za vrhunske sportaše . Za naše razmatranje usvojimo slučaj slabog plivanje sa položajem tijela u vertikalnom položaju i brzinom od w = 0,15 (m/s) .




Slika 2-Model tijela u vodi

Toplinski gubitak sa ljudskog tijela na vodu određen je izrazom :

$H_k = (F-F_1) \cdot \alpha \cdot (t_k - t_w)$ ( W) ............................................ (5)

U izrazu poznate veličine su $F(m^2)$ ukupna površina ljudskog tijela, temperatura vode $t_w(^0C)$ a $F_1(m^2)$ površina tijela izvan vode . Podrazumijevamo da je izvan vode glava i vrat. Iz podataka iz literature [5] možemo izračunati $F_1$ kako glasi:

$F_1 = F_{glave} + F_{vrata} = 0{,}103 + 0{,}029 = 0.132$ $(m^2)$

Vrijednosti koje treba proračunati su koeficijent prolaza topline na ljudskoj koži $\alpha(W/m^2,^0)$ i granična temperatura kože $t_k$ $(^0C)$.
Kad dva tijela različite temperature dođu u kontakt tada se na dodirnim površinama javlja kontaktna temperatura $t_k(^0C)$. Za slučaj ljudskog tijela u vodi , sa malom brzinom relativnog kretanja, usvojena je kao aproksimacija postupak određivanja kontaktne temperature prema izrazu :

$$t_k = \frac{t_1 . b_1 + t_2 . b_2}{b_1 + b_2} \qquad \text{.................................(6)}$$

U izrazu figurira ,pored temperatura , i koeficijenti prodiranja toplote „b“ . Ono se izračunava prema izrazu iz literature [3]:

$$b = \sqrt{\lambda . \rho . c_p} \quad (W, s^{0,5}, m^{-2}, K^{-1}) \text{ ............................. (7)}$$

U medicinskoj literaturi ljudska koža je kao zasebni organ naznačena kao „izmjenjivač topline“ . Prema navodu iz literature [5] ljudska koža se modelira kao dvoslojna materija sa različitim fiziološkim svojstvima .Unutarnji sloj kože je debljine cca 1mm u kojemu se dovodi metabolička toplina sa krvlju. U vanjskom sloju kože ,slične debljine, ne postoji nikakvi izvori topline . Međutim taj sloj sa znojnim žlijezdama igra važnu ulogu kada nastupa hlađenje tijela putem evaporacije . Ispod kože nalazi se sloj masnog tkiva koji svojim fizikalnim karakteristikama određuju intenzitet provođenja topline.Prenosimo stoga u Tablici 2 iz medicinske literature [5] vrijednosti za spomenute slojeve ljudskog tijela kao i proračunati koeficijent „$b_2$“. U proračunima koristi će se srednja vrijednost koeficijenta prodiranja topline kao „$b_{2\,sr}$“.

| | L(cm) | $\alpha_{sr}$ | $d_e$(cm) | $\lambda$ | $\rho$ | $c_p$ | $b_2$ |
|---|---|---|---|---|---|---|---|
| Toraks - kožno tkivo | 30,6 | 7,4 | 25,28 | 0,47 | 1085 | 3,68 | 43,31 |
| Toraks -potkožno tkivo | 30,6 | 7,4 | 25,28 | 0,16 | 850 | 2,30 | 17,68 |
| Srednja vrijednost koeficjenta prodiranja topline $b_{2\,sr}$ | | | | | | | 30,495 |

Tablica 2.- Fizikalne vrijednosti za dijelove ljudskog tijela

Ostaje još da se determinira koeficijent prolaza topline . Koeficijent prolaza topline je definiran iz Nusseltove bezdimenzionalne značajke strujanja,skraćeno Nu i to iz izraza :

$$\alpha_k = \frac{N_u.\lambda}{l_e} \ (\mathrm{W/m^2,{}^0K}) \ \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots..(8)$$

U izrazu (8) $l_e$ je ekvivalentna dužina uronjenog tijela za poprečno nastrujavanje i određena je izrazom uz poznavanje ekvivalentnog promjera tijela. Konačni izraz za Nuseltov bezdimenzionalni broj prema literaturi [3] :

$$\mathrm{Nu} = 0{,}3 + \left[\ 0{,}441.\mathrm{Re}.\mathrm{Pr}^{0{,}667} + \frac{\mathrm{Re}^{1{,}6}.\mathrm{Pr}^{2}}{[\ 27{,}027 + 66{,}027.\mathrm{Re}^{-0.1}.(\mathrm{Pr}^{0{,}667} - 1)\ ]^{2}}\ \right]^{0{,}5} \ \ldots\ldots\ldots\ldots\ldots.(9)$$

Primjenjivost izraza vrijedi uz ograničenja 10< Re >$10^7$ , te i uz 0.6< Pr >2000. U gornjem izrazu figuriraju također bezdimenzionalne začajke Reynolds (Re) i Prandlova (Pr). One su definirane izrazima :

$$\mathrm{Re} = \frac{w.l_e}{\nu} \ ,$$

$$\mathrm{Pr} = \frac{\eta.c_p}{\lambda} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(10)$$

| | $t_w = 12^0C$ | $15^0C$ | $17^0C$ | $20^0C$ | $26^0C$ |
|---|---|---|---|---|---|
| λ | 0,5846 | 0,590 | 0,591 | 0,599 | 0,608 |
| cp | 4,196 | 4,191 | 4,187 | 4,182 | 4,179 |
| ρ | 998,8 | 998,5 | 998,3 | 998,0 | 996,2 |
| ν | $1{,}322.10^{-6}$ | $1{,}205.10^{-6}$ | $1{,}123.10^{-6}$ | $1{,}01.10^{-6}$ | $0{,}904.10^{-6}$ |
| Re | 40847 | 44 813 | 48 085 | 53 465 | 59 708 |
| Pr | 8,91 | 8,19 | 7,70 | 6,98 | 6,18 |
| Nu | 508,09 | 529,0 | 546,60 | 573,09 | 596,69 |
| α | 825,08 | 866,97 | 897,34 | 953,55 | 1007,74 |
| b1 | 1565,29 | 1571,33 | 1575,47 | 1581,14 | 1590,96 |
| $b2_{sr}$ | 30,49 | 30,49 | 30,49 | 30,49 | 30,49 |
| $t_k$ | 12,401 | 15,342 | 17,303 | 20,245 | 26,131 |
| $H_k$ (W) | 551,8 | 494,5 | 453,5 | 389,6 | 220,2 |
| ΣQ (W) | **-415,8** | **-358,5** | **-317,5** | **-253,6** | **-84,2** |

Tablica 3- Numeričke vrijednosti fizikalnih veličina za tijelo uronjeno u vodu w=0,15 m/s

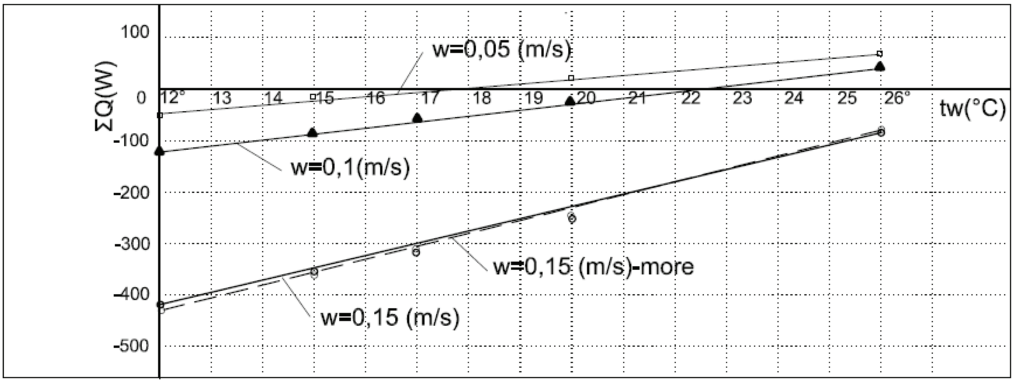
Sa spomenutim izrazima od (5) do (10) možemo numerički odrediti potrebne veličine koje su prikazane u tabeli 3. U razmatranje uzet je raspon temperatura koje se mogu očekivati kao ekstremne .To su na primjer kao temperature mora u zimskom periodu ($12^0C$) i maksimalna tokom ljeta ($26^0C$).

Dosadašnje razmatranje je vršeno za vodu . Postavlja se pitanje koliko su dobiveni rezultati primjenjivi i za more. More po sastavu je voda sa rastopljenim solima i mineralima . Otopljeni minerali mijenjaju fizikalne karakteristike a količina rastopljenih minerala i soli varira od lokacije do lokacije. Stoga je uveden parametar nazvan „salinitet mora „ koji označava količinu minerala u gramima na 1 kilogram morske vode. Salinitet mora iznosi od 33 do 37 promila i u proračunima je preporučljivo usvajati 35 . U donjoj tablici su dane vrijednosti ulaznih parametara za salinitet 35 te i proračunate konačne vrijednosti

| | $t_m= 12^0C$ | $15^0C$ | $17^0C$ | $20^0C$ | $26^0C$ |
|---|---|---|---|---|---|
| λ | 0,581 | 0,586 | 0,589 | 0,594 | 0,600 |
| $c_p$ | 3,988 | 3,991 | 3,992 | 3,993 | 3,996 |
| ρ | 1024,8 | 1023,84 | 1023,2 | 1022,2 | 1018,7 |
| $H_k$ (W) | 558,4 | 498,5 | 450,9 | 376,8 | 214,8 |
| ΣQ (W) | **-422,4** | **-362,5** | **-314,9** | **-240,8** | **-78,8** |

Tablica 4 – Numeričke vrijednosti fizikalnih veličina za tijelo uronjeno u more uz w=0,15

Dobivene vrijednosti iz tablice 3 i tablice 4 predočimo u grafički prikaz (slika 3). Pored toga predočene su i dobivene vrijednosti istim postupkom za brzine w=0,05 (m/s) i w=0,1 (m/s). Sa grafičkim prikazom rezultati odnosno zaključci nam se zornije prikazuju. To su :




Slika 3- dijagram proračunatih numeričkih gubitaka

-Što je temperatura vode niža to se toplinski gubici sa tijela povećavaju.
-Što je relativna brzina kretanja u vodi veća toplinski gubici sa ljudskog tijela se značajno povećavaju . Ta zakonitost je osnova u uputama za preživljavanje utopljenika . Naime u okruženju hladne vode utopljenicima se preporuča održavanje vertikalnog položaja tijela sa skupljenim ekstremitetima i bez nepotrebnog gibanja [literatura 10.].
-Razlika toplinskih toka sa ljudskog tijela u vodi i morskoj vodi se neznatno razlikuje . Neznatna razlika je primjetljiva pri nižim temperaturama promatranog raspona .
-U provedenim proračunima dobiveni su numerički rezultati toplinske bilance u rasponu od -415(W) pa do +72(W) . Izjednačavanje metaboličke topline sa ukupnim gubicima ljudskog tijela kada je u vodi (ΣQ=0) nastupa kod temperature vode $18^0$ C pa na više , ovisno o intenzitetu gibanja .
Dobivene rezultate možemo prodiskutirati u odnosu na spomenuti članak u uvodu [2]. U odnosu kada je tijelo izvan vode tj. period isparivanja vode sa ljudske kože , toplinski gubici u vodi mogu biti vrlo značajni pa i u ekstremnim slučajevima opasni . To nam potvrđuje podatak iz literature [9] . Period preživljavanja za prosječno nago ljudsko tijelo, u vodi temperature $12^0$ navedeno je da iznosi samo 5 sati. Dakle period boravka u vodi ,generalno govoreći, pridonosi ukupnom gubitku topline tj. energije iz ljudskog tijela .
Na kraju osvijetlimo dobivene rezultate kako koreliraju sa saznanjima u drugoj grani nauke koja proučava ljudsko tijelo. To je medicina odnosno podgrana fiziologija . U fiziologiji ovakav pristup za čitavo ljudsko tijelo ne primjenjuje se odnosno nije uobičajen. Naime u medicinskim istraživanjima žele se dobiti rezultati sa većom preciznosti . Zbog toga ljudsko tijelo raščlanjuje se na dijelove koji pokazuju slične karakteristike i zakonitosti .U fiziologiji glavni su pokušaji da se razumije kontrolni mehanizmi termoregulacije u odnosu na temperaturni podražaj. Naznačeno je u fiziološkim razmatranjima [lit. 6 ], da i promjena položaja tijela, ne mijenja samo parametre u modelu nego i sam model. Taj zaključak sasvim se podudara sa našim saznanjima iz termodinamike . Naime za promijenjeni položaj ljudskog tijela u vodi primijenjeni izrazi u članku bi se morali mijenjati .

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# AUTOMATSKI STABILNI SUSTAV ZA GAŠENJE POŽARA

# AUTOMATIC STABLE FIRE EXTINGUISHING SYSTEM

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## SAŽETAK

U prostoru limarske radionice sa lakirnicom predviđena je zaštita prostora stabilnim sustavom za gašenje požara s plinom FM-200. Ukratko je opisana cijela instalacija i to strojarski i njen elektro dio, te plan uzbunjivanja sa pripadnim crtežima i shemama. Uz to su dani proračuni sustava i prikaz opasnosti eksploatacije i prijedlozi za njihovo uklanjanje. Dan je i prikaz svih primijenjenih propisa i zakona koji se pritom primjenjuju.

**Ključne riječi:** lakirnica, stabilni sustav za gašenje požara, plin FM-200

## SUMMARY

The body and paint workshop provides for protection of the area by a stable fire extinguishing system with FM-200 gas. The entire installation and its mechanical and electrical parts are briefly described, as well as the alerting plan with the accompanying drawings and charts. In addition, the paper provides the system calculation and an overview of exploitation hazards, including the proposals for their removal. The paper also provides an overview of all the regulations and laws applied.

**Key words:** paint workshop, stable fire extinguishing system, FM-200 gas

## 1. UVOD

U prostoru limarske radionice sa lakirnicom predviđena je zaštita prostora stabilnim sustavom za gašenje požara. Spremnik s plinom kao navedeno rješenje postaviti će se izvan lakirne kabine i to na način da će se pričvrstiti obujmicom za betonski stup. Vrijeme gašenja je maksimalno 10 sekundi, a koncentracija FM-a 7 do 9%. Sustav protupožarne zaštite s plinom "FM-200" naročito je podoban radi slijedećeg:
- atmosfera plina u projektiranim koncentracijama (7 do 9 %) ne predstavlja opasnost za ljude, radi neznatne toksičnosti medija,
- velika efikasnost medija kojim se požar gasi,
- velika brzina djelovanja,
- minimalno smanjenje plina unutar prostorija,
- dobro miješanje plina sa zrakom bez rizika za raslojavanjem,
- plin nije korozivan, ne provodi struju i ne izaziva hladne šokove na elektronici.

## 2. OPIS UREĐAJA

### 2.1. Izbor zaštite

2.1.1. Plin FM-200

Plin za gašenje, pod komercijalnim nazivom FM-200 je kemijski spoj za gašenje požara, sustava heptafluorpropan, a u standardima NFPA 2001 pod nazivom HFC-227ea (kemijska formula: $CF_3CHFCF_3$). Gasi požare tako što inhibira reakciju između gorivog materijala i kisika. Kao sigurno i efikasno sredstvo upotrebljava se kod gašenja požara klase A (krutih materijala), klase B (zapaljive tekućine) i klase E (električne instalacije).
Kod vrednovanja na osnovu težine, tj. efikasnosti, danas je FM-200 jedna od efikasnih zamjena za halon 1301, koji je Montrealskim sporazumom zabranjen od 31.12.1995., iz ekološkog razloga, zbog utjecaja na proširenje ozonskih rupa.

2.1.2. Instalacija s plinom FM-200

Za gašenje požara plinom FM-200 u komori za lakiranje predviđen je jedan spremnik s plinom. Spremnik s plinom FM-200 za automatsko gašenje požara plinom FM-200 potrebno je postaviti izvan lakirne kabine, u sjeverozapadnom dijelu prostora lakirnice, pričvršćen obujmicom za betonski stup.
Na spremniku se nalazi automatski ventil, sa sigurnosnim ventilom, manometrom, presostatom, električnim okidačem za daljinsko aktiviranje i okidačem za mehaničko (ručno) aktiviranje. S ventila spremnika, s plinom FM-200, razvodi se cjevovod. Na kraju cjevovoda nalazi se mlaznica za FM-200. Spremnik s plinom čvrsto je obujmicom pričvršćen za zid.

### 2.2. Sastavni dijelovi instalacije

2.2.1. Strojarski dio instalacije

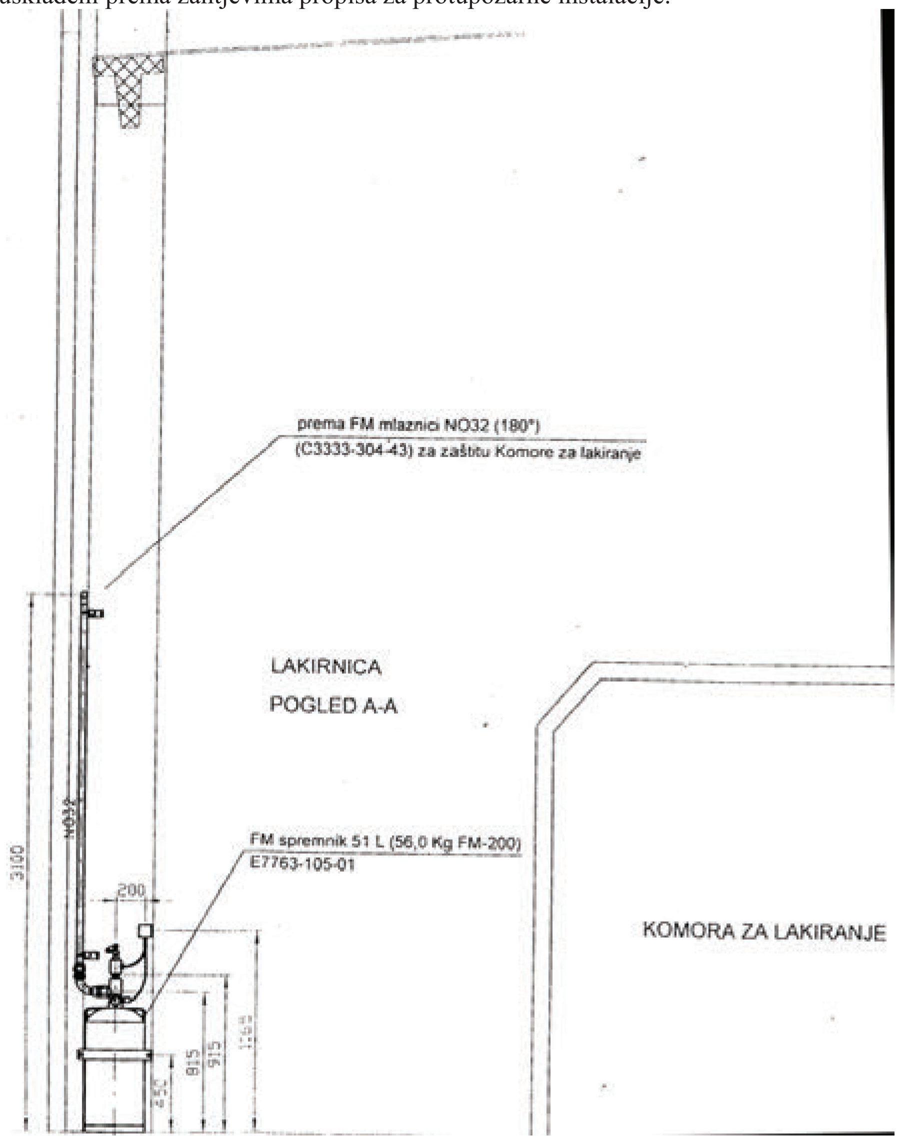
Instalacija za gašenje sastoji se od slijedećih osnovnih dijelova i uređaja:
- spremnik s plinom FM-200,
- cjevovoda s mlaznicama (tip 180°).

Aktiviranje instalacije može se provesti na jedan od slijedećih načina:
a) automatski – instalacija za gašenje aktivira se pomoću optičkih javljača požara smještenih u dvije zone,
b) poluautomatski – instalacija za gašenje aktivira se pomoću ručnog javljača ( na principu "razbi staklo") koji se nalazi na samoj centrali za gašenje.
c) ručno – instalacija za gašenje aktivira se vađenjem (povlačenjem) osigurača i pritiskom na polugu ručnog prekidača na ventilu spremnika FM-200.

Tlak u spremniku je 24,8 bar kod 20°C. Svaki spremnik je ispitan na ispitni tlak od 46 bar. Sigurnosni ventil se otvara na 55 bar.
Kontrola tlaka u spremniku "FM-200" može se vršiti na dva načina:
- vizualno na manometru,
- daljinskom signalizacijom preko presostata, koji na centrali signalizira "KVAR", ako tlak u spremniku padne ispod 17 bara.

Vrata prostorije u kojoj je predviđena zaštita gašenjem plinom FM-200 moraju biti stalno zatvorena, odnosno moraju imati na sebi ugrađen uređaj za samozatvaranje.
Cjevovod FM-200 potrebno je izvesti iz čeličnih, crnih, šavnih cijevi (prema DIN 2440). Prije ugradnje, cijevi je potrebno očistiti iznutra. Nakon ugradnje cjevovoda poslije tlačne probe, a

prije montaže mlaznica, cjevovod je potrebno osušiti i propuhati stlačenim zrakom, dušikom ili ugljičnim dioksidom. Cjevovod će se spojiti zavarivanjem. Isti je potrebno ispitati zrakom na 10 bara, a sve prema standardu NFPA 2001 (točka 4-7.2.2.12) u vremenu od 10 minuta. U periodu ispitivanja dopušteno je sniženje ispitnog tlaka za 20%. Dizanje tlaka vrši se u koracima po 3,5 bar i uz zadržavanje od 1 minute.
FM-200 spremnik potrebno je ličiti nitro lakom ili sivom bojom i 100 mm širokim crvenim prstenom oko spremnika, dok će se cjevovod ličiti temeljnom i završnom crvenom lak bojom RAL 3000.
Za konzoliranje cjevovoda potrebno je koristiti standardizirana učvršćenja (čvrste točke). Presjeci nosača cjevovoda i sidrenih vijaka, te međusobne udaljenosti oslonaca, proračunati su i usklađeni prema zahtjevima propisa za protupožarne instalacije.




**Slika 2.1. Dispozicija FM-200 spremnika u komori za lakiranje - pogled**

2.2.2. Elektro dio instalacije

Elektro dio stabilnog sustava za gašenje požara plinom FM-200 sastoj ise od slijedećih elemenata automatske zaštite plinom FM-200: centrala za gašenje plinom FM-200, tipa Prescient II, konvencionalnih termomaksimalnih javljača u "S" izvedbi, sirena sa bljeskalicom u "S" izvedbi, samosigurnosnih uređaja, razvodnih kutija za spremnike plina FM-200 i spremnika s plinom FM-200 na kojem se nalaze elektromagnetski okidači i presostati.
Samosigurnosni uređaji (SSU) koji osim za napajanje i nadziranje stanja javljača u "S" izvedbi", služe i da galvanski odvoje "S" zonu. Investitor je pritom dužan uzemljiti SSU unutar "S" zone.
Centrala za gašenje se postavlja iznad boce za gašenje pored lakirnice. Na centralu su spojena dva termomaksimalna javljača požara u "S" izvedbi i to u svakoj zoni po jedan javljač. Sirene sa bljeskalicom u "S" izvedbi se postavljaju iznad ulaznih vrata u štićeni prostor, te iznad istih vrata unutar štićenog prostora. Presostat i aktuator sa spremnika sa plinom spajaju se na centralu za gašenje požara preko razvodne kutije za FM-boce. Boce se uzemljuju preko spomenutih razvodnih kutija na uzemljenje centrale za gašenje.
Sa centrale za gašenje na mjesto 24-satnog dežurstva na porti u prizemlju zgrade, koja se naslanja na zapadno pročelje lakirnice prosljeđuju se tri signala:

- signal alarma 1. stupnja – bilo koji optički javljač požara u alarmu u jednoj od dviju zona,
- signal alarma 2. stupnja – bilo koji optički javljači požara u obje zone ili ručni javljač požara u sklopu centrale za gašenje,
- kvar na sustavu za gašenje – nestanak mrežnog napajanja, prazne baterije za rezervno napajanje, nizak tlak u boci, kvar javljača itd.

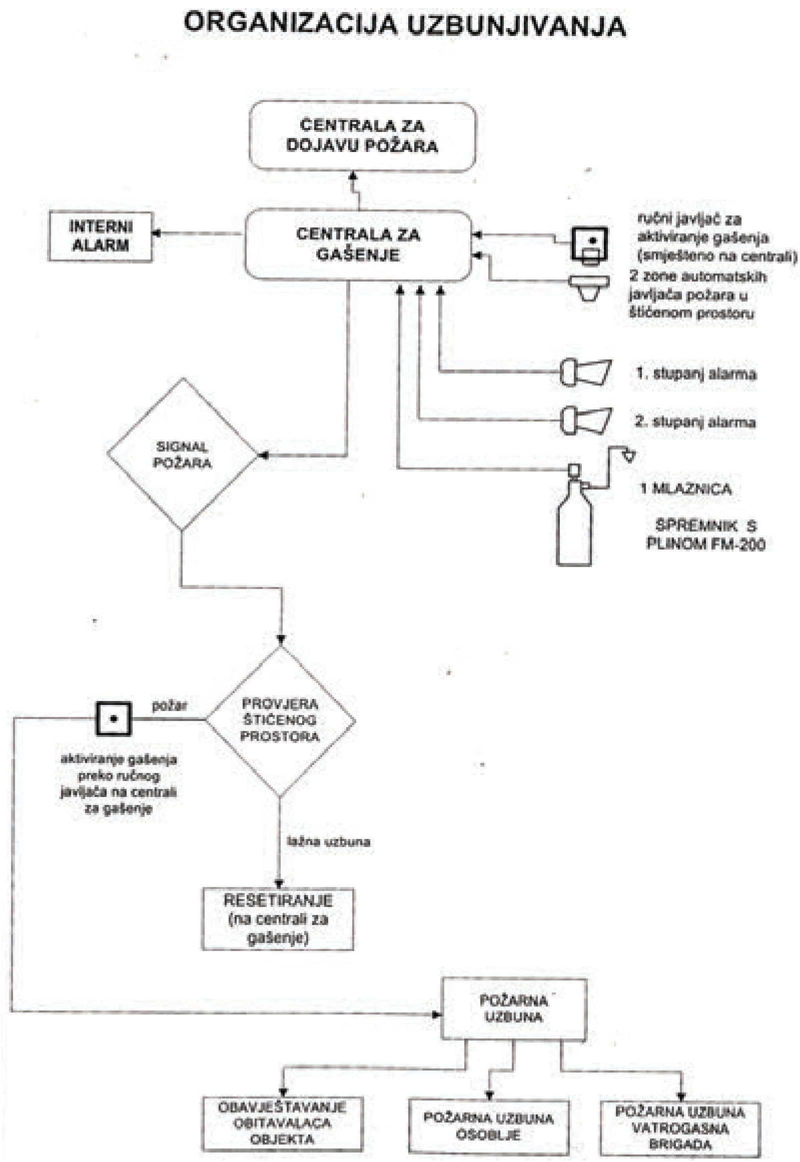
Sve informacije o stanju sustava za gašenje mogu se vidjeti na centrali za gašenje. Centrala za gašenje ima mogućnost automatskog i poluautomatskog režima rada.
U automatskom režimu rada da bi došlo do automatskog uključenja gašenja, potrebno je da su obje zone u štićenom prostoru u alarmu. Dvozonska ovisnost se određuje preko centrale za gašenje.
U poluautomatskom (ručnom) režimu rada da bi došlo do automatskog uključenja gašenja potrebno je aktivirati ručni javljač požara u sklopu centrale za gašenje.
U automatskom režimu rada u slučaju alarma 1. stupnja (jedna zona u alarmu) uključuje se sirena sa bljeskalicom S1 u "S" izvedbi u štićenom prostoru, što predstavlja upozorenje osobama u prostoru. Alarm 2. stupnja se uključuje po ulasku i druge zone u alarm, te se isključuje sirena sa bljeskalicom S1 u štićenom prostoru, a uključuje se sirena sa bljeskalicom S2 u "S" izvedbi ispred pripadajućeg prostora, koja upozorava da će doći do aktiviranja gašenja nakon programiranog kašnjenja. Nakon toga, ako nije došlo do zaustavljanja gašenja na centrali za gašenje, dolazi do ispucavanja plina FM-200.
U poluautomatskom režimu rada u slučaju aktiviranja ručnog javljača u sklopu centrale uključuje se alarm 2. stupnja, te se uključuje sirena sa bljeskalicom S2 ispred pripadajućeg prostora, koja upozorava da će doći do aktiviranja gašenja nakon programiranog kašnjenja. Nakon toga, ukoliko nije došlo do zaustavljanja gašenja na centrali za gašenje, dolazi isto tako do ispucavanja plina FM-200.
Ako je sustav u poluautomatskom režimu rada, a u slučaju ulaska zona 1 i 2 u alarm, neće se aktivirati gašenje, već će sustav ostati kod alarma 1. stupnja (uključena sirena sa bljeskalicom S2 u štićenom prostoru). Do gašenja će doći samo ako se aktivira ručni javljač postavljen na prednjoj ploči centrale za gašenje.
Elektro dio stabilnog sustava za gašenje plinom FM-200 sastoji se od:
a) centrale za gašenje – služi za upravljanje gašenjem i to putem dvozonske ovisnosti ili ručnim okidanjem putem ručnog javljača tipkalom za aktiviranje. Ima ukupno 4 zone, i

kompletni sustav za nadzor i okidanje presostata na ventilu za gašenje, nadzor niskog tlaka i nadzor ispucavanja plina preko tlačne sklopke.
b) termomaksimalnih javljača požara u "S" izvedbi – predviđeni su u slučaju automatskog aktiviranja požara.Koriste se u predmetnom prostoru zbog ambijentalnih uvjeta (radne temperature, ventilacije). Hermetički su zatvoreni i otporni na agresivnu atmosferu. Porast okolne temperature iznad fiksne temperature prorade javljača uzrokuje da bimetalni disk zatvori kontakt, te se uslijed promjene otpora linije registrira alarm na centrali. Bimetalni disk je električki i termički izoliran fenolom ili keramikom, a porast temperature registrira se prijelazom iz konkavnog u konveksno stanje. Temperatura odaziva u ovom slučaju je 107 °C. Spaja se u samosigurni strujni krug.
c) sirena sa bljeskalicom u "S" izvedbi – izvedene su kao "neprodorni oklop", te su kao takve pogodne za montažu u "S" zoni, s napajanjem kojeg dobiva preko centrale za gašenje. Snaga zvučnog signala je 120 dB.
d) samosigurnosnih uređaja – samosigurnost je vrsta protueksplozijske zaštite, koja ograničavanjem struje i napona osigurava da iskrenje na tako zaštićenom uređaju neće izazvati paljenje čak ni u slučaju kada se eksplozivna smjesa pojavi u najzapaljivijoj koncentraciji, a na uređaju nastane iskrenje direktno u toj smjesi. To znači da ni iskrenja ni oštećenja kabela u samosigurnosnom krugu ne može izazvati paljenje eksplozivne smjese. Uređaj služi da galvanski odvoji zonu opasnosti od ostataka vatrodojavnog sustava, te da se detektori u opasnoj zoni mogu priključiti na vatrodojavnu centralu.
e) razvodne kutije za FM boce
f) prijenosnih puteva, tj. kabela koji povezuju elemente sustava
Mrežno napajanje je osigurano preko automatskog osigurača koji se nalazi u razdjelniku električne energije u prostoriji IT sobe.

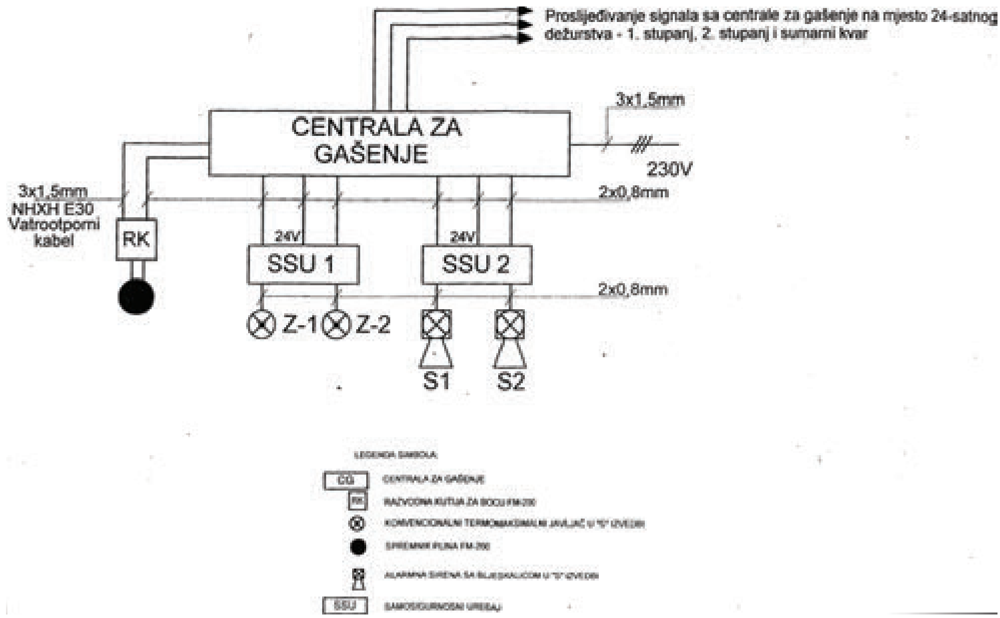
**2.3. Plan uzbunjivanja**

U slučaju požara (alarma s jednog od optičkih javljača) u štićenoj prostoriji, uključuje se svjetlosno zvučni signalizator unutar štićenog prostora, te se signal alarma 1. stupnja prosljeđuje na sustav vatrodojave, tj. na centralu za dojavu požara.
Nakon dojave požara na centralu za dojavu požara, ovlaštena osoba ide utvrditi požar u prostoru štićenom plinom FM-200. Ukoliko utvrdi požar u štićenom prostoru, a sustav još nije ušao u alarm 2. stupnja, odmah aktivira gašenje preko ručnog javljača smještenog na centrali za gašenje. Nakon toga pristupa upozoravanju obitavalaca objekta, obavještava prisutno osoblje, te obavještava vatrogasnu postrojbu. Nakon ispucavanja FM-200 plina, potrebno je provjetriti prostor.
Ukoliko je signal lažan, sustav se resetira na centrali za gašenje.
Organizacija uzbunjivanja prikazana je na slici 2.2, dok je na slici 2.3 prikazana blok shema sustava FM-200.

Nakon ugradnje protupožarne instalacije potrebno je provesti funkcionalno ispitivanje bez aktiviranja FM-200 spremnika. Uz to potrebno je provjeriti rad svih komandi i signalizacije.




Slika 2.2. Prikaz organizacije uzbunjivanja




**Slika 2.3. Blok shema sustava FM-200**

## 3. PRORAČUNI

### 3.1. Proračun potrebne količine plina FM-200, izbor FM uređaja i mlaznica

Potrebna količina FM-200 određuje se u ovisnosti o volumenu štićenog prostora i potrebne koncentracije za gašenje. Presjek cjevovoda i površina otvora na mlaznicama određuje se na temelju protoka i pada tlaka u cjevovodu i armaturama.

3.1.1. Proračun potrebne količine FM-200

$Q = V \cdot k$ [kg]

gdje je:
Q – potrebna količina FM-200 [kg]
V – volumen štićenog prostora [m$^3$]
k – specifična toplina [kg/m$^3$]

6.1.2. Proračun pada tlaka

$p_u = p_c + p_g - p_k$ [bar]

gdje je:
$p_u$ – ukupni pad tlaka [bar]
$p_c$ – pad tlaka zbog trenja u cjevovodima [bar]
$p_g$ – pad tlaka zbog razlike u geodetskim visinama [bar]
$p_k$ – porast tlaka zbog kinetičke energije sredstava na izlazu [bar]

6.1.3. Proračun mlaznica

$$A = \frac{q}{q_0}\ [\mathrm{mm}^2]$$

gdje je:
A – presjek otvora na mlaznici [mm$^2$]
q – protok FM-200 na mlaznici [kg/s]
$q_0$ – specifični protok [kg/s*mm$^2$]

**6.2. Proračun autonomije centrale za gašenje**

Potrebno je predvidjeti autonomiju vatrodojavnog sustava od 30 sati ($t_n$ = 30 h) u slučaju nestanka glavnog napajanja u normalnom režimu rada i jedan sat rada u alarmu ($t_{al}$ = 1h), te u skladu s tim rezervno napajanje baterijama. Potrošnja sustava u normalnom stanju se zanemarivo razlikuje od potrošnje centrale, dok se za potrošnju kod alarma 1. i 2. stupnja mora uzeti u obzir i potrošnja sirena.

$$C = \frac{I_n \cdot t_n + I_{al} \cdot t_{al}}{\eta}$$

gdje su:
$I_n$ – ukupna potrošnja u normalnom stanju [A]
$I_{at}$ – ukupna potrošnja u alarmnom stanju [A]
η – koeficjent napunjenosti baterije

**4. OPASNOSTI EKSPLOATACIJE I PRIJEDLOZI ZA NJIHOVO UKLANJANJE KOD INSTALACIJA ZA GAŠENJE POŽARA S PLINOM FM-200**

**4.1. Eksploatacija cilindričnog spremnika za FM-200**

Spremnik za FM-200, ispitan je hladnim vodenim tlakom na 46 bar. Tlak dušika u boci sa FM-200 kod 21°C iznosi 24,81 bar. Za slučaj prekomjernog porasta tlaka ugrađen je ventil sigurnosti koji se otvara kod tlaka od 55 bar, te je na taj način otklonjena opasnost.

**4.2. Eksplozija cjevovoda**

Cjevovod dolazi pod tlak samo za vrijeme aktiviranja uređaja. Za vrijeme eksploatacije plina FM-200, tlak u cjevovodu je uglavnom manji od 24 bar. Da bi se spriječila opasnost od eksplozije, cjevovod se ispituje zrakom na 10 bar, u skladu s NFPA 2001/96 (točka 4-7.2.2.12), u vremenu od 10 minuta.

**4.3. Ugradnja protupožarnih instalacija i primjena zaštite na radu**

Prilikom ugradnje protupožarnih instalacija primjenjivati će se propisana pravila zaštite na radu, Pravilnik o zaštiti na radu izvođača radova, te eventualno upute od strane investitora. Prilikom izvođenja radova radnici su dužni koristiti osobna zaštitna sredstva predviđena za pojedine radove Elaboratom, odnosno Pravilnikom o zaštiti na radu izvođača.

4.3.1. Rukovanje spremnicima s plinom FM-200

Transport spremnika od mjesta istovara do mjesta ugradnje, vršiti će se pomoću, za tu svrhu predviđenih kolica i naprava.Za vrijeme rukovanja spremnicima na ventile FM spremnika postaviti zaštitne kape, kako bi se spriječilo oštećenje ventila i nehotično pražnjenje boca.

4.3.2. Montaža cjevovoda

**4.4. Opasnosti po zdravlje prisutnih**

Plin FM-200 u projektiranim koncentracija 7 do 9%, nije opasan po zdravlje, a li se ipak preporuča napuštanje prostorija u kojima je došlo do ispucavanja plina FM-200.

**4.5. Atesti**

Izvođač je dužan pribaviti ateste za ugrađenu opremu.

**4.6. Pregled instalacija**

Preglede instalacija treba vršiti barem jednom godišnje i to: od strane ovlaštene organizacije pribaviti atest o ispravnom funkcioniranju instalacija najmanje jednom godišnje.

**NAZIVLJE**

Latinične oznake:

A – presjek otvora na mlaznici [$mm^2$]
C – potrošnja sustava za gašenje požara [Ah]
$I_n$ – ukupna potrošnja u normalnom stanju [A]
$I_{at}$ – ukupna potrošnja u alarmnom stanju [A]
k – specifična toplina [$kg/m^3$]
$p_c$ – pad tlaka zbog trenja u cjevovodima [bar]
$p_g$ – pad tlaka zbog razlike u geodetskim visinama [bar]
$p_k$ – porast tlaka zbog kinetičke energije sredstava na izlazu [bar]
$p_u$ – ukupni pad tlaka [bar]
Q – potrebna količina FM-200 [kg]
q – protok FM-200 na mlaznici [kg/s]
$q_0$ – specifični protok [$kg/s*mm^2$]
V – volumen štićenog prostora [$m^3$]
$t_{al}$ – vrijeme autonomije sustava u alarmnom stanju [h]
$t_n$ – vrijeme autonomije sustava u normalnom stanju [h]

Grčke oznake:

$\eta$ – koeficijent napunjenosti baterije

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3. Primjenjeni propisi i zakoni:
- Zakon o gradnji (NN RH br. 175/03 i 100/04),
- Zakon o normizaciji (NN RH br. 163/03),
- Zakon o zaštiti od požara (NN RH br. 58/93 i 33/05),
- Pravilnik o izradi procjene ugroženosti od požara i tehnoloških eksplozija (NN RH br. 55/94),
- Pravilnik o sadržaju plana zaštite od požara i tehnoloških eksplozija (NN RH br. 35/94),
- Ispravak pravilnika o sadržaju plana zaštite od požara i tehnoloških eksplozija (NN RH br. 55/94),
- Pravilnik o uvjetima za ispitivanje uvezenih uređaja za gašenje požara (NN RH br. 75/94),
- Pravilnik o uvjetima za obavljenje ispitivanja stabilnih sustava za dojavu i gašenje poažra (NN RH br. 67/96),
- Pravilnik o sustavima za dojavu požara (NN RH br. 56/99),
- Pravilnik o tehničkim noramtivima za zaštitu elektroenergetskih postrojenja i uređaja od požara (Sl. list br. 74/90),
- HRN EN 54,
- DIN norme,
- Zakon o zaštiti na radu (NN RH br. 59/96 i 114/03),
- Pravilnik o poslovima s posebnim uvjetima rada (NN br. 7/84),
- Pravilnik o načinu ispitivanja određenih sredstava rada i radne okoline, te sadržaju, obliku i načinu izdavanja isprava (NN br. 52/84),
- Pravilnik o zaštiti na radu pri korištenju električne energije (NN br. 9/87),
- Pravilnik o sadržaju plana uređenja privremenih i zajedničkih privremenih radilišta (NN br. 45/84),
- Pravilnik o tehničkim noramtivima za električne instalacije niskog napona (Sl. List br. 53/88),
- NACIONAL ELECTRICAL CODE-NEC.

# ISPITIVANJE PROTUUDARNOG VENTILA ZA SKLONIŠTA

# TESTING OF A BLAST VALVE FOR SHELTERS

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**SAŽETAK**

Ovim radom dan je prikaz ispitivanja protuudarnog ventila za skloništa prema navedenim pravilnicima. Uz kratki prikaz elemenata sistama za provjetravanje skloništa i opis tehničkih karakteristika ispitivanog ventila, dan je cijeli tok ispitivanja. Od opisa i sheme mjerne linije, opisa samog mjerenja i tabličnog prikaza rezultata ispitivanja. Sve je popraćeno i slikama, te relevantnim shemama.

**Ključne riječi:** sklonište, protuudarni ventil, ispitivanje

**SUMMARY**

This paper provides an overview of a test performed on a blast valve for shelters according to the listed policies. In addition to the brief overview of the elements within the shelter ventilation system and description of the technical characteristics for the tested valve, the paper provides the course of the entire testing. This includes a description and chart of the measuring line, description of the test itself, and a tabular display of the test results. All of this is accompanied with pictures and relevant charts.

**Key words:** shelter, blast valve, testing

## 1. UVOD

### 1.1. Općenito

Kao sklonište za zaštitu stanovništva od ratnih djelovanja smatra se građevinski objekt, odnosno dio objekta koji se mora sastojati od zatvorenih i funkcionalno povezanih prostorija koje omogućavaju zaštitu od mehaničkog, toplinskog, radijacijskog i kemijskog djelovanja oružja.

### 1.2. Vrste skloništa

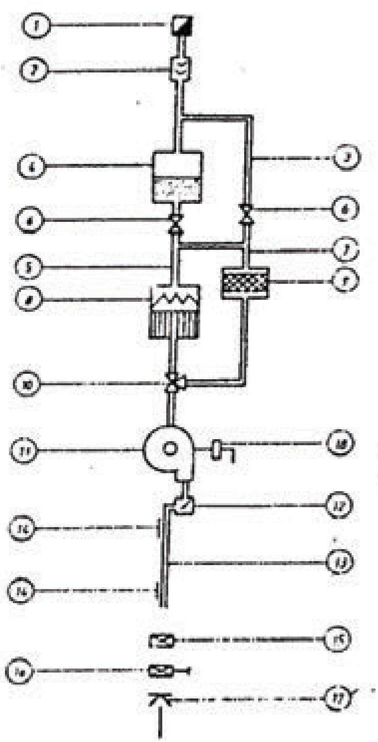
Sklonište dopunske zaštite mora imati:
1. Područje zaštite od 50 kPa pretlaka,
2. Funkcinalno rješenje prostorije opremljene za višesatno zadržavanje do 50 osoba.

Sklonište osnovne zaštite mora imati:
1. Područje zaštite od 100 kPa do 300 kPa pretlaka,
2. Funkcinalno rješenje prostorije opremljene za sedmodnevni neprekidni boravak do 300 osoba.

Sklonište pojačane zaštite mora imati:
1. Područje zaštite od direktnog pogotka avionske bombe kalibra najmanj 300 kg,
2. Funkcinalno rješenje prostorije opremljene za četnaestodnevni neprekidni boravak do 200 osoba.

### 1.3. Raspored elemenata sistema za provjertavanje skloništa

Sistem za ventilaciju skloništa treba sadržavati:



**Slika 1.1. Elementi sistema za provjetravanje skloništa**

Legenda:
1. Usisni otvor
2. Protuudarni ventil
3. Cjevovod za normalno provjetravanje
4. Pješčani predfilter

5. Cjevovod za zaštitno provjetravanje
6. Brzozatvarajući ventil
7. Obilazni cjevovod
8. Filter za kolektivnu zaštitu
9. Filter za grubu prašinu
10. Ventil za promjenu načina provjetravanja
11. Centrifugalni ventilator
12. Mjerač protoka
13. Cjevovod za raspodjelu uzduha
14. Anemostati
15. Ventil za regulaciju pretlaka
16. Uasun za uzduh
17. Protuudarni ventil
18. Mjerač pretlaka

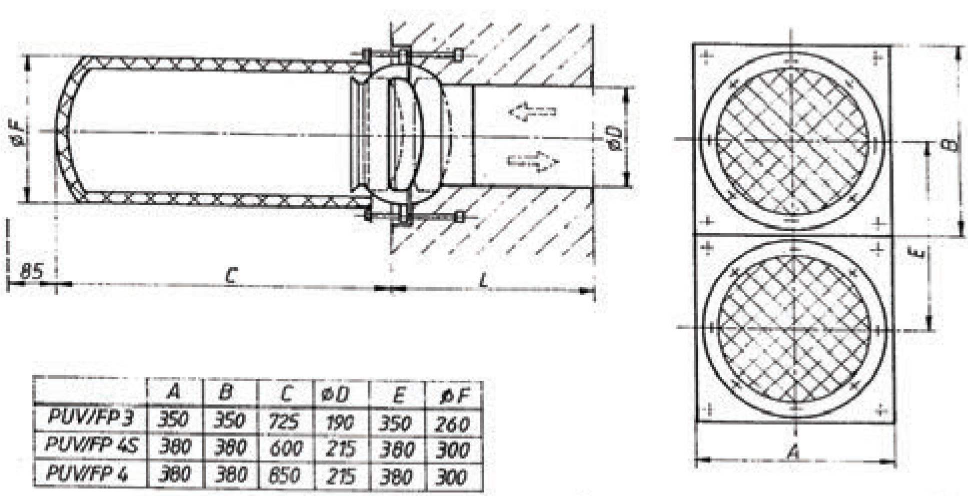
### 1.4. Tehničke karakteristike ispitivanog ventila

Ispitivani protuudarni ventil je proizvod firme ANDAIR AG , oznaku PUV . Namijenjen je da štiti otvor za dovod zraka od prodora zračnog udarnog vala, prouzročenog konvencionalnom ili nuklearnom eksplozijom, koji može naškoditi ljudima ili oštetiti uređaje i opremu.

Sastoji se od :

- zaštitni poklopac,
- pričvrsni prsten,
- protuudarni ventil,
- matice M12,
- filter za prašinu.

Filter za prašinu štiti elemente ventilacije od kontaminiranih grubih čestica prašine. Protuudarni ventil s filterom za prašinu koristi se u skloništima za civilne i vojne namjene i u drugim civilnim i industrijskim građevinama kod kojih se traži povećana zaštita.
Tehničke karakteristike ispitivanog ventila su prikazane na slijedećoj slici:

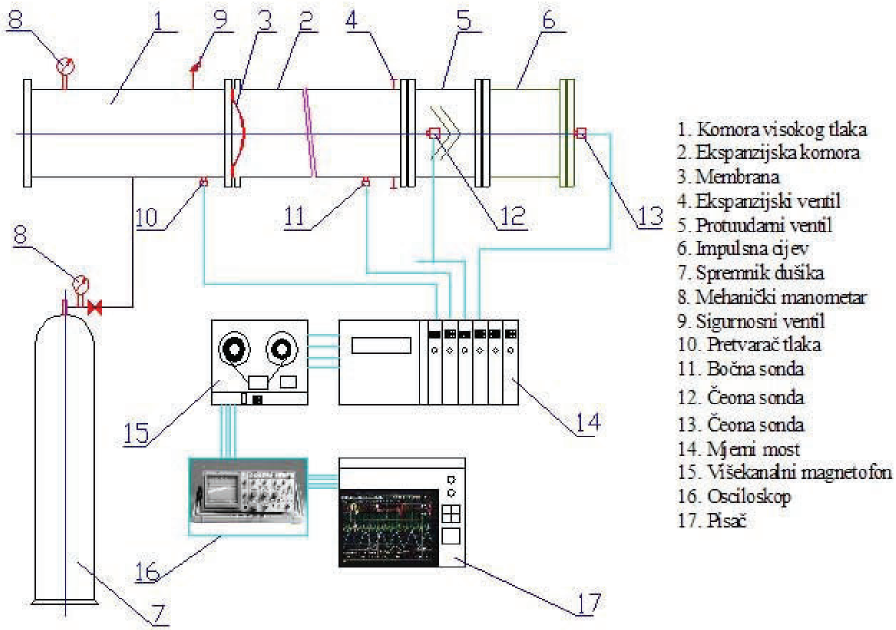



| | A | B | C | ØD | E | ØF |
|---|---|---|---|---|---|---|
| PUV/FP 3 | 350 | 350 | 725 | 190 | 350 | 260 |
| PUV/FP 4S | 380 | 380 | 600 | 215 | 380 | 300 |
| PUV/FP 4 | 380 | 380 | 850 | 215 | 380 | 300 |

**Slika 1.2. Tehničke karakteristike ispitivanog ventila**

## 2. ISPITIVANJE PROTUUDARNOG VENTILA

### 2.1. Opis i shema mjerne linije

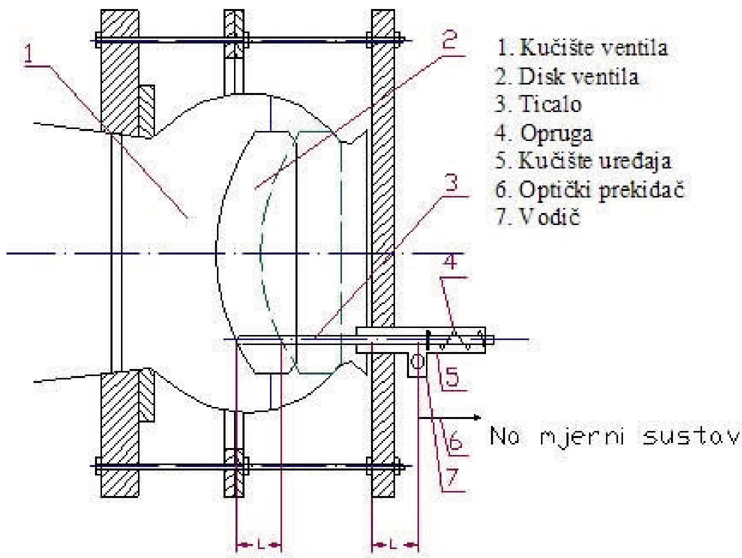



**Slika 2.1. Shema mjerne linije**

Na slici 2.1. prikazani su dijelovi mjerne linije.
Mjerenje značajki udarnog vala vrši se pomoću mjernih sondi na ekspanzijskoj cijevi i na ploči. Signal se sa mjernih sondi prenosi preko pretpojačala na višekanalni mjerni magnetofon *TI-8*, osciloskop *Philips* PM 3232 i pisač *Gould* ES 1000 koji su povezani u mjerni sustav.
Na slikama 2.2. i 2.3. prikazana je mjerna linija na Fakultetu strojarstva i brodogradnje u Zagrebu, dok je način ugradnje ispitivanog protutudarnog ventila na mjernu liniju.

Slika 2.2. Sklop tlačne i ekspanzijske komore bez ugrađenog protuudarnog ventila

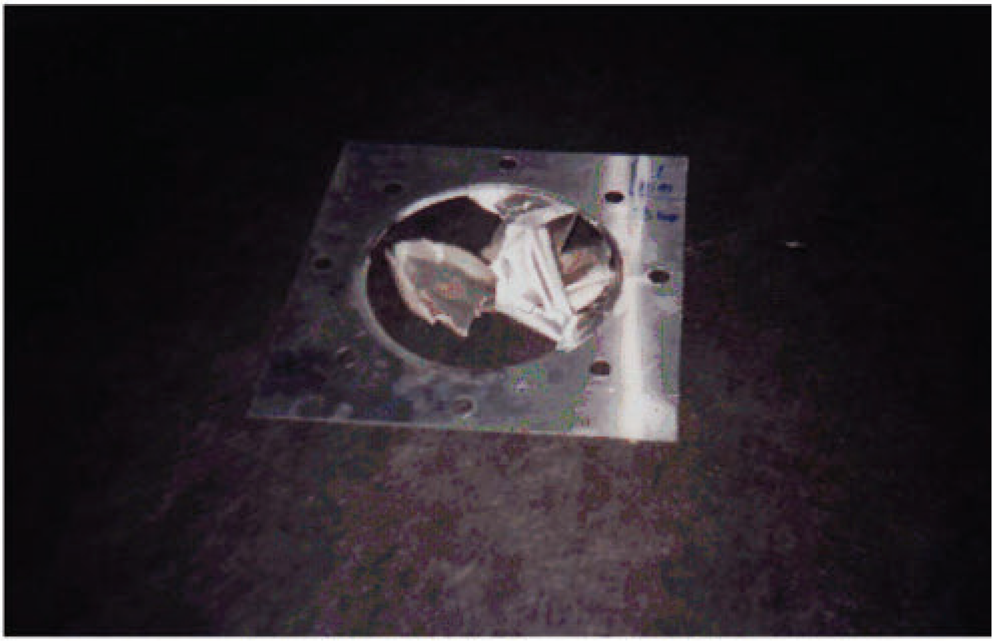
Slika 2.3. Detalj tlačne komore s kontrolnim manometrom, sigurnosnim ventilom i tlačnim ventilom



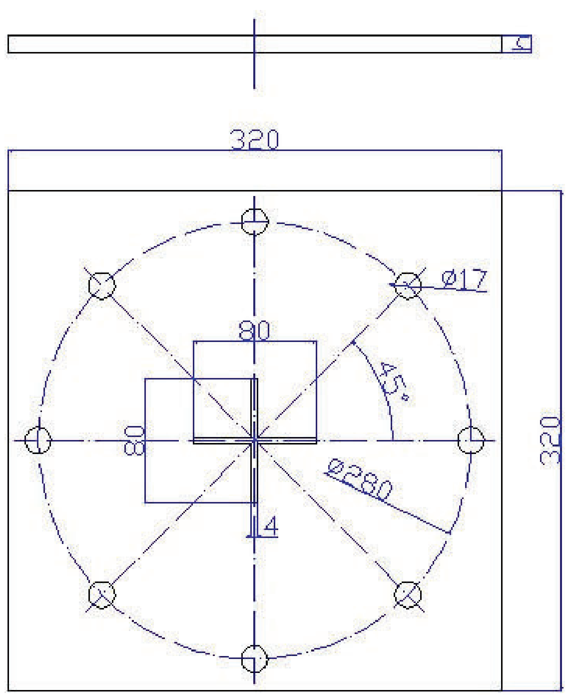

**Slika 2.4. Detalj ugradnje ispitivanog protuudarnog ventila**

## 2.2. Opis mjerenja

Na raspolaganju su bile ploče od 1, 2 i 3 mm debelog lima, koje su izrađene u radionicama na Fakultetu strojarstva i brodogradnje, Sveučilišta u Zagrebu.
Da bi se odredila točnost teorijske dobivene debljine ploče, bilo je potrebno praktično provjeriti rezultat i dobiti ponovljivost eksperimenta.
Ispitivana je prvo ploča debljine 1 mm. Ne uzimajući u obzir vrijeme porasta tlaka (u prvom mjerenju povećavali smo tlak konstantno približno 1.5 min do postizanja tlaka pucanja membrane) u tlačnoj komori linije, ploča je pukla pri tlaku od 12.5 bara, što je nešto više od proračuna.
U drugom mjerenju, uz usporavanje brzine porasta tlaka, ploča je pukla na tlaku od 13.1 bar.
Na slici 2.5. vidljiv je izgled ploče debljine 1 mm poslije pucanja. Bitno je napomenuti da ovakav rezultat pucanja nije zadovoljavajući, jer se dogodilo kidanje ploče po unutarnjem promjeru ekspanzijske cijevi linije, a otkinuto središte postaje opasno za sklop protuudarnog ventila, jer bi velikom brzinom udarilo u protuudarni ventil i oštetilo ga.
Da bi se izbjeglo razlijetanje ploče uzete su ploče većih debljina lima od kritične, na koje su se urezale inicijalna oslabljenja u obliku križa (vidi sliku 2.6.), da se dobije kontrolirano pucanje ploče po oslabljenju, a ne po obodu. Na ploči je urezano oslabljenje u obliku križa dubine 0.1 debljine ploče.



Slika 2.5. Izgled ploče debljine 1 mm nakon pucanja




Slika 2.6. Radionički crtež ploče sa oslabljenjem

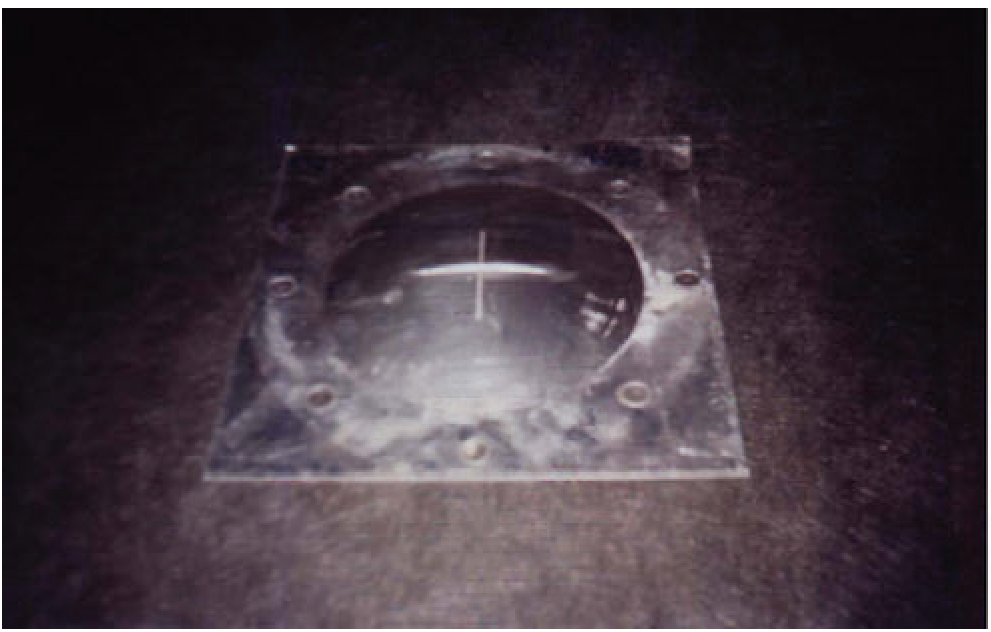
## 2.3. Prikaz rezultata ispitivanja

U tablici 1. dan je prikaz rezultata laboratorijskih ispitivanja.

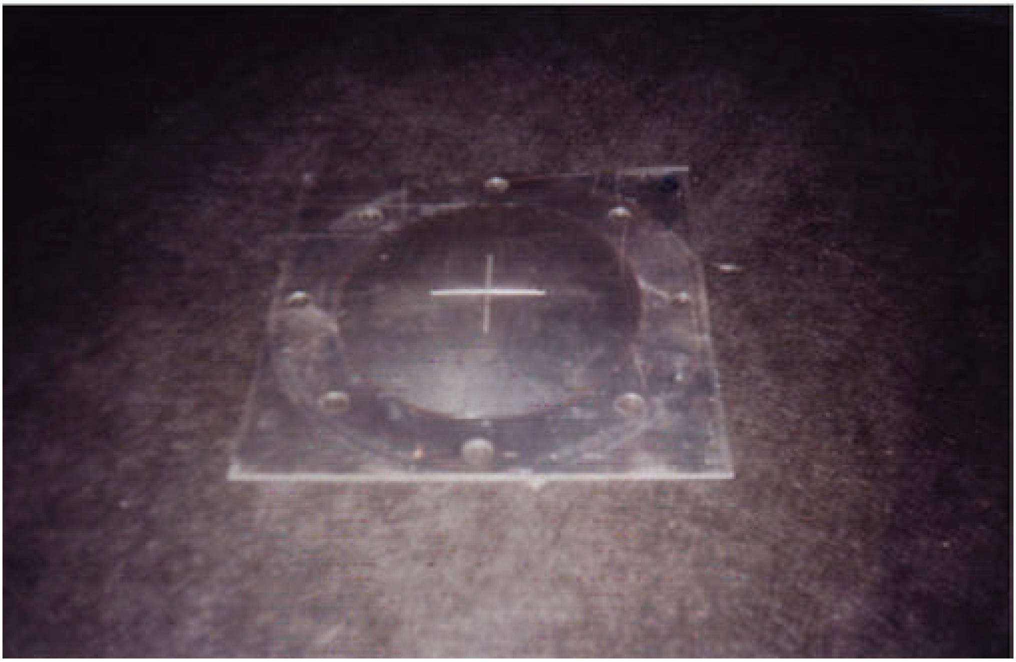
**Tablica 1. Prikaz rezultata ispitivanja**

| Debljina ploče h [mm ] | Tlak pucanja ploče bez oslabljenja [bar] | | | Tlak pucanja ploče sa oslabljenjem dubine *0.1h* [bar] | | |
|---|---|---|---|---|---|---|
| | 1. mjerenje | 2. mjerenje | 3. mjerenje | 1. mjerenje | 2. mjerenje | 3. mjerenje |
| 1 | 12.5 | 13.1 | 9 | 4.5 | 5 | x |
| 2 | x | x | x | 9 | 8.5 | x |
| 3 | x | x | x | >25* | | |

**Legenda:**
x...nije mjereno
*...nije moguće mjeriti iznad te vrijednosti, jer je mjerna linija atestirana na tlak do 24 Bara.



**Slika 2.7. Izgled oslabljene membrane debljine 2 mm sa oslabljenjem 0.2 mm poslije prsnuća**



**Slika 2.8. Izgled oslabljene membrane debljine 3 mm sa oslabljenjem 0.3 mm**

## 3. ZAKLJUČAK

Cilj ovoga ispitivanja je bilo upoznavanje sa normama za ispitivanje protuudarnog ventila prema mehaničkom udarnom opterećenju radi postizanja karakteristika, koje su u granici tolerancija propisanih pravilnikom. Protuudarni ventil je predviđen kao jedan od elemenata u sustavu ventilacije skloništa. Ispitivanje je provođeno prema potrebnim pravilnicima i literaturi. Svi rezultati ispitivanja su prezentirani tablično.
U laboratorijskom ispitivanju smo služeći se iskustvenim metodama modelirali križna oslabljenja na predimenzioniranim pločama u svrhu pucanja po oslabljenju, a ne po obodu ploče, što bi rezultiralo mehaničkim oštećenjem protuudarnog ventila. Mijenjanjem dubine oslabljenja željeli smo postići rasprskavanje pri potrebnom tlaku, određenom geometrijom mjerne linije.
Sa stanovištva ponovljivosti pojedine mjerne točke, ispitivanje je provedeno zadovoljavajuće.

## NAZIVLJE

Latinične oznake:

D – promjer ploče $[mm]$,
h – debljina stijenke $[mm]$,
p – tlak $[bar]$
R – polumjer ploče $[mm]$.

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7. ...Atoming, katalog, 1998.

# DOBRA KLIMA

*Prezentacija dr.sc. Fedor Kritovac za skup INTERKLIMA, Zagreb travanj 2007*

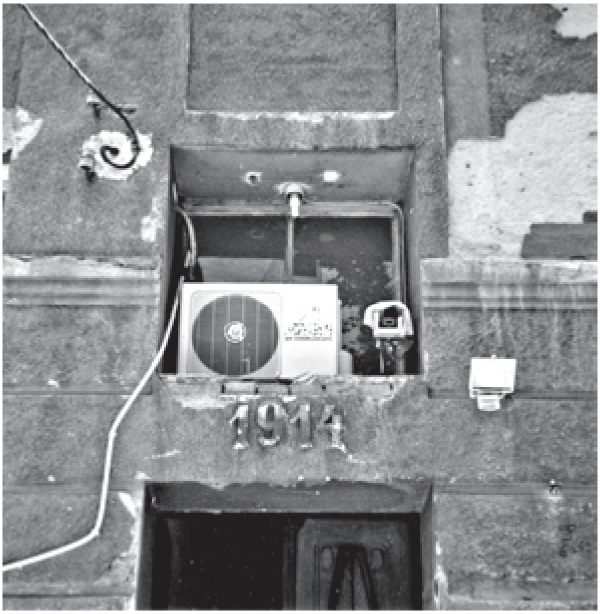
Energetski pokazatelji ne samo pokazuju već i upozoravaju ; mijenjaju se ubrzano i mikroklimatski uvjeti boravljenja (u stanu, na poslu...) , pogotovu sezonski u toplije vrijeme. Klima uređaji za individualni komfor, postaju ne samo predmet izbora subjektivnih i objektivnih sklonosti i potreba i preferencija već i statusa.

Kao dodatni instalacijski elementi prepušteni su u postavi vještini i nahođenju naručitelja i izvoditelja,. Kao takvi stvaraju nedoumice arhitektima , sanitarcima, komunalcima i čuvarima baštine, k tome u svojevrsnom regulativnom vakuumu.

Ove instalacije su - vidimo - na neki su način proturječne u postizanju kvalitete građevnog i korisničkog prostora.

Bez obzira kako se gleda na pridošle a i neočekivane promjene navika i izgleda pojedinačnih objekata, dijelova i detalja građevina i kako se sa svime time postupa, - klima- uređaji ( ili popularno već, "klime" ) iskazuju se u urbanom tkivu grada kao vrlo slikovite pa i čudnovate formacije u svom bližem kontekstu (na pročelju, u vežama, prolazima, nad lokalima, ,na krovovima itd.) tvoreći gotovo nadrealne prizore usred dana, sumraka ili noći.

Klima uređaji nađu se tako usamljeni , pridodavani, povezani u grupama sa svojim karakterističnim kućištima koja se- bez obzira na tipove i varijante - prepoznaju kao karakteristične gotovo geometrijske tvorevine bliske kvadru, kugli, kružnicama , pravokutnicima a koje se ne samo fizički već i vizualno povezuju s elementima ahitekture, osvjetljenja i vegetacije i druge opreme u nove vizualno-skulpturalne cjeline .




U okruženju se tih instalacija katkada tako nađe fasadna ili neka druga kućna vegetacija čime se nepredviđeno simbolički naglasila funkcija KLIME. A ponekad, kao što smo uočili, mjesta montaže radi pritoka zraka svojim izborom i stanjem (nadsvođeni prolazi i slično) kao da demantiraju namjenu. Kad se nađu interpolirani među okna, ispod balkona ili na njima, u fragmente pročelja historicističke ili moderne arhitekture ti elementi i sami poprimaju određenja neke nove urbane plastike (kao što je to i s nekim drugim instalacijama : tv sat antene, antene mobilne telefonije i t sl.)

U susjedstvima KLIMA s drugim predmetima, bojama, likovima i znakovima, - nerijetko i zbog pomanjkanja održavanja interijera i eksterijera takve scene postaju apsurdne, tajnovite ili komične. Tako, da se opravdava ovdje odabran metaforički i pomalo ironični naziv prezentacije uočenih i odabranih primjera s područja grada Zagreba ( 2003- 2007) : DOBRA KLIMA.

Ova prezentacija nije registriranje primjera radi njihove valorizacije s bilo kojeg stanovišta. Stoga adrese ostaju namjerno nepoznate, osim gdje to nije sasvim očito ili iznimno nekome poznato.

Prezentacija smjera na to da niti jedan fenomen u prostoru grada nije beznačajan ,naprotiv, da je vrijedan pozornosti . Uz to - da se u prezentacijskom uvidu prepustimo estetskom užitku, iznenađenju zanimljivosti ili negodovanju zbog stanja stvari . Ili pak u tehničkoj analizi , inspiracijama za rekonstrukcije, adaptacije itd. - već prema tome.

Autor ove prezentacije (snimaka i njihovog izbora ) javlja se pri tome kao posrednik i pomoćnik orijentirajući se prema vlastitoj znatiželji i opservacijama - ponukan da i s drugima podijeli uvide u posebnosti nekih fenomena DOBRE KLIME u Zagrebu

*Fedor Kritovac*

*Osnovni biografski podaci i značajke*

*Dr.sc. Fedor Kritovac, dipl. inž. arh.*

*Rođen 1938. u Zagrebu gdje se školuje, radi i živi. Studij arhitekture i doktorat društvenih znanosti iz područja urbane sociologije.*

*Radi u Centru za industrijsko oblikovanje u Zagrebu i do mirovine u Institutu građevinarstva Hrvatske.*

*Stručni, znanstveni, pedagoški, publicistički i javni rad usmjeren prema pitanjima metodologije, sustava, kvalitete proizvoda, prostora, objekata i okoliša, prostornog programiranja, planiranja i projektiranja. Interdisciplinarni pristup (tehnološki, ergonomski, marketinški, organizacijski, sociološki i umjetnički aspekti suvremenog svijeta - prvenstveno u urbanom okružju.*

*Urbani svijet - kao mjesto stvaranja, proizvodnje, potrošnje, primjene ,odbacivanja i recikliranja različitih materijalnih elemenata i sastava (industrijskih, obrtnih, građevina,) postave i djelovanja raznih vizualnih i audio medija/komunikacija. Kritičko i kreativno istraživanje i interpretiranje pojava koje se formiraju - katkada neočekivano , neobično i zanimljivo - u slici i doživljaju suvremenog svijeta.*

*Također dugogodišnje sudjelovanje u uredništvima časopisa (Čovjek i prostor i drugi) i udrugama .Stvaralaštvo i fotografijom, kolažom , karikaturom.*

*Istraživačko i kreativno usmjerenje od sredine 1980-tih god. odnosi se naročito na grad Zagreb. Uz brojne publicirane priloge (15 Dana, Komunalni vjesnik) niz tematskih izložaba o urbanim pojava (Galerija Modulor, Galeriia Prozori, Muzej grada Zagreba itd.)*

# OTVORENI NADZORNO UPRAVLJAČKI SUSTAV ZA REGULACIJU HVAC INFRASTRUKTURE U KONTEKSTU INTELIGENTNE ZGRADE (BMS)

*Ninoslav KURTALJ, dipl. ing.*

*Elma Kurtalj, d.o.o.*

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**Proizvodnja opreme**
**Distribucija i logistika**

**ČLANSTVA**

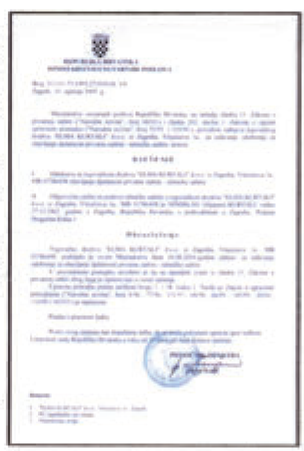
Članovi smo više važnijih svjetskih asocijacij iz područja djelovanja kao: LonMark, Central Automation Building Association i Oasis.

**KONFERENCIJE**

Redovno posjećivanje konferencija te aktivno sudjelovanje na njima.

| | |
|---|---|
| LonWord-LonWorks conference: | 2000;2001;2002;2003;2004;2005;2006 |
| BuilConn: | 2002;2003;2004;2005;2006 |
| BuilSpec: | 2005 |
| FTTH Council: | 2006, 2007 |
| KOM - OPATIJA | 2006 |

**Organizacijska jedinica - INTELIGENTNE ZGRADE**



Središte interesa ove organizacijske jedinice su energetski učinkovita rješenja te integracija distribuiranih sustava nadzora i upravljanja u zgradama. Način djelovanja organizacijske jedinice počiva na temelju principa **„potpuno otvorenog znanja"** .

Princip „potpunog otvorenog znanja" predstavlja koncept u kojem je u svakom trenutku moguće dodavati funkcionalnost sustava prema potrebi, i to kroz otvorni protokol na M2M razini bez obzira na proizvođače opreme.

Osim toga, u potpunosti smo sposobni izvesti **komisioniranje** zgrade od ideje do rješenja s permanentnom post-instalacijskom podrškom uključujući i daljinski nadzor.

**Tvrtka posjeduje ovlaštenje za izvođenje instalacija tehničke zaštite.**

U okviru ove organizacijske jedinice djeluje Odjel razvoja koji trenutno upošljava 11 ljudi od čega 7 software inženjera, te 4 visoko kvalitetna programera.

ELMA KURTALJ d.o.o. Vitezićeva 1a, 10110 Zagreb, Croatia, tel: 385-1-3035-555 , fax: 385-1-3035-599, e-mail: elma@elma.hr; web: www.elma.hr

# IZGRADNJA NOVIH ENERGETSKIH POSTROJENJA TE UPRAVLJANJE I ODRŽAVANJE ENERGETSKOG SUSTAVA U KBC-u ZAGREB - BOLNICA REBRO - NOVI ISKORAK U POSLOVANJU HEP TOPLINARSTVA d.o.o.

*Robert KRKLEC, dipl. ing., direktor HEP Toplinarstvo d.o.o.*

**ENERGIJA** Nakon potpisivanja ugovora HEP-a i KBC-a

## HEP u energetski sustav Rebra ulaže 53 milijuna kuna

Hrvatska elektroprivreda opskrbljivat će električnom i toplinskom energijom te upravljati cjelokupnom energetikom na lokaciji Kliničkog bolničkog centra Rebro.

Uz to HEP će uložiti oko 53 milijuna kuna u gradnju trafostanice i energane i prema dogovorenom modelu javno-privatnog partnerstva KBC će zadržati vlasništvo nad opremom i infrastrukturom, a brigu o održavanju preuzima HEP.

Okvirni ugovor takvog sadržaja 4. prosinca potpisali su ravnatelj Rebra Željko Reiner i predsjednik uprave HEP-a Ivan Mravak. Prema terminskom planu izgradnja postrojenja za hlađenje, potrebnih trafostanica kao i rekonstrukcija kotlovnice trajala bi oko 14 mjeseci što je uvjetovano završenjem potrebnih građevinskih radova - istaknuto je na potpisivanju sporazuma. DKB

Gradilište novoga kompleksa KBC Rebro

# PROJEKTI ENERGETSKE UČINKOVITOSTI U BOLNICAMA U IZVEDBI HEP-ESCO d.o.o.

HEP-ESCO d.o.o.
HEP grupa
mr. sc. Gordana Lučić, dipl. ing. stroj.

## 1. UVOD

Bolnice su objekti koji imaju zahtjevne potrebe za opskrbom energije. Kao prvo radi se o vrlo velikoj potrošnji energije i to tijekom cijelog dana, mjeseca i godine. Opskrba energijom mora biti pouzdana i sigurna jer se radi o najosjetljivijim potrošačima – bolesnicima i osoblju koje obavlja složene poslove zdravstvenog zbrinjavanja bolesnika. Kada se pri tome doda, da su velikim dijelom bolnice u RH vrlo stare s starim energetskim postrojenjima, jer se skromna investicijska sredstva koja se mogu osigurati za bolnice prioritetno koriste za nabavu medicinske opreme, vidi se složenost obnove bolnica.

Gledajući sa stanovišta projekata energetske učinkoviti bolnice spadaju u grupu najboljih projekata. Radi se o kontinuirano velikoj potrošnji energije, pa svaki zahvat na njezinom smanjenju rezultira značajnim uštedama. Priprema i izvedba projekata energetske učinkovitosti u bolnicama je složena zbog opsežnosti mjera koje se primjenjuju i velikih investicijskih ulaganja. Također je tu bitna organizacija izvedbe projekata i utjecaja izvedbe na okoliš tijekom izvedbe. Važno je da se niti jednom predviđenom mjerom ne dovode u pitanje higijenski i mikroklimatski uvjeti u bolnicama niti glede bolesnika niti radni uvjeti osoblja.
Pošto se redovno radi o velikim uštedama u energiji, primjena mjera energetske učinkovitosti u bolnicama, ima također značajan efekt u smanjenju emisije štetnih plinova i čestica u atmosferu. To se postiže direktno promjenom goriva (smanjenje zagađenosti okoliša) i indirektno manjom potrebom za proizvodnjom energije (npr. električne i toplinske energije)

U nastavku je dan pregled projekata energetske učinkoviti koji se pripremaju i izvode od strane HEP-ESCO d.o.o..

## 2. BOLNICE KOJE PRIPREMA I IZVODI HEP-ESCO d.o.o.

HEP-ESCO d.o.o. je do sada analizirao 13 bolnica, od kojih je za 6 napravljena samo prethodna Studija izvodljivosti mogućnosti primjene mjera energetske učinkovitosti, tzv. Walk Through Audit – WTA, za 2 se radi Investicijska studija, tzv. Investment Grade Audit, 3 bolnice su pred ugovaranjem i 1 je u izvedbi.

Tu se mora napomenuti da u svojim analizama HEP-ESCO d.o.o. uvijek razmatra cjelokupnu energetiku bolnica i predlaže primjenu najučinkovitijih mjera i to obično u nekoliko scenarija, kako bi vlasnici bolnica mogli odabrati najpovoljniji scenarij za njih.

### 3.1. Bolnice u pripremi

Za bolnice za koje je napravljena prethodna Studija izvodljivosti (OB Karlovac, Lječilište Lipik, Traumatološka bolnica Zagreb, OB "Dr. Tomislav Bardek" Koprivnica,

Specijalna bolnica Novi Marof, Specijalna bolnica za rehabilitaciju Varaždinske toplice) pokazano je da postoji veliki potencijal u primjeni mjera energetske učinkovitosti koje su se uglavnom odnosile na promjenu energenta – lož ulje u plin i optimiranje rada elektro potrošača u bolnicama.
Za bolnice OB Zadar i KBC Rebro rade se Investicijske studije.

HEP-ESCO d.o.o. je dao ponudu za izvedbu bolnice OB Zadar. Investicijska studija će biti podloga za odluku Zadarske županije o dodjeli poslova na izvedbi Projekta. Radi se modernizaciji bolnice u cilju smanjenja potrošnje energije primjenom solarne energije i regulacijom toplinske energije.

Za KBC Rebro radi se Investicijska studija primjene mjera energetske učinkovitosti na starom dijelu bolnice u skladu s okvirnim Ugovorom o opskrbi energijom KBC Rebro koji je potpisan između KBC Rebra i HEP-Toplinarstva d.o.o. 4. prosinca 2006.godine.

### 3.2. Bolnice koje su pred ugovaranjem za izvedbu Projekta energetske učinkovitosti – Klinička bolnica Osijek, Opća bolnica Varaždin i Opće bolnica "Dr. Josip Benčević" Slavonski Brod

Nakon gotovo dvogodišnje pripreme projekata energetske učinkovitosti u navedene 3 bolnice pripremljeni su ugovori za izvedbu projekata te se očekuje od bolnica, županija i Ministarstva zdravstva i socijalne skrbi, davanje suglasnosti i potpisivanje ugovora.

#### 2.2.1. Klinička bolnica Osijek

Projekt u Kliničkoj bolnici Osijek je najopsežniji gledajući opseg radova i vrijednost investicije, no ujedno je najpovoljniji sa stanovišta energetske učinkovitosti.

Radi se o prijedlogu modernizacije 12 sustava. To su slijedeći sustavi:

1. Povrat kondenzata iz cijele bolnice u HEP Toplinarstvo
2. Ugradnja odvodnika kondenzata i potrebne regulacijske armature
3. Sanacija sustava kondenzata i ekspandera pare u praonici rublja
4. Nova toplinska stanica Kuhinja
5. Ugradnja termostatskih ventila na radijatore
6. Rekonstrukcija klima komora (Ginekologija) - ugradnja rekuperatora
7. Modernizacija rasvjete-ugradnja fluokompaktnih sijalica i elektroničkih predspojnih naprava
8. Frekvencijski regulirani regulatori (gdje je moguće)-klima komore i dizala
9. Zamjena postojećih hladnjaka vode
10. Ugradnja CNUSA
11. Rekonstrukcija hidro stanice
12. Korištenje vode za tehnološke potrebe iz vlastitog bunara

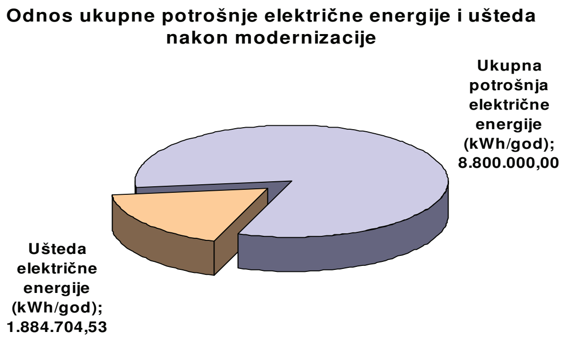
Iz prethodno navedenih mjera energetske učinkoviti vidi se da se radi o zahvatima na cijeloj bolnici. Radovi bi trajali najmanje 2 godine.

Procijenja vrijednost radova i opreme u Investicijskoj studiji od listopada 2005. godine bila je 11.137.000 kuna. Uz moguću ostvarenu uštedu u energiji od 2.624.000 kuna godišnje povrat troškova radova i opreme moguće je ostvariti u 4,2 godine (Jednostavni povrat investicije).

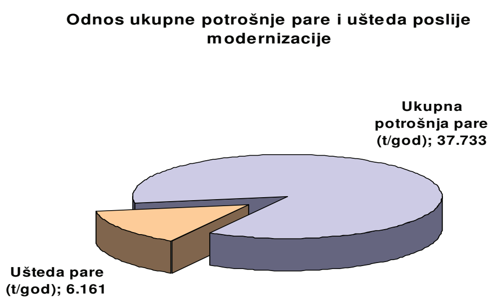
Kod bolnica se svi troškovi računaju s PDV-om zato što bolnice nisu u sustavu PDV-a pa je to dodatni trošak za njih.

Na slikama 1, 2, i 3, prikazane su ukupne uštede koje se mogu ostvariti primjenom mjera energetske učinkovitosti na potrošnji električne energije, pare i vode u Kliničkoj bolnici Osijek.

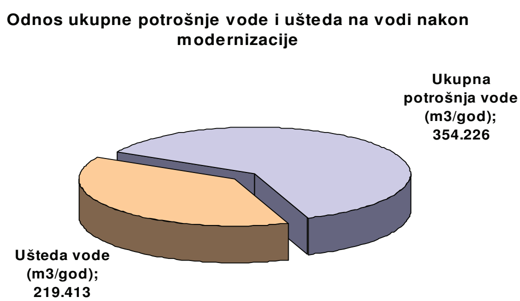
**Slika 1. Ukupne uštede električne energije nakon primjene mjera energetske učinkovitosti**




**Slika 2. Ukupne uštede pare nakon primjene mjera energetske učinkovitosti**




**Slika 3. Ukupne uštede vode nakon primjene mjera energetske učinkovitosti**



3.2.2. Opća bolnica Varaždin

Projekt primjene mjera energetske učinkovitosti u Općoj bolnici Varaždin najvećim dijelom odnosi se na optimiranje proizvodnje ogrjevne topline i pare. To su slijedeće mjere energetske učinkovitosti:

1. Zamjena kotlova i optimizacija sustava grijanja tj. smanjivanje broja kotlovnica, ukidanjem starih manjih kotlovnica na neurologiji, porti, radiologiji i laboratoriju. enju.
2. Sanacija sustava pare i kondenzata – praonica rublja ugradnjom reducijskih stanica,spremnika i ostale opreme.
3. Ugradnja termostatskih ventila
4. Zamjena parnog kotla, tj. zamjena jednog od dva parna kotla gdje bi novi kotao preuzeo najveći dio proizvodnje pare.

Trošak opreme i radova iznosi 7.044.280 kuna, a moguća ušteda u energiji je 1.404.832 kuna (u iznose je uključen PDV). Slijedi da je povrat investicije u radove i opremu moguć u 5 godina.

3.2.3. Opća bolnica "Dr. Josip Benčević" Slavonski Brod

Mjere energetske učinkovitosti koje su predviđene u Općoj bolnici "Dr. Josip Benčević" Slavonski Brod uglavnom se odnose na prelazak energenta – s mazuta na plin.

Bolnički godišnji troškovi za električnu energiju su 820.000 kuna, za srednje lako loživo ulje 1.520.000 kuna te za vodu 1.270.000 kuna. Ukupni izdaci za energiju su 3.630.000 kuna godišnje bez PDV-a. Potrošači pare u bolnici su sustav pripreme tople vode, sterilizacija, praonica rublja, kuhinja i slično. Zasićena para tlaka 8 bar proizvodi se u vlastitoj kotlovnici na lokaciji bolnice. Na lokaciji se ne koristi plin.

Mjere energetske učinkovitosti koje su predviđene odnose se na:

1. Zamjenu vrelovodnih kotlova na mazut kotlovima na plin,
2. Zamjenu dotrajalih parnih kotlova na mazut na kotlove na plin,
3. Rekonstrukcija automatike, podstanica i dijela toplovodnog sustava.

Trošak opreme i radova iznosi 8.174.000 kuna, a moguća ušteda u energiji je 1.059.252 kuna (u iznose je uključen PDV). Slijedi da je povrat investicije u radove i opremu moguć u 7,7 godina.

## 3.3. Projekt energetske učinkovitosti u izvedbi u Općoj bolnici "Dr. Ivo Pedišić"Sisak

Izvedba Projekta energetske učinkovitosti u bolnici započela je krajem 2006. godine. Osnovna koncepcija mjera energetske učinkovitosti je modernizacija postrojenja za proizvodnju toplinske nergije i prelazak na Centralni Toplinski Sustav (CTS) grada Siska.

U bolnici se koriste pet primarnih energenata: električna energija, voda, lako lož ulje, ekstra lako lož ulje i ukapljeni naftni plin. Potrošnja lakog lož ulja je oko 560.000 kg godišnje. Električna energija koristi se za pumpe, ventilaciju, kompresore, rashladne strojeve, rasvjetu i u manjoj mjeri za kuhinju. Potrošnja električne energije je oko 1.550.000 kWh godišnje. Voda se u najvećoj mjeri koristi u sanitarne svrhe, a u manjoj mjeri za nadoknadu gubitaka u sustavima grijanja i hlađenja. Godišnja potrošnja vode je oko 71.500 $m^3$ godišnje. Godišnji troškovi (po sada važećim cijenama) za električnu energiju su 670.000 kuna, za ekstra lako lož ulje 885.000 kuna te za vodu 940.000 kuna. Ukupni izdaci za energiju su približno 2.500.000 kuna godišnje.

Kao osnovni izvor toplinske energije, trenutno je niskotlačna para koju proizvode dva parna kotla smještena u prostoru kotlovnice zajedno sa svom potrebnom pripadajućom opremom. Para se po krugu bolnice distribuira parovodom sve do toplinskih podstanica (6 jedinica) po pojedinim objektima.

Mjere energetske učinkovitostosti koje su predviđene u bolnici odnose se na:

1. Uvođenje vrelovodnog priključka
2. Osiguranje rezervnog napajanja
3. Izgradnja nove kotlovnice
4. Izgradnja nove distribucijske vrelovodne mreže s rekonstrukcijom podstanica, uključivo demontaža postojeće opreme

Trošak opreme i radova iznosi 5.222.881,49 kuna a moguća ušteda u energiji je 548.640 kuna godišnje. ( PDV je uključen). Slijedi da je povrat investicije u radove i opremu moguć u 9,5 godina. Pošto HEP-ESCO d.o.o. može financirati projekte energetske učinkovitosti koji se otplaćuju u roku od 8 godina u financiranju izvedbe ovog projekta sudjeluje bolnica sa značajnim iznosom od 40% ukupne vrijednosti investicije.

## 4. UTJECAJ PROJEKTA ENERGETSKE UČINKOVITI U BOLNICAMA NA ZAŠTITU OKOLIŠA

Utjecaj projekata energetske učinkovitosti u bolnicama promatra se sa dva stanovišta To su:

- negativni utjecaji na okoliš tijekom izvedbe radova koji u što manjoj mjeri smiju smetati bolesnicima i zaposlenicima bolnice i
- pozitivni utjecaji na okoliš nakon završetka radova sa stanovišta smanjenja emisija SO2, CO2, NOx , krutih čestica i smanjenje temperature kondenzata koji se baca u kanalizaciju.

### 4.1.Utjecaj na okoliš tijekom izvedbe Projekta

Tijekom izvedbe radova obično dolazi do slijedećih negativnih utjecaja:

- emisija prašine i eventualno u manjoj mjeri plinova u atmosferu
- proizvodnja, skladištenje i odvoz otpada
- stvaranje buke

HEP- ESCO d.o.o. sa svojim podizvođačima definira najveći stupanj koncentracije prašine koja se može pojaviti tijekom izvedbe projekta, sustav upravljanja otpadom i najviši nivo buke koji može biti u krugu bolnice u skladu s aktualnom zakonskom regulativom.

Vezano za prašinu, tijekom izvedbe radova radit će se stalne kontrole npr. s uređajem za mjerenje koncentracija prašine u zraku.
Vezano za sustav otpada izvođač će biti obvezan izraditi elaborat o upravljanju otpadom u skladu s Zakonom o otpadu. Nadzorom nad radom izvođača to će se kontrolirati.

Vezano za buku, mjerenje i analize količine buke vršit će se tijekom rada izvođača radova putem ogovarajućih mjerača buke.

### 4.2. Utjecaj na okoliš nakon završetka modernizacije

U okviru Investicijske studije se na temelju moguće ostvarenih ušteda u energiji u bolnicama, proračunavaju iznosi smanjenja procjenjuje se emisije $CO_2$, $SO_2$, NOx i krutih čestica u atmosferu.

Tako npr. za **Kliničku bolnicu Osijek** modernizacijom 12 energetskih sustava može se uštedjeti pare od 6.161 t /god i i električne energije od 1.932.746 kWh/god. To rezultira smanjenjem emisije $CO^2$ od 910 t/god, $SO_2$ od 1.566 kg/god i NOx od 850 kg/god.

Smanjenje emisija zbog ušteda električne energije izračunavaju se na temelju podataka o emisijama cjelokupnog elektroenergetskog sustava Hrvatske po proizvedenom kWh u određenoj godini.

Smanjenje emisija $CO_2$ zbog smanjenih potreba za parom izračunate su s pretpostavkom proizvodnje pare u parnim kotlovima na plinsko gorivo (najpovoljniji slučaj)

Jedna od važnih komponenti je zaštita voda. To se odnosi na npr. na bacanje vode npr. – kondenzata, koji često ima temperaturu veću od dozvoljene od $40^0$ C u kanalizaciju.

## 5. MJERENJE I VERIFIKACIJA UŠTEDA

Mjerenje i verifikacija ušteda u energiji u bolnicama HEP-ESCO d.o.o. provodi prema International Performance Measurement & Verification Protocol[1] (IPMVP). Prema tom protokolu postoji 4 osnovne vrste mjerenja i verifikacije ušteda. To su:

- Metoda A – temelji se na utvrđivanju ušteda prema karakteristikama ugrađene opreme i projekciji ušteda do povrata investicijskih ulaganja,
- Metoda B - temelji se na mjerenju svih potrošača energije i utvrđivanju ušteda na temelju tih mjerenja
- Metoda C - temelji se na prikupljanju i obradi računa za potrošnju energenata
- Metoda D - temelji se na kompjuterskoj simulaciji potrošnje energije i ušteda uz korištenje specijalnih programa – kalibriranje potrošnje

Osnova za mjerenje i verifikaciju ušteda u bolnicama je Baseline, tj snimak postojeće energetske bilance u bolnici koja se priprema na temelju prikupljenih računa za energente u protekle tri godine i karakteristika ugrađene opreme i vremena njezinog korištenja.

Izbor metode bira se prema mogućoj primjeni. Tako npr. za bolnice koje ima i do 30 objekata na velikoj površini i koje koriste niz energenata (pare, vode, električne energije, plina, mazuta i dr.) jako je teško za cijeli bolnički krug primijeniti jednu metodu mjerenja i verifikacije ušteda. Pored toga npr. metoda B koja predviđa mjerenje svih energenata znatno povećava troškove projekta jer se obično radi o mjerenju u roku od 8 godina.

[1] Protokol je dostupan na www.ipmvp.org.

Metoda C koja se temelji na računima je prihvatljiva za globalno praćenje potrošnje energenata, no stvarnu uštedu je teško kontrolirati u ovako velikom Projektu jer se ne može kontrolirati način i trajanje rada pojedinih potrošača, te prikapčanje novih potrošača u krugu bolnice, tj. na mjestu obračuna energije.

Primjena Metode D je komplicirana zbog potrebe izrade specijalnog programa simulacije potrošnje energenata, pa je ocijenjena kao neprikladna za bolnicu.

Obično se metode mjerenja i verifikacije ušteda kombiniraju po pojedinim objektima bolnica i energetskim sustavima, tj. tamo gdje je to optimalno rješenje gledajući troškove mjerenja i mogućnost dobivanja realnih podataka.

Tu se mora napomenuti da HEP-ESCO d.o.o. u bolnicama, nakon izvedbe modernizacije opreme primjenom mjera energetske učinkovitosti, izlazi iz bolnice i prepušta vođenje i održavanje tehničkom osoblju bolnice. Pri kraju izvedbe Projekta osoblje se trenira i školuje za rad s novom opremom uz osiguranje uputa i priručnika od strane HEP-ESCO d.o.o.

## 5. ZAKLJUČAK

ESCO model modernizacije energetike zgrada s cjelokupnom opremom, gdje se investicijska ulaganja vraćaju kroz ostvarene uštede u energiji, je gotovo idealan model za primjenu u bolnicama. Bolnice stalno rade i kontinuirano troše velike količine energije, pa svaka ušteda u energiji se isplati i investicija se brzo vraća. Problem je bolnica u RH koje su velikim dijelom jako stare i ulaganje u obnovu energetike bolnica je vrlo skromno, pa uz projekte energetske učinkovitosti nužno je provesti obnovu koja bi se mogla tretirati kao tekuće održavanje zgrada i postrojenja, što poskupljuje Projekte u bolnicama.

HEP-ESCO d.o.o. ima iskustva u pripremi projekata energetske učinkoviti u bolnicama i planira to razvijati u obliku posebnog dijela tvrtke koja bi se osobito s tim bavila. Za sada HEP-ESCO d.o.o. nakon završetka modernizacije opremu u bolnicama prepušta na upravljanje i održavanje stručnim službama bolnica.

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