

MATHEMATICAL MODELLING AND SIMULATION OF THE THERMAL PERFORMANCE OF A SOLAR HEATED INDOOR SWIMMING POOL

by

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Buildings with indoor swimming pools have a large energy footprint. The source of major energy loss is the swimming pool hall where air humidity is increased by evaporation from the pool water surface. This increases energy consumption for heating and ventilation of the pool hall, fresh water supply loss and heat demand for pool water heating. In this paper, a mathematical model of the swimming pool was made to assess energy demands of an indoor swimming pool building. The mathematical model of the swimming pool is used with the created multi-zone building model in TRNSYS software to determine pool hall energy demand and pool losses. Energy loss for pool water and pool hall heating and ventilation are analyzed for different target pool water and air temperatures. The simulation showed that pool water heating accounts for around 22%, whereas heating and ventilation of the pool hall for around 60% of the total pool hall heat demand. With a change of preset controller air and water temperatures in simulations, evaporation loss was in the range 46-54% of the total pool losses. A solar thermal sanitary hot water system was modelled and simulated to analyze its potential for energy savings of the presented demand side model. The simulation showed that up to 87% of water heating demands could be met by the solar thermal system, while avoiding stagnation.

Key words: *indoor swimming pool, mathematical model, solar heating*

Introduction

Indoor swimming pool buildings have large energy consumption. Here, the greatest energy consumer is the swimming pool hall (SPH) [1], where energy is used for maintaining thermal comfort conditions in the hall and pool water (PW). The breakdown of energy consumption in indoor swimming pool buildings shows that: 45% of energy is used for SPH ventilation, 33% for heating PW, 10% for heating and ventilation of the rest of the building, 9% for lighting and equipment, and 3% for sanitary hot water (SHW) [1], but a detailed insight into the PW and PH loss has not been reported. The SPH together with the swimming pools is the greatest energy consumer, and may account for up to 60% of the total energy use of the pool building [1-3], therefore energy balance of the pool and PH is modelled with more detail in this paper. In indoor SPH, air temperature and humidity are maintained at a predetermined level, as analyzed in the

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paper. Pool water heating and pool hall air heating using heating, ventilation and air conditioning (HVAC) equipment increases water evaporation and relative air humidity, which raises energy demands of the building. A careful design and control of the HVAC system is needed to maintain water and air temperatures and air humidity at optimal levels for efficient operation. In this paper, target values of PW and SPH air temperatures were varied in the simulations to determine their impact on energy losses and address the problem of optimal values of these parameters. Energy balance models are usually a starting point of any performance, design or energy efficiency analysis of swimming pools [2, 4-6], but the impact of desired air and water temperatures on the SPH energy balance was not considered. The highest thermal loads in indoor swimming pools, often originate from water evaporation from the pool water surface, but the effect of evaporation is difficult to model with precision [7].

Due to a similar profile of heat demand and available solar radiation, solar energy has been used to heat swimming pools for decades. The utilization of solar energy, due to relatively low temperatures at the demand side, is especially significant for open swimming pools. In 1963, Czarnecki [8] proposed a method for solar heating of outdoor swimming pools using a transparent plastic pool cover, which raised solar radiation gains while reducing evaporative loss. Szeicz and McMonagle [9] made an energy balance of urban swimming pool in 1982, and found the use of solar collectors and pool blankets to be most effective. The performance of an open absorption system and mechanical heat pump for use in indoor swimming pool was analyzed in [10], and 40-50% of total energy demand was found to be related to hot water production. Hahne and Kubler [3] performed a simulation of outdoor swimming pool using TRNSYS software, and reported that the use of unglazed solar collectors could reduce natural gas consumption for heating by 40%, but sanitary hot water was neglected. In this paper, the application of Flat Plate Glazed (FPC) solar collector system for an indoor swimming pool is analyzed.

A detailed research of public open swimming pools in Central Europe shows that pure solar radiation is sufficient for heating pool water [10,11]. Heat losses of pool water depend on the temperature difference of water and ambient air, air humidity, air flow velocity above the pool, flow rate and temperature of the cold water, and thickness and quality of pool insulation. The daily heat loss of indoor swimming pools is estimated to 2.5 kWh/m² per day [10, 12, 13], but detailed structure of contributing factors to the SPH was omitted. Energy efficiency of the indoor swimming pool using the preliminary energy balance method indicated possibilities for improvement using solar and heat pump systems [14, 15]. Solar energy systems used as primary energy source to heat SHW represent around 80% of installed solar applications [10, 16].

In this paper, a case study analysis of energy efficiency of indoor swimming pools of the Sport and Recreation Centre (SRC) "Dubočica" in Leskovac, Serbia, is performed. Effects of the application of FPC for water heating are also analyzed. The focus of analysis and results was the SPH, as the greatest energy consumer in the building, and the influence of target values of water and air temperatures on the balance. A mathematical model of the swimming pool was created to interact with input and output variables of TRNSYS Type 56 building model and the modelled HVAC system during annual behaviour simulations. Free water surface evaporation correlation for occupied and unoccupied swimming pools were chosen for the simulations [17]. The performance of solar systems with respect to building layout and design are discussed in [18]. In this paper, the surface of the modelled FPC array was increased in simulations, to determine the size of the solar system with the maximum ratio of heating demand met by the solar system without stagnation.

Mathematical model of indoor swimming pools

HVAC systems in indoor swimming pool buildings are designed to provide suitable thermal comfort conditions for pool users in the SPH areas. Indoor temperature is kept at relatively high levels (24-29 °C), creating conditions for significant evaporation rates from the pool water surface.

Main contributors to the energy losses of an indoor SPH are (fig. 1): conduction through the pool walls Q_{cond} , convection from the pool surface Q_{conv} , radiation from the pool surface Q_{rad} , evaporation from the pool surface Q_{Evap} , heat loss due to fresh water flow \dot{Q}_{fw} for water loss compensation, and heat flow rate from heating Q_{aux} .

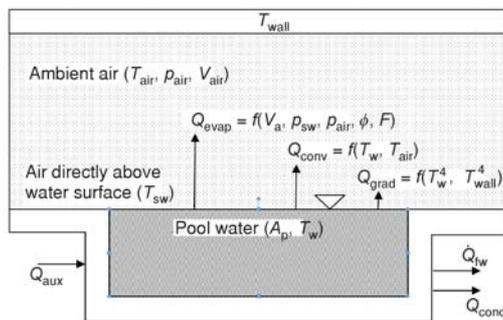


Figure 1. Swimming pool energy balance model scheme

Conduction through the pool walls is usually negligibly small, unless the pool is above ground or in direct contact with cold groundwater, which is rare [3]. Convection from the pool surface is a function of temperature of the air in the SPH and air speed above the water surface. Radiation losses are more important for outdoor swimming pools [3], but can also occur as a consequence of long wave radiation exchange with indoor walls [19]. Evaporation heat losses contribute to as much as 50-60% of pool heating losses [20].

Energy balance of outdoor swimming pools [2, 3] is used to simulate the performance of different heating applications. The energy balance of indoor swimming pool presented in this paper is tailored to be used with TYPE 56 multi-zone building model, from TRNSYS 17 component model library. It uses output variables for pool hall air properties, and wall temperatures for each time step of the simulation, and interacts with the modelled HVAC system inputs, to contribute to air properties in the pool hall zone.

It is assumed that PW is ideally mixed and that the fluid is incompressible. Water density and conductivity are considered constant. A balance model of an indoor swimming pool presented in [5], takes into account air infiltration losses of the SPH and outside air intake, but it neglects radiation heat transfer and losses due to fresh water supply.

The temperature change of PW over time τ can be calculated as:

$$\rho_w c_{pw} V_p \frac{dT}{d\tau} = \dot{Q}_{\text{aux}} - (\dot{Q}_{\text{fw}} + Q_{\text{evap}} + Q_{\text{conv}} + Q_{\text{rad}}) \quad (1)$$

Evaporation heat losses are proportional to the flow of water evaporated from the swimming pool water surface:

$$Q_{\text{evap}} = A_p q_{\text{evap}} = A_p \dot{E} r \quad (2)$$

where \dot{E} is the mass flow rate of evaporated water and r – the latent heat of evaporation.

Heat transfer by convection from the water surface per unit surface area can be written as:

$$Q_{\text{conv}} = A_p q_{\text{conv}} = A_p \alpha (T_w - T_{\text{air}}) \quad (3)$$

where α is the convective heat transfer coefficient, T_w – the temperature of the pool water and T_{air} – the indoor air temperature in the pool hall. Convective heat transfer coefficient can be expressed as [2]:

$$\alpha = 2.8 + 3.0 V_{\text{air}} \quad (4)$$

where V_{air} stands for the air velocity above the water surface.

Radiation losses to the sky calculated according to Stefan-Boltzmann law for outdoor swimming pools are considered in [2, 3]. The proposed indoor swimming pool model takes into account heat transfer by long-wave radiation exchange with pool hall walls [21], given per unit of pool area:

$$Q_{\text{rad}} = A_p q_{\text{rad}} = A_p \varepsilon \sigma (T_w^4 - T_{\text{wall}}^4) \quad (5)$$

where ε [-] is the emissivity average, σ – the Stefan-Boltzmann constant ($\sigma = 5.67 \cdot 10^{-8} \text{ Wm}^{-2}\text{K}^{-4}$), and T_{wall} – the temperature of the wall surface.

The proposed swimming pool water assumes a constant water level in the pool. Evaporated water mass and waste water losses in the water treatment plant are compensated by fresh water supply. Heat loss caused by supply of fresh water with lower temperature than that of the pool can be calculated by:

$$\dot{Q}_{\text{fw}} = \dot{m}_{\text{fw}} c_{\text{pw}} (T_w - T_{\text{fw}}) \quad (6)$$

The pool heat losses must be compensated by heat gains. Direct solar irradiation on the water surface is considered in [3, 21] for outdoor swimming pools. Heat gains from solar collectors are considered in [3], and heat pump heating system in [6]. Heat gains from auxiliary heating equipment to the swimming pool can be expressed as:

$$\dot{Q}_{\text{aux}} = \dot{m}_{\text{aux}} c_{\text{pw}} (T_w - T_{\text{aux}}) \quad (7)$$

where \dot{m}_{aux} is the mass flow rate supplied by the auxiliary heating equipment, and T_{aux} – the temperature water supplied for auxiliary heating.

Water evaporation from the pool water surface

The phenomena of water evaporation from the free water surface of an indoor swimming pool is a significant contributor to the total energy balance of the pool building [22-25]. If the total pressure of humid air on the water surface, which is equal to the sum of the partial pressure of saturated water vapor at a given temperature and partial pressure of air, is greater than the saturation pressure of water vapor, then water evaporates only from its free surface. The high rate of occurrence of this particular phenomena in many technological applications, led to many correlations for predicting water evaporation rate from a free water surface to both still and moving air available in the literature [20-34]. Bowen [29] obtained a general solution to determine heat transfer by convection and evaporation from a water surface of an element of volume for three different conditions, where he first determined a model of vapour diffusion from a unit area. Water evaporation rate from a surface depends on flow regime (laminar or turbulent) which can be categorized according to the governing convection mechanism to free convection or forced convection. Most of the correlations are based on the Daltons theory, but there are also attempts of creating correlations based on the analogy between heat and mass transfer [22, 29], where a ratio between heat loss by conduction to heat loss by evaporation is determined. Even for detailed numerical simulations [30, 32] it is necessary to first determine the evaporation rate from water surface and the evaporation rate coefficient.

The first equation of evaporation from the free water surface was given by Dalton [26], when it was found to be proportional to the partial pressure difference of water vapor near the boundary surface p_{sw} and away from the surface p_{a} :

$$-dE = K(p_{\text{sw}} - p_{\text{air}})dA_{\text{sw}} \quad (8)$$

where dA_{sw} is the element of evaporation surface, dE – the evaporation rate per unit time, and K – the coefficient affected by properties of air flow over the boundary water surface.

Most of the models are an effort to improve on Dalton's evaporation equation, acquired as mathematical approximation of evaporation coefficient using experimental data [23, 27, 28]. A review and comparison of mathematical models for predicting evaporation rates was done by Sartory [22], where the results obtained using evaporation models from literature showed a large scattering of the prediction results. A general mathematical representation of literature correlations for predicting evaporation rate from the free water surface can be written as [17]:

$$E = \frac{C(A + BV_a)^n (p_{sw} - \phi p_{air})}{r} \quad (9)$$

Here, A , B , C , and n are the correlation constants usually acquired through experiment, V_{air} – the velocity of air over the water surface, r – the latent heat of evaporation of water, and ϕ – the relative humidity of air. The term in the second bracket of eq. (9.) represents the vapour pressure difference between the saturated air directly above the pool surface and the pressure of the air in the space higher above the water level (*i. e.* pool hall air).

To improve energy efficiency of the building, evaporation rates should be kept at a reasonably low level, while maintaining thermal comfort for the swimmers in both water and pool hall space. Pool hall heating and ventilation equipment design guides [20] recommend setting 2-3 °C higher SPH air temperature over PW temperature, which was adopted for determining target values for the control system in the simulations in this paper.

Air velocity in indoor SPH originates from heating and ventilation equipment operation, and typically ranges from 0.035-0.2 m/s [20, 23, 30]. The field of air velocity over the indoor swimming PW body is not uniform but extremely complex, with variations in both intensity and direction as seen in results of a CFD simulation of a public swimming pool [30]. The nature of air flow over water surface in indoor swimming pools, affected by the Reynolds number and Sherwood number, has a strong influence on evaporation phenomena [31]. Other measurements indicate that the velocity values could be expected in the range from 0.035-0.5 m/s, but an average value of 0,15 m/s was found [23], similar to engineering standards which recommend adopting a value between 0.05-0.15 m/s for system design purposes [20].

A review of equations for determination of free water surface evaporation indicates a great disagreement between results of evaporation rates obtained by analyzed correlation models, which makes the choice of a right model difficult [22]. Sartori [22] compared results of equations for air velocity of 3 m/s, relative humidity of 100% and 45%, and water temperature in the range from 10-40 °C. These conditions are not a good match for the ones found in indoor swimming pools [5, 20, 23-25, 33] where: (1) PW temperature ranges between 24-29 °C, (2) air temperature is 1-3 °C higher than the PW temperature; (3) relative humidity ranges from 50% up to 80%, and (4) average air velocity ranges between 0.05-0.2 m/s. Experimental data corresponding to these conditions should be used to evaluate these models. The results obtained in a scaled chamber model of an indoor swimming pool using specially designed laboratory apparatus of Asdrubali [33] seem to best correspond to values of parameters commonly found in indoor swimming pools, and are used as such to evaluate evaporation rate results obtained using different correlations for unoccupied swimming pools [17]. Correlation constants for eq. (9) are chosen based on [17] providing results closest to the measured results of Asdrubali [33] for air velocities ranging from 0.05-0.17 m/s, and relative humidity from 50-70%. In this paper,

evaporation rate for an unoccupied swimming pool is predicted using eq. (9). with the following constants: $A = 0.0163$, $B = 3.1$, $C = 4.1$, and $n = 0$; chosen according to [17, 33].

Occupied swimming pools

Evaporation from occupied swimming pools is much higher than that of unoccupied swimming pools [21, 24, 30]. There are several approaches and efforts aimed to account for the difference in evaporation rate of occupied pools. The differences in evaporation rates is related to the number of occupants, their activity, but also to the conditions of water and ambient air in the SPH determined by pool type. ASHRAE recommends using a correction factor ranging from 0.5 for residential pools to 1.0 for public pools with Carrier evaporation rate equation [20] for predicting evaporation rates of occupied swimming pools.

The following phenomena affect evaporation rates in occupied swimming pools: (1) disturbance of the water surface and formation of waves causes an increased water surface area (A_{ww}) where the mass transfer takes place, (2) splashing and drip from the swimmers lead to a wet deck, which increases water surface area where the mass transfer occurs (A_{sp}), (3) wet bodies of swimmers are exposed to air resulting in additional increase of water-air surface area (A_b), with higher rates of evaporation than that of the pool surface due to higher body temperatures compared to PW temperature, and (4) sprays of water caused by activity of the occupants additionally increase the water-air contact surface area (A_{sy}). If temperature variation of swimmer bodies is neglected, then the evaporation rate of occupied swimming pools can be calculated by multiplying the evaporation rate per unit of PW surface area with the increased water surface area of occupied swimming pools A_{pi} :

$$A_{pi} = A_p + A_{ww} + A_{sp} + A_b + A_{ww} + A_{sy} \quad (10)$$

Auer [21] presented a function of pool occupancy, with respect to pool opening and closing time but it is difficult to acquire surface values for eq. (12) using this model. To avoid difficulty of assuming these values, the pool occupancy factor is introduced as [24]:

$$F = \frac{A_{max}}{\frac{A_p}{N}} \quad (11)$$

where N is the number of occupants and A_{max} – the pool area per person at maximum pool occupancy. Smith *et al.* [35] proposed a formula for the occupied swimming pool as a function of pool occupancy factor based on correlation with test data. Biasin and Krumme [30] proposed their correlation for occupancy values between 0.1 and 0.7, but their correlation was influenced by evaporation from showers. Shah [24] proposed a model for occupancy factor larger than 0.1 and corrections of values of evaporation rates calculated for unoccupied swimming pools. In this paper, daily and monthly changes of the occupancy factor F , are modelled using TRNSYS Type 14. Evaporation rate of unoccupied swimming E pool is multiplied by a linear function of the occupancy factor F , to obtain the evaporation rate of the occupied swimming pool E_0 [24]:

$$E_0 = \begin{cases} E(3.3F + 1), & F < 0.1 \\ E(1.3F + 1.2), & 0.1 \leq F \leq 0.1 \end{cases} \quad (12)$$

Simulation of indoor swimming pool

In this paper, a case study of indoor swimming pools in the Sport and Recreation Centre “Dubočica” with 55000 users annually is analyzed. The total water surface of the PW area is

1480 m². Indoor thermal comfort conditions in the SPH and other rooms are achieved using an energy system with installed heat exchangers rated at around 3.35 MW in total [14, 17]. PW heating accounts for 45.08% of the total installed heat exchanger power, and heating and ventilation 31.93% [14, 17]. SHW heating represents only 4.33% of the installed heat exchanger capacity.

In this paper, the building was modelled as multi-zone in TRNSYS software, with typical masonry construction and 5 cm mineral wool insulation. The rooftop has a steel plate construction with 10 cm mineral wool insulation, and outside windows and doors have double glazing and metal frame. Most attention in the simulation was paid to the SPH, PW, and their interaction.

The mathematical model of the swimming pool presented in this paper, Type 56 model of the building and air distribution HVAC system was modelled to simulate the behaviour of the SPH. Partial pressure of saturated vapor is calculated as a function of temperature in the simula-

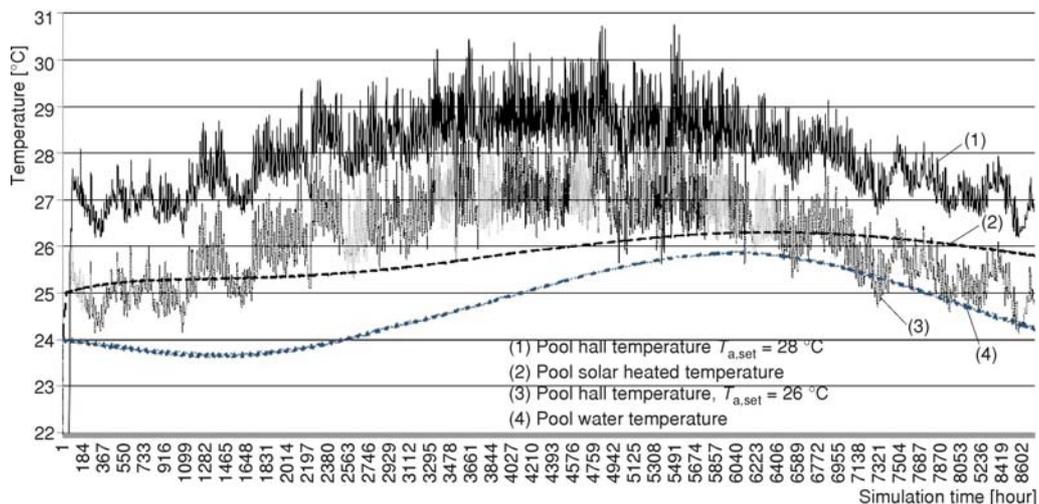


Figure 2. Annual pool hall air and pool water temperature for different simulation scenarios

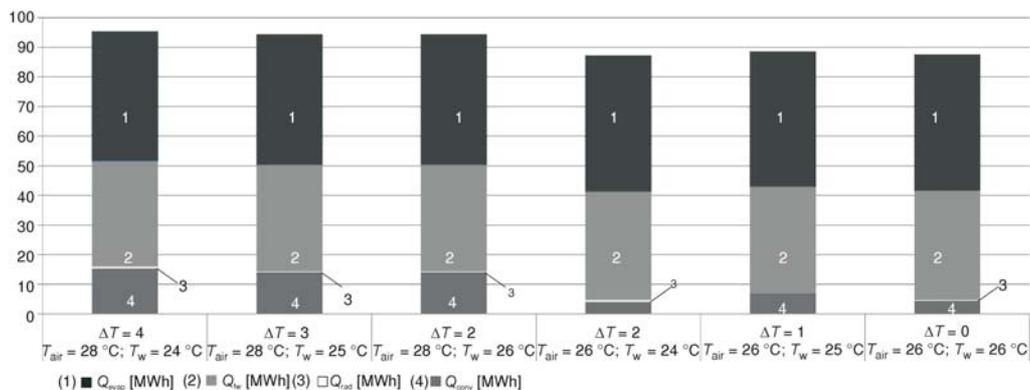
tion [36]. The SPH is heated by an air duct system. Air velocity in the SPH was assumed constant and equal to 0.08 m/h. Air temperature in the SPH changed during the simulation, due to slow response of the control system and a large volume of air to be heated (fig. 2). The simulation was done with Meteonorm hourly weather data, which included mains water temperature used for fresh water supply. Annual change of ambient air temperature in the SPH and PW temperature change obtained by simulation is shown in fig. 2. The simulation indicated a demand for HVAC system operation and heating even in the summer which can be related to the highest number of visitors leading to higher SHW demands, but also SPH and PW losses (tab. 1, fig. 3). Raised heating demands in the summer justify solar thermal applications, as described in this paper. To determine the impact of pool losses and PW surface evaporation, SPH controller preset temperature ($T_{a,set}$) was varied from 26 °C to 28 °C, while PW controller preset temperature was kept 1-3 °C lower (tab. 1). Annual energy balance of the SPH, showed that around 60% of energy was used for HVAC of the pool hall (tab. 1), around 20% for PW heating, and that the total SPH loss could be lowered by 10.3% just by adjusting SPH air and PW target temperatures in the control modules.

Table 1. Distribution of the pool hall loss

	Unit	$T_{air} = 28\text{ }^{\circ}\text{C}$ $T_w = 24\text{ }^{\circ}\text{C}$	$T_{air} = 28\text{ }^{\circ}\text{C}$ $T_w = 25\text{ }^{\circ}\text{C}$	$T_{air} = 28\text{ }^{\circ}\text{C}$ $T_w = 26\text{ }^{\circ}\text{C}$	$T_{air} = 26\text{ }^{\circ}\text{C}$ $T_w = 24\text{ }^{\circ}\text{C}$	$T_{air} = 26\text{ }^{\circ}\text{C}$ $T_w = 25\text{ }^{\circ}\text{C}$	$T_{air} = 26\text{ }^{\circ}\text{C}$ $T_w = 26\text{ }^{\circ}\text{C}$
Temperature difference	$^{\circ}\text{C}$	$\Delta T = 4$	$\Delta T = 3$	$\Delta T = 2$	$\Delta T = 2$	$\Delta T = 1$	$\Delta T = 0$
SHW heating	%	15.93	15.80	15.80	17.03	17.23	16.94
PW heating	%	21.20	21.34	21.34	22.12	21.79	22.08
HVAC of the SPH	%	62.86	62.86	62.86	60.85	60.98	60.98
Total SPH loss	MWh	95.4	94.5	92.4	87.4	88.5	85.5

Annual simulated temperature change of SPH air with HVAC controller ambient air temperature ($T_{a,set}$) set to $26\text{ }^{\circ}\text{C}$ and $28\text{ }^{\circ}\text{C}$, and the PW temperature when heated by the solar and auxiliary heating as well as just by solar heating are presented in fig. 2. The PW temperature difference with and without auxiliary heating applied, indicates that most of the pool heating demand could be met by the solar system. Solar energy alone may cover energy demands for PW heating for pools used for professional sport events and training, where PW water temperatures are preferred.

Evaporation pool (Q_{evap}) loss is significantly larger than other losses, ranging from 46-54% of total PW loss (fig. 3). Simulations showed that pool loss by wall radiation (Q_{rad}) could be neglected. Pool convection loss (Q_{conv}) rose with increased temperature difference between SPH air and PW ranging from 4.7% to 15.9% of the total SPH loss. SPH losses increased with higher preset controller temperatures of both PW and SPH ambient air.

**Figure 3. Breakdown of pool losses, for different set point temperatures of ambient air (T_{air}) and pool water (T_w)****Table 2. Ratio of energy supplied by the solar system in annual hot water demand**

Collector surface gross area	[m^2]	301.20	331.32	361.44	391.56
No. of solar collectors	%	120	132	144	156
Solar PW heating	%	68.25	78.74	88.49	96.08
Solar SHW heating	%	74.55	75.15	75.70	76.40
Total solar heating ratio	%	71.07	77.14	82.77	87.28

According to the hot water demand profile, the maximum collector surface which could be used while avoiding collector stagnation was 391.56 m^2 , *i. e.* 156 collectors, which corresponds to 26.4% of the PW surface. The obtained PW temperature curve, presented in fig. 2. corresponds to the mentioned largest solar collector array area analyzed in the paper. The number of collectors was lowered, with a step of 12 collectors, to investigate effects and performance of the solar system (fig. 4). The ratio of hot water demand met by solar collectors is shown in tab. 2. The ratio of solar PW heating is higher than the ratio for solar SHW heating since PW heating was modelled as primary heat sink in the solar loop. The solar collector loop transfers heat first to PW via a heat exchanger, and then to the 5 m^3 SHW boiler. When the PW reaches the preset temperature, the controller turns of hot water flow through to the swimming pool loop, and then hot water from the FPC array is used only for SHW heating. Both SHW boiler and swimming pool use auxiliary heat exchangers. SHW boiler water is mixed with the water from the mains, to simulate shower use. The output shower mixing temperature was set to $40 \text{ }^\circ\text{C}$. SHW design load corresponds to 650 swimmers, which occurs only in the summer months and gradually decreases to a minimum of 10% of design load in the winter. The same trend of occupancy was used to account for monthly change of occupancy factor F .

The solar energy gain acquired using different total gross surface area of FPC (fig. 4), indicates correspondence with the summer heating demand. With the possibility of lower pool occupancy than assumed by this model, lower ratio of solar energy supply should be used for the solar system design and sizing, or additional heat storage and/or stagnation prevention system should be installed. Simulation results with lower FPC surface area are shown in fig. 4. Careful modelling of the demand side is an important factor for solar system sizing and simulation to avoid stagnation issues.

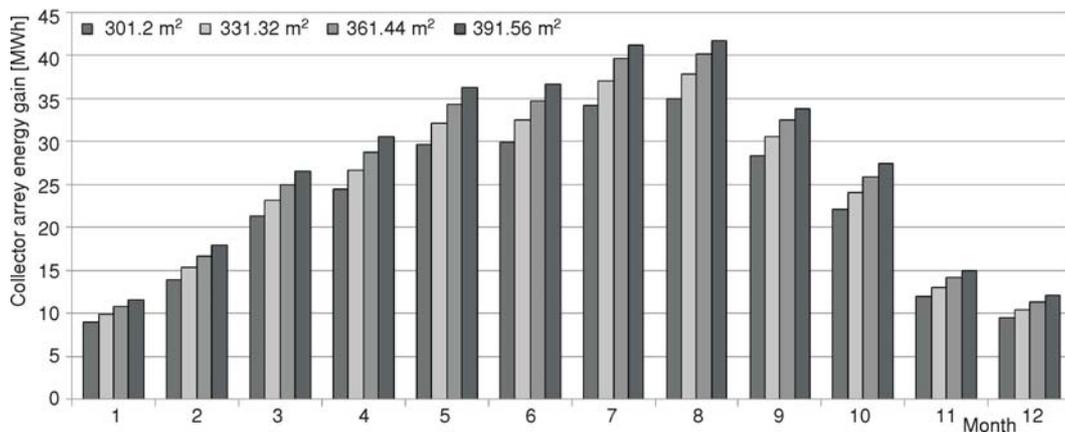


Figure 4. Energy gain of analyzed gross surfaces of solar collector arrays

Conclusions

In this paper, an energy balance model of an indoor swimming pool was created to simulate annual energy demands of such buildings. A physical model of the pool was adopted and a corresponding mathematical model was presented. Correlation for predicting evaporative rate from the PW surface can greatly affect system sizing and building behaviour simulation.

For the best results, the evaporation model should be calibrated using the measured results from the object of interest.

A simulation of solar heated swimming pool was made using TRNSYS software and the mathematical model of the swimming pool described in the paper. Energy balance of the SPH, as the zone with greatest energy loss in the building, showed that SPH heating and ventilation accounts for around 60% of total SPH loss, while PW loss is around 22%. Preset controller temperatures of SPH air and PW were varied in the simulations. The pool energy loss rose with the increase in both SPH air and PW preset control temperatures, by as much as 10.3%. The evaporation loss had the highest values compared to other PW losses, ranging from 46-54% of the total PW loss, and the convective PW heat loss increased 2 times with the increase in preset temperature difference between SPH air and PW to 4 °C.

Precise modelling of annual heating demand is necessary for proper solar system sizing. A maximum solar collector area corresponding to 26.4% of the PW surface was obtained while avoiding stagnation with the assumed SHW demand profile. Due to the similarity of the solar energy gain profile and heat demand profile of the swimming pool, 71-87% of SHW and pool heat loss could be met using FPC. Due to low temperatures of PW modelled as primary heat sink in the solar loop, the ratio of pool heating demand met by the FPC was higher than the ratio for SHW ranging from 68-96%. A significant share of hot water demand can be met by solar thermal technologies but careful system design and sizing with proper demand side modelling is necessary to avoid stagnation issues.

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Nomenclature

A	– surface area, [m ²]
c_{pw}	– specific heat of water, [Jkg ⁻¹ K ⁻¹]
E	– evaporation rate, [kgs ⁻¹]
F	– pool occupancy factor
p	– partial pressure of water vapor in humid air, [Pa]
Q	– heat flux, [W]
q	– heat transfer per surface area, [Wm ⁻¹]
r	– latent heat of evaporation of water, [Jkg ⁻¹]
T	– temperature, [°C]
V	– velocity, [ms ⁻¹]
V_p	– swimming pool volume, [m ³]

Greek symbols

α	– convective heat transfer coefficient, [Wm ⁻² K ⁻¹]
ρ	– density, [kgm ⁻³]

Subscripts

cond	– conduction
conv	– convection
evap	– evaporation
fw	– fresh water
p	– pool
rad	– radiation
sw	– at water surface
w	– wall, water

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