A NEW METHOD FOR THE OPTIMIZATION OF INSULATION THICKNESS FOR RADIANT WALL HEATING SYSTEMS

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Abstract: In this study we proposed a new modified New Degree-Day Method (NDDM) for the optimization of insulation thickness of the wall where the radiant panels are mounted (WMRP) in which heat generation inside the wall is considered. The existing Standard Degree-Day Method (SDDM) is not applicable to estimate the optimum insulation thickness for the buildings where the WMRP is mounted. Because SDDM method uses indoor air temperature as a base temperature, hence heat generation through the WMRP cannot be taken into account. In the new method, important parameters were obtained from the series of the CFD analysis for different thermal transmittance coefficient (U) and outdoor air temperature (To) values are used to create an empirical equation for the estimation of Tp (new base temperature) with the multiple polynomial regression method. Then the numerical results were validated with experimental results which were obtained from the real-size test chamber. Using the new method optimum insulation thickness, net energy saving and payback periods for radiant wall heating systems were calculated (for Istanbul climate) and compared with the results which were obtained using the standard degree-day method (SDDM). The results showed that, the SDDM significantly lower (85-95%) estimates the optimum insulation thickness and can't be used for the buildings where the WMRP is used. The new method can be used for radiant wall heating systems where the performance of radiant heating systems is significantly affected by the insulation capabilities and has a great importance in the sizing process of the radiant systems.
Keywords: Radiant wall heating; Optimum insulation thickness; New degree-day method.

IŞINIMLA DUVARDAN ISITMA SİSTEMLERİNDE YALITIM KALINLIĞI OPTİMİZASYONUNDA KULLANILABİLECEK YENİ BİR YÖNTEM

Özet: Bu çalışmada ışınım ısıtma panellerin kullandığı duvarlardaki yalıtım kalınlığının optimizasyonu için, duvarlardaki ısı üretimini dikkate alan, yeni bir derece-gün yöntemi (NDDM) geliştirilmiştir. Standard Derece-Gün Yöntemi (SDDM) temel sıcaklık olarak mahal hava sıcaklığını dikkate almakta ve duvara monte edilmiş ışınım panellerindeki ısı üretimini dikkate alamamaktadır. Bu yüzden standart yöntem ile ışınım panellerin bulundugu duvarlar için yalıtım kalınlığı optimizasyonu yapmak imkansızdır. Önerilen yeni metotta kullanılan yeni temel sıcaklık değerinin (Tp) elde edilmesinde kullanılan ampirik ifade (3. dereceden polinom) farklı yapı ısı geçirgenlik katsayısı (U) ve farklı dış hava sıcaklıkları (To) parametreleri için sayısal analizlerden elde edilmiştir. Daha sonra sayısal çalışmaların sonuçları aynı şartlarda yürütülen gerçek ölçekli deney sisteminde doğrulanmıştır. İstanbul iklim şartları için yeni yöntem ve eski yontem kullanılarak ideal yalıtım kalınlıkları, enerji tasarrufları ve geri dönüş süreleri hesaplanmış, iki yöntemden elde edilen sonuçlar kıyaslanmıştır. Sonuçlara göre eski yöntemle hesaplanan ideal yalıtım kalınlığı yeni yöntemden elde edilen değerin çok altında (%85-95) kalmaktadır. Bu yüzden standart yöntemin ısı üretimi olan duvarlarda kullanılmasının mümkün olmadığı görülmüştür. Önerilen yeni yöntem ise, ışınımla ısıtma sistemlerinin projelendirilmesinde önemli bir kriter olan ısı kayıplarının hesaplanması ve ideal yalıtım kalınlığının belirlenmesinde kullanılabilecektir.
Anahtar Kelimler: Işınımla duvardan ısıtma, Optimum yalıtım kalınlığı, Yeni derece-gün yöntemi.

NOMENCLATURE

A	Area [m^2]
C_A	Annular heating cost [TL/yr]
C_f	Fuel cost [TL/kg]
C_{tins}	Insulation cost per unit area [TL/m^2]
h_i	Inner heat transfer coefficient [W/m^2.K]
h_o	Outer heat transfer coefficient [W/m^2.K]
H_u	Lower heating value of the fuel [j/kg]
k	Heat conduction coefficient [W/m.K]
m_f	Annular fuel mass [kg]
R_i	Inside air film thermal resistance [m^2.K/W]
R_o	Outside air film thermal resist. [m^2.K/W]
R_w	Total thermal resistance [m^2.K/W]
T_b	Base temperature [°C]
T_i	Indoor air temperature [°C]
T_o	Outdoor air temperature [°C]

T_p	WMRP backside temperature [°C]
T_{si}	Inner surface temperature of the wall [°C]
T_{so}	Outer surface temperature of the wall [°C]
q''	Heat flux [W/m²]
U	Coefficient of thermal trans. [W/m².K]
x	Insulation thickness [m]
x_{op}	Optimum insulation thickness [m]
η	Efficiency of heating system [%]

Abbreviations

CFD	Computational Fluid Dynamics
DD	Degree-Day
HVAC	Heating Ventilating Air Cond.
LCCA	Life Cycle Cost Analysis
NDDM	New Degree Day Method
PBP	Pay Back Period
SDDM	Standard Degree-Day Method
WMRP	Wall-Mounted Radiant Panel
MPR	Multiple Polynomial Regression

INTRODUCTION

It is a fact that energy consumption is one of the world's biggest problems since energy need is increasing proportionally to the population and conventional sources are diminishing evenly. As a result of this, the International Energy Agency (2013) predicts an increase in global energy consumption by 56% from 2010 to 2040. For the present, fossil fuels are used as major energy sources but they will not able to meet energy requirement in the near future. Thus, it is important to ensure energy efficiency when using fossil fuels and place an emphasis on finding alternative energy solutions. Energy management and efficiency will be an important matter in the coming years. Therefore developing systems which promote energy saving is inevitable. Although the economy has been growing gradually in Turkey as well as energy demand and energy policy is heavily dependent on imported energy. The government invested 25$ billion in energy production between the years 2002 and 2011, but still Turkey has to import 71% of energy needs. According to the Energy and Natural Resources Ministry of Turkey (2013) 31% of Turkey's energy is being used in buildings. This high percentage is due to the fact that most buildings do not meet general energy efficiency criteria such as thermal insulation requirements for external walls. Thermal insulation is an easy and applicable method to increase energy efficiency by means of decreasing the heat flux from indoor to outdoor and vice versa (Çomaklı and Yüksel, 2003).

Insulation thickness is a parameter which balances investment and operational cost. In the literature, there are many studies which have been investigating how to determine optimum insulation thickness (Yıldız et al., 2008; Dikmen, 2011; Bolattürk and Dağıdır, 2013; Kaynaklı, 2013; Kaya et al., 2016; Duman et al., 2015). Çomaklı and Yüksel (2004) used life cycle cost analysis (LCCA) based on the degree day method for calculation of optimum insulation thickness and annual energy savings of some cities from 4th climatic region of Turkey and also they discussed the subject from an environmental point of view. Optimum insulation thickness of Denizli region for different fuel types and different insulation materials was obtained by Dombaycı et al. (2006) by using the standard degree-day method. Differently, Arslan and Köse (2006) also took into account the effect of condensed vapor within the standard degree day method. Further, Sisman et al. (2007) calculated optimum insulation thickness of roofs for different degree-day regions of Turkey. Kaynaklı (2008) chose Bursa as a model city and evaluated residential energy requirement for heating season and calculated optimum insulation thicknesses for different types of fuels. Bolattürk (2008) calculated the optimum insulation thickness using his method and compared the results with the standard heating degree-hour method. Ucar (2010) determined optimum insulation thicknesses for four different climatic regions of Turkey by using exergy analysis method. Optimum insulation thicknesses and energy savings were also studied by Ucar and Balo (2010) for different regions of Turkey. Ozkan and Onan (2011) considered effects of glazing areas on the optimum insulation thickness. Ozel (2011) determined optimum insulation thickness by using a dynamic method. Kaynaklı (2012) reviewed the existing studies with focusing on reported optimum insulation thickness results. Ekici et al. (2012) calculated optimum insulation thickness using different wall structures and fuels for different regions of Turkey. De Rosa et al. (2014) evaluated energy demand by a method which combines dynamic model based on the lumped capacitance approach and electrical analogy method.

Radiant heating systems are different from typical HVAC systems because they heat surfaces rather than air and can save large amounts of energy while providing higher levels of thermal comfort. The radiant heating system consists of large radiant heat transfer surfaces can be installed on room walls, floors or ceilings. A conditioned surface is called as a radiant system if 50% or more of the designed heat transfer on the temperature-controlled surface takes place by thermal radiation.

Radiant heating systems are quite convenient alternatives to the traditional HVAC systems. They reduce energy consumption because of low-temperature heating and high temperature cooling operations. In the literature heat transfer, thermal comfort performances and energy efficiency capabilities of these kinds of systems have been studied in detailed (Kilkis, 2006; Tye-Gingras and Gosselin, 2012; Seyam et al., 2014; Bojic et al., 2015; Rehee and Kim, 2015; Jeong et al., 2013; Stetiu, 1999; Franc, 1999; Miriel et al., 2002; Koca et al., 2016; Koca et al., 2014; Koca et al., 2013; Koca, 2011; Erikci Çelik et al., 2016; Kanbur et al., 2013; Acikgoz and Kincay, 2015). Therefore this proven technology should be disseminated in Turkey to achieve energy efficiency goals of the country.

In the literature, not many studies available dealt with the optimization of insulation thickness for radiant heating cooling systems. There is only one study (Cvetkovi and Bojic, 2014) available in the literature, in which the investigators used Energy Plus© software to

evaluate the energy consumption of the simulate building. They reported that radiant wall insulation requires a higher insulation thickness when compared with other radiant systems. Moreover the thickness of thermal insulation is the highest for the location where the radiant panels are located. The house with the optimal thermal insulation thickness has significant energy saving compared to house with older customary thermal insulation (Cvetkovi and Bojic, 2014). But in their study, they conducted the simulations according to Serbian climate conditions without any experimental validation. Moreover, they did not compare the classical methods with their newly reported results.

In most of the aforementioned studies researchers dealt only with theoretical examination using either degree-day or hourly-based methods without any experimental validation. As a result of this the results of these studies are not valid for the radiant heating systems, since the indoor air temperatures are taken into account as a heat source (or base temperature). In radiant heating systems there is no indoor heat loss through the walls where the radiant heating panels are mounted. In such cases heat loss occurs through the panel backside surfaces. Therefore, in this study we propose a new method to evaluate the optimum insulation thicknesses of radiant wall heating systems that takes into account the radiant panel backside temperature as a heat source where the huge amount of heat leakage occurs. For this reason, our first goal is to evaluate optimum insulation thicknesses for radiant wall heating systems since the performance of radiant heating systems is significantly affected by the insulation capabilities and has a great importance in the sizing process of the radiant systems. In the new method important parameters, that were obtained from a series of the CFD analysis for different thermal transmittance coefficient (U) and outdoor air temperature (To) values, are used to create an empirical correlation for the estimation of Tp (the new base temperature) with multiple polynomial regression method. A new experimental test set-up which simulates typical conditions of occupancy in an office or residential room was used for the validation of the computational results. Then serial CFD analyses were conducted to create a new correlation for the estimation of the base temperature (Tp) with multiple polynomial regression method (MPR). Based on the new base temperature, LCCA based Standard Degree-Day Method (SDDM) is adjusted for Wall-Mounted Radiant Panel (WMRP) and created a New LCCA based Degree-Day Method (NDDM). Using the new method optimum insulation thickness, net energy saving and payback periods for radiant wall heating systems were calculated for the city of Istanbul and compared with the results which were obtained using the standard degree-day method (SDDM).

NUMERICAL MODEL

In this work, a numerical model was developed using the commercial CFD package ANSYS-FLUENT© Version 15 to simulate the WMRP. Fluent uses a control-volume-based technique to convert an inclusive scalar transport equation to an algebraic equation that is solved numerically. The steady simulations were performed with the software, using the coupled double precision solver.

Physical Model

A 3D model that has the same dimensions as the experimental set-up (1.2 m in height, 0.6 m in length) was used for the simulations. The pipe in the panel which is made of cross-linked polyethylene (PEX) has 10.1 mm outer diameter and 1.1 mm thickness with 150 mm pipe spacing. The gypsum board which is exposed to the inside room has 1.5 cm thickness. EPS (Expanded Polystyrene) was used as an insulation material which has a coefficient of thermal transmittance value of 0.039 W/m.K (at 25˚C). To decrease the mesh number and precisely solve the heat loss from the WMRP, experimental room domain wasn't considered in our numerical model. Instead, average total heat transfer coefficient – comprised of the radiation and convection – was implemented as a surface boundary condition. The implemented average total heat transfer coefficient was obtained experimentally (using the same test chamber in this study) in our previous study (Koca et al., 2014) using different WMRP surface temperatures and which are valid for wide range (25 - 45 °C) of WMRP surface temperature. According to the results of Koca et al. (2014), the measured average values of radiant heat transfer coefficient is about 5.46 W/m^2.K and convection heat transfer coefficient is about 2.32 W/m^2.K resulting in an average total heat transfer coefficient of 8.33 W/m^2.K – which was considered in this study. Obtained average total heat transfer coefficient for wall (8.33 W/m^2.K) is compatible with the ones typically shown in standards of EN 15377-1 (2008) and EN 1264-5 (2008).

## Computational Domain, Mesh and Mesh Independency Analysis

The computational mesh was generated using tetra and hexahedral elements with ANSYS Meshing tool. In order to accurately resolve the solution fields in the vicinity of the heat transfer surface between pipe and the surrounding interface, the mesh was refined at the area where the heating pipes are embedded. The grid was fine enough for the other areas to solve the simple heat conduction problem. A sample of the computational model and grid was shown in Fig. 1.

Before the computations, a grid independence study was carried out to ensure the results' accuracy. Eight different mesh configurations (varies from 1M to 27M) were analyzed for same boundary conditions and the calculated panel backside surface temperature (T_p) – which is our main parameter – was compared. As shown in Fig. 2, at the range between the first and third configurations of the results slightly vary with the grid resolution but after the fourth mesh configuration, T_p tends towards constant. So after that point the results can be considered grid independent. In regard to this mesh independence results, simulations were carried

out with the mesh configuration which has 6.5M elements and average skewness value of 0.20.
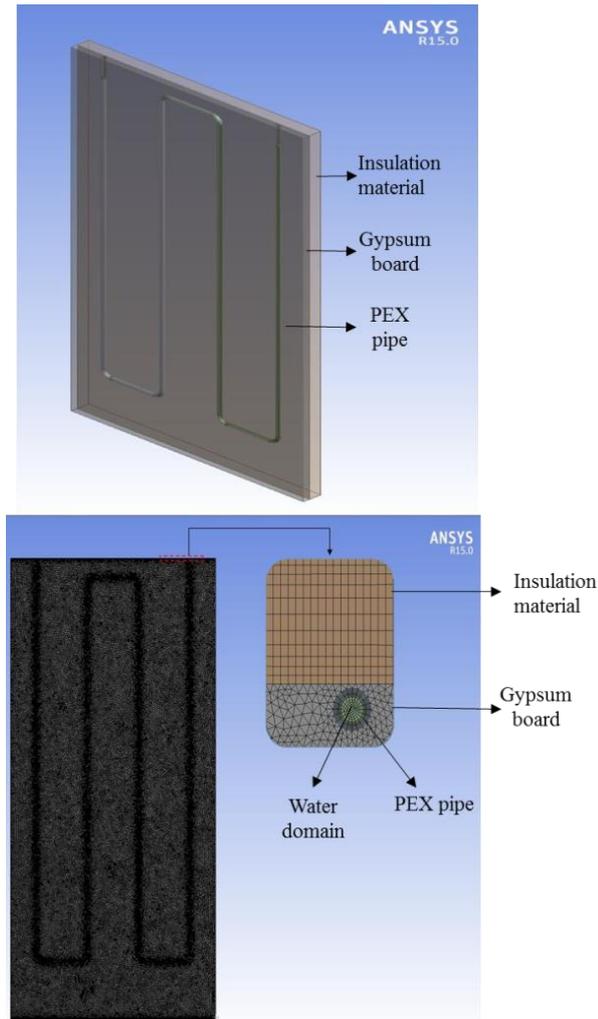
Figure 1. CAD model of the panel (left) and computational mesh grids (right)
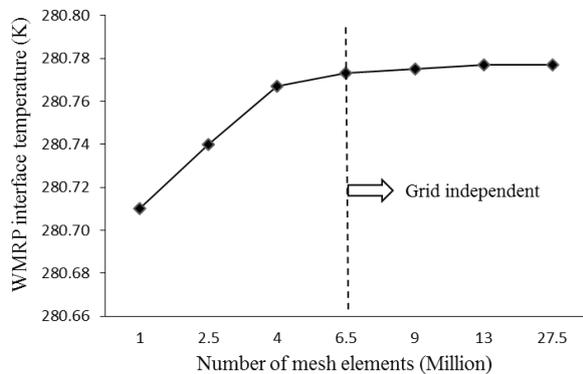
Figure 2. Mesh independence study

Solution Methods and Procedure

3D, steady-state, CFD analysis were performed by using ANSYS FLUENT 15©. Mass, momentum and energy conservation equations were solved numerically. As main boundary condition, WMRP surface was set as mixed (radiation and convection) heat transfer surface, where the average total heat transfer coefficient (sum of radiation and convection) was defined as the value of 8.33 $W/m^2.K$ (radiant and convective heat transfer coefficients were obtained from our previous work done by Koca et al., 2014). The other side walls were set as adiabatic boundary conditions. The backside of the panel was set to the convection surface having the certain value of total heat transfer coefficient of 25 $W/m^2.K$.

The outside air temperature of the WMRP – which is exposed to outside conditions – was set at certain values and studied as a parameter in the simulations. The indoor room temperature (T_i) was defined constant as 15°C for all cases. Inlet water mass flow rate was defined as constant at the value of 0.04 kg/s (Reynolds number is 4200). Therefore a turbulence model (Realizable k-ε model) was chosen for the calculations. Also enhanced wall treatment was implemented for the heat transfer surface where the y^+ value is varied between 1 and 5. Among the different code options, Second Order Upwind law interpolation scheme and the discretized equations were chosen and numerically solved by the SIMPLE algorithm. In the present work, all the solutions were considered to be fully converged when the sum of residuals was below 1×10^{-4}.

Multiple simulations were conducted by varying outdoor air temperature (T_o) and insulation thickness (Δ_p) of the WMRP. Outdoor air temperature values between -15 °C and +18 °C (34 different outdoor temperatures increased numerically between -15 °C and +18 °C within a value of around 1 °C) were set as a boundary conditions of the WMRP while the insulation thickness of the panel was varied between 0 - 16 cm (16 different insulation thickness with 1 cm variation). On the basis of the obtained simulations, the backside surface temperatures of WMRP (T_p) were evaluated using area-weighted-average method (cell-centered) and the results are presented in Table 1.

In the table, values of total U (overall coefficient of thermal transmittance of WMRP) and outdoor air temperature (T_o in °C) are the input parameter, WMRP backside temperature (T_p in °C) is the output parameter of the simulations.

Table 1. Result summary of the numerical simulations

T_o	U_{NDDM}	x	T_p	T_o	U_{NDDM}	x	T_p	T_o	U_{NDDM}	x	T_p	T_o	U_{NDDM}	x	T_p
-15	0.23	16	20.7	2	0.23	16	23.1	18	0.23	16	25.8	1	0.23	16	22.9
-14	0.24	15	20	3	0.24	15	23	17	0.24	15	25.6	0	0.24	15	22.5
-13	0.26	14	19.8	4	0.26	14	23	16	0.26	14	25.3	-1	0.26	14	22.1
-12	0.28	13	19.6	5	0.28	13	22	15	0.28	13	24	-2	0.28	13	21.6
-11	0.30	12	19.4	6	0.30	12	22.9	14	0.30	12	24.6	-3	0.30	12	21.1
-10	0.32	11	19.1	7	0.32	11	22.9	13	0.32	11	24.2	-4	0.32	11	20.5
-9	0.35	10	18.8	8	0.35	10	22.8	12	0.35	10	23.8	-5	0.35	10	19.8
-8	0.38	9	18.5	9	0.38	9	22.8	11	0.38	9	23.3	-6	0.38	9	19
-7	0.42	8	18.1	10	0.42	8	22.7	10	0.42	8	22.7	-7	0.42	8	18.1
-6	0.47	7	17.6	11	0.47	7	22.6	9	0.47	7	22	-8	0.47	7	17
-5	0.53	6	17	12	0.53	6	22.5	8	0.53	6	21.2	-9	0.53	6	15.8
-4	0.61	5	16.4	13	0.61	5	22.4	7	0.61	5	20.3	-10	0.61	5	14.3
-3	0.72	4	15.7	14	0.72	4	22.3	6	0.72	4	19.2	-11	0.72	4	12.6
-2	0.88	3	14.7	15	0.88	3	22.1	5	0.88	3	17.7	-12	0.88	3	10.3
-1	1.12	2	13.5	16	1.12	2	21.9	4	1.12	2	15	-13	1.12	2	7.6
0	1.56	1	11.9	17	1.56	1	21.6	3	1.56	1	13.6	-14	1.56	1	3.9
1	2.55	0	9.8	18	2.55	0	21.3	2	2.55	0	10.3	-15	2.55	0	-1.2

METHODOLOGY

A New Degree-Day Calculation Method for Wall-Mounted Radiant Panels

In (SDDM), degree-day value (DD) is calculated using inner and outer air temperatures for a fixed base temperature. This method cannot be applied for buildings heated by WMRP due to the fact that WMRP has different heat transfer characteristic comparing to the conventional systems. Because in these kinds of systems, heat source is part of the wall structure and this causes two-way conduction in the structure. However, as shown in Fig. 3, the temperature gradient from inner surface temperature (T_{si}) to outer air temperature (T_o) in conventional systems is similar to the temperature gradient (from backside temperature, T_p to outer air temperature, T_o) in radiant systems. Therefore, a correlation for the evaluation of T_p which is based on the T_o and overall heat transfer coefficient of wall (U) was obtained using a numeric and statistical methods.
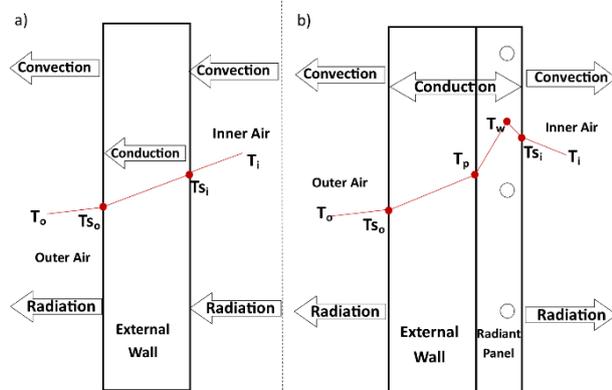
Figure 3. Heat transfer through external wall in a) conventional system b) WMRP system

Moreover base temperature (T_i in Fig. 3) was defined as 15°C for the calculation of degree-day value in SDDM whereas internal heat sources (human, electronic devices etc.) exist. Eqs. 1 and 2 show the general calculation procedure of the DD value:

$$f(D) = \begin{cases} 0 \,, & T_b < T_o \\ T_p(U, T_o) - T_o \,, & T_b \geq T_o \end{cases} \tag{1}$$

$$DD = \sum_{D=1}^{N} f(i) \tag{2}$$

25-year averages of daily air temperatures were used to calculate the degree-day value and the degree day value is a series sum of day 1 to day 365 as a function of T_p. Also T_p is a function of x, which was obtained from regression of CFD results.

Multiple Polynomial Regressions

Multiple polynomial regressions, is a statistical approach for modeling of the relationship between a dependent variable and independent variables. WMRP backside temperature (T_p) values were obtained from series of the CFD analysis for different thermal transmittance coefficient (U) and outdoor air temperature (T_o) values were used to create an empirical equation for the estimation of T_p with multiple polynomial regression method (MPR) where the Newton-Raphson method was implemented. In MPR, 3rd degree of the polynomial equation (Eq. 3) was chosen in which the results were best yielded (correlation coefficient (R^2) is 0.998 and Residual Sum of Squares (RSS) is 0.74).

$$T_p = A T_o^3 + BT_o^2 U + CT_o U^2 + DU^3 + ET_o^2 + FT_o U + GU^2 + HT_o + IU + J \tag{3}$$

$$DD = \sum_{D=1}^{N} A T_o^3 + BT_o^2 U + CT_o U^2 + DU^3 + ET_o^2 + FT_o U + GU^2 + (H-1)T_o + IU + J \tag{4}$$

The constant coefficients of the above equations are given in Table 2.

Table 2. Coefficients of the polynomial equation

A	B	C	D	E
$-1.40 \cdot 10^{-4}$	$-2.36 \cdot 10^{-4}$	$-9.61 \cdot 10^{-2}$	-1.27	$1.16 \cdot 10^{-1}$
F	**G**	**H**	**I**	**J**
$6.16 \cdot 10^{-1}$	34	-32	-169.1	3223.15

Calculation of Optimum Insulation Thickness Using New and Standard Degree Day Methods

Eqs. 5-6 and Eqs. 9-15 are common for both new and standard methods. Calculation method of overall heat transfer coefficient (U) is similar both in NDDM and SDDM. Nevertheless in NDDM inner heat transfer coefficient (h_i) and thermal resistance of panel (R_p) are not included into the equation since these parameters have already been taken into account in the CFD calculations and embedded into the proposed equation of T_p (Eq. 3).

The heat loss per unit area of external walls is given by:

$$Q = U(T_b - T_o) \tag{5}$$

Annual heat loss per unit area from external walls (q'') in the terms of degree-days is given by:

$$q'' = 86400 \cdot DD \cdot U \tag{6}$$

Overall heat transfer coefficient (U) and equivalent thermal resistance of wall (R_{tw}) were calculated using the Eqs. 7-8. Whereby, Eq. 8a and Eq. 8b were used to calculate the U values of interest for SDDM and NDDM respectively:

$$R_{tw} = R_w + R_{ins} + R_i + R_o \tag{7}$$

$$U_{SDDM} = \frac{1}{R_w + \frac{x}{k} + \frac{1}{h_o} + \frac{1}{h_i}} \tag{8a}$$

$$U_{NDDM} = \frac{1}{R_w + \frac{x}{k} + \frac{1}{h_o}} \tag{8b}$$

Where, R_o and R_i are the inside and outside air film thermal resistance, R_w is the total thermal resistance of the wall associated with the structural components of the wall and panel, h_i (W/m^2.K) is the inner, h_o (W/m^2.K) is the outer convective heat transfer coefficients, x is the insulation thickness (m) and k (W/m.K) is the heat conduction coefficient of insulation material.

Annual energy requirement (E_A) was calculated by the Eq. 9 and it corresponds to annual heat loss.

$$E_A = m_f \cdot H_u \cdot \eta = \frac{86400 \cdot DD}{\left(R_{tw} + \frac{x}{k}\right)} \tag{9}$$

Where, m_f (kg) is the annular fuel mass consumption, H_u (j/kg) is lower heating value of the fuel and η (%) is the efficiency of the heating system.

Then, the annular fuel mass consumption (m_f) was obtained by dividing the annual heat loss (E_A) by lower heating value (H_u) and efficiency (η) of fuel, yields:

$$m_f = \frac{86400 \cdot DD}{\left(R_{tw} + \frac{x}{k}\right) \cdot H_u \cdot \eta} \tag{10}$$

If we multiply annular fuel mass consumption (m_f) with the fuel cost C_f (TL/kg), we get annular heating cost per unit area (C_A):

$$C_A = m_f \cdot C_f \tag{11}$$

$$C_A = \frac{86400 \cdot DD \cdot C_f}{\left(R_{tw} + \frac{x}{k}\right) \cdot \eta \cdot H_u} \tag{12}$$

The LCCA used in this paper calculates the heating cost over the lifetime of the building. The total heating cost over a lifetime of N years is estimated in present value Turkish Liras using the PWF (Present Worth Factor). The PWF depends on the inflation rate g, and the interest rate i. The PWF, which is dependent on the inflation rate g and the interest rate i, is adjusted for inflation rate as shown below (Hasan A., 1999). The interest rate modified for inflation rate, i^*, is defined by the following equations:

$$i^* = \frac{i-g}{1+g} \qquad (i>g) \tag{13}$$
and

$$PWF = \frac{(1+i^*)^N - 1}{i^*(1+i^*)^N} \tag{14}$$

Where N is the lifetime and which is assumed to be 20 years.

Total cost (C_t) was calculated by multiplying the Present Worth Factor (PWF) into annual heating cost (C_A) and adding the total insulation cost per unit area ($C_{t_{ins}}$).

$$C_t = \frac{86400 \cdot DD \cdot C_f \cdot PWF}{\left(R_{tw} + \frac{x}{k}\right) \cdot \eta \cdot H_u} + C_{t_{ins}} \tag{15}$$

The cost of investment of the insulation material is given by the following equation:

$$C_{t_{ins}} = C_{ins} \cdot x \tag{16}$$

Where, C_{ins} (TL/m^2) is the cost of insulation per unit insulation area. Therefore, the following equation gives the total cost of heating of insulated building in present sum of Turkish Liras:

$$C_t = \frac{86400 \cdot DD \cdot C_f \cdot PWF}{\left(R_{tw} + \frac{x}{k}\right) \cdot \eta \cdot H_u} + C_{ins} \cdot x \tag{17}$$

Then, the optimum insulation thickness was obtained by minimizing the total heating cost (C_t). Therefore, the derivative of C_t with respect to the insulation thickness (x) was taken and set equal to zero, from which the optimum insulation thickness (x_{op}) values were derived for SDDM (Eq. 18a) and NDDM (Eq. 18b) as follows:

$$\frac{dC_t}{dx} = C_{ins} - \frac{86400 \cdot DD \cdot C_f \cdot PWF \cdot k}{\left(k \cdot R_{tw} + x\right)^2 \cdot \eta \cdot H_u} \tag{18a}$$

$$\frac{dC_t}{dx} = \sum_{D=1}^{N} C_{ins} - \left(\frac{86400 \cdot PWF \cdot C_f \cdot U_{NDDM}^3}{k \cdot H_u \cdot \eta} \cdot \left(4 \cdot D \cdot U_{NDDM}^2 + 3 \cdot U_{NDDM} \cdot \left((C \cdot T_o(i)) + G \right) + 2 \cdot \left(I + \left(F \cdot T_o(i) \right) + B \cdot T_o(i)^2 \right) + \frac{A \cdot T_o(i)^3 + E \cdot T_o(i)^2 (H-1) \cdot T_o(i) + J}{U_{NDDM}} \right) \right)$$

(18b)

Eqs. 18a and 18b are different in both methods since in the NDDM T_p is a function of U (also U is a function of x). Hence, derivation results of the Eq. 19 are different for NDDM than SDDM.

Despite of the aforementioned differences in the previous steps, the methodology of the last two steps is common for both SDDM and NDDM. A root of Eq. 19 which is the minimum point of the Eq. 18, gives the optimum insulation thickness values.

$$\frac{dC_t}{dx} = 0$$

(19)

For the SDDM, the derivative of the Eq. 18a gives the optimum insulation thickness value, which is obtained as follows:

$$x_{opt} = \sqrt{\frac{86400 \cdot DD \cdot C_f \cdot PWF \cdot k}{C_{ins} \cdot \eta \cdot H_u}} - k \cdot R_{t_w}$$

(20)

In Eq. 18a, DD value is constant for SDDM, while the DD value in Eq. 18b for NDDM is a function of U_{NDDM} and accordingly function of x. For this reason in NDDM, optimum insulation thickness values were obtained using the Matlab© code because of the complicity of the dependent variables in the Eq. 18b. In the code, to sum up the series, T_o values were defined as a matrix. Since only the value of x is independent variable in the equation, the solution matrix was equalized to zero and then the values of x obtained.

Parameters used in the optimization calculations are shown in Table 3.

Table 3. Parameters used in the optimization of insulation thickness

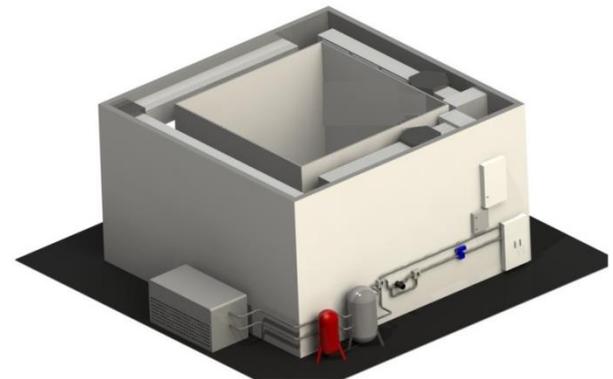| Parameters		
Interest rate, i	%	8 %
Inflation rate, g	%	7.49 %
Life cycle, N	yr	20
Present Worth Factor, PWF	-	14.99
h_0	W/m².K	25
C_f (N. Gas)	TL/kg	1.003
C_f (Lignite)	TL/kg	0.38
H_u (N. Gas)	j/kg	3.46 x 10⁷
H_u (Lignite)	j/kg	2.10 x 10⁷
η_u (N. Gas)	%	93
η_u (Lignite)	%	65
C_{EPS}	TL/m²	300
k_{EPS}	W/m.K	0.4

EXPERIMENTAL VALIDATION

The Arrangement of the Test Chamber

The climatic test chamber was constructed to simulate radiant wall heating system under various boundary conditions which are listed in Table 4. As seen in Fig. 4 and Fig. 5, the test chamber is composed of four zones: Ceiling (volume-4), floor (volume-3), façade (volume-2) and the tested zone (volume-1). WMRP were mounted into the tested zone which is characterized by a floor area of 16 m² (4 m × 4 m) and an internal height of 3 m as recommended in EN 1264-5 (2008) and BS EN 14037-5 (2016).

The wall types were chosen as the sandwich type panel with polyurethane insulation between two layers is made out of sheet steel which has engagement and locking mechanism to increase the strength. The coefficients of thermal transmittance of the wall and ceiling were decided according to Turkish Standard TS 825 (thermal insulation requirements for buildings) (2008). The enclosed volumes were conditioned with mechanical air conditioners to ensure relevant boundary conditions in the tested volume. The emissivity of the indoor wall and WMRP surfaces, were estimated by the use of an infrared thermal imaging camera and calibrated thermocouples.

First, the surface temperatures were obtained by using precise temperature sensors, and then the surface emissivity was changed in the pyrometer setup in order to get the same temperature of the analyzed surface as obtained before by the use of the temperature sensors (Olesen et al., 2000). The physical dimensions and the thermo-physical properties of the test chamber and the testing equipment meet the general requirements of the ANSI/ASHRAE Standard 138 (2005). The main difference from the standard is; to ensure appropriate boundary conditions in the tested volume, the inner surfaces in the test zone were conditioned through the surrounded volumes as they are equipped with a mechanical air conditioner whereas the ANSI/ASHRAE Standard 138 (2005) recommends directly conditioning the testing room through pipes embedded in the surfaces.

Figure 4. General view of the test room

**Table 4.** Controlled parameters in the zones

	Ceiling	Floor	Facade	Tested zone
Temperature Range	-10°C / +40°C	+0°C / +30°C	-10°C / +40°C	+0°C / +30°C
Temperature Tolerance	± 0.5 °C	± 0.5 °C	± 0.5 °C	± 0.5 °C
Humidity Range	n/a	n/a	%35 / %85 RH	n/a
Humidity Control Steps	n/a	n/a	%1	n/a
Humidity Tolerance	n/a	n/a	± % 0.5 RH	n/a
Air Velocity	n/a	n/a	0.5 – 5 m/s	n/a
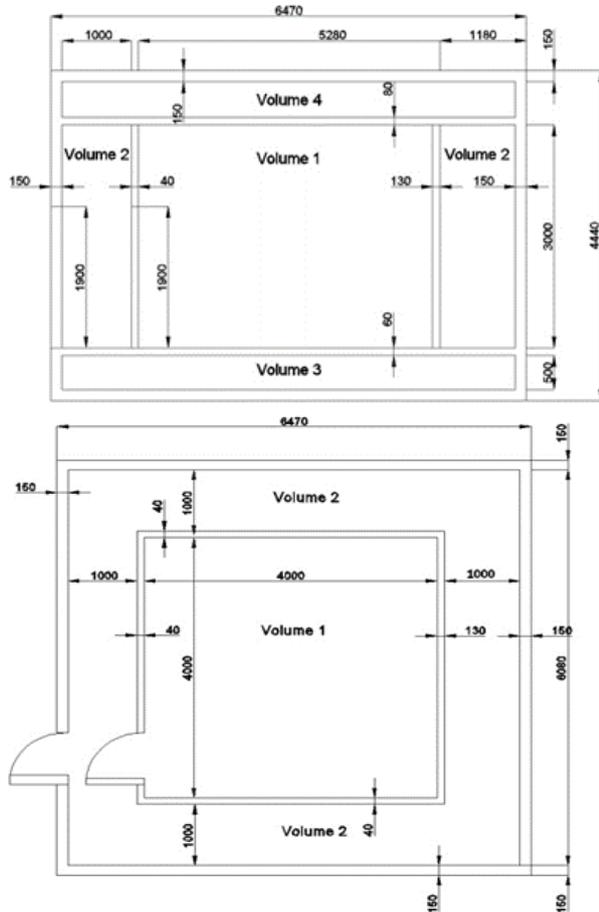
Figure 5. Dimensions of the test chamber (all units are in millimeters)

The Wall Mounted Radiant Panel

The WMRP's were manufactured for this study consist of three layers which are gypsum board, serpentine heating pipe and insulation material; from inner to outer layers. The pipe serpentine was inserted into the gypsum board. The thickness of the gypsum board is 15 mm while the panel insulation thickness is 30 mm. The serpentine has cross-linked polyethylene (PEX) pipes with a 10.1 mm external diameter and 55 mm pipe spacing. Expanded polystyrene (EPS) was used as a backside thermal insulation material which has a coefficient of thermal transmittance value of 0.040 W/m.K (at 30 °C). WMRP has the standard insulation thickness which is attached during its manufacturing process. It should be noted that, obtained optimum thickness values (x_{opt}) in this study are an additional insulation to be attached to the external walls.

The general dimension of the WMRP was 1.2 m in height and 0.6 m in width (same as the CFD model). The test chamber was configured with six WMRP panels which are shown in Fig. 6 but measurements were performed from one of them. Six WMRP panels were attached to the wall instead of attaching single panel, this more closely simulates typical application allowing for more realistic results.
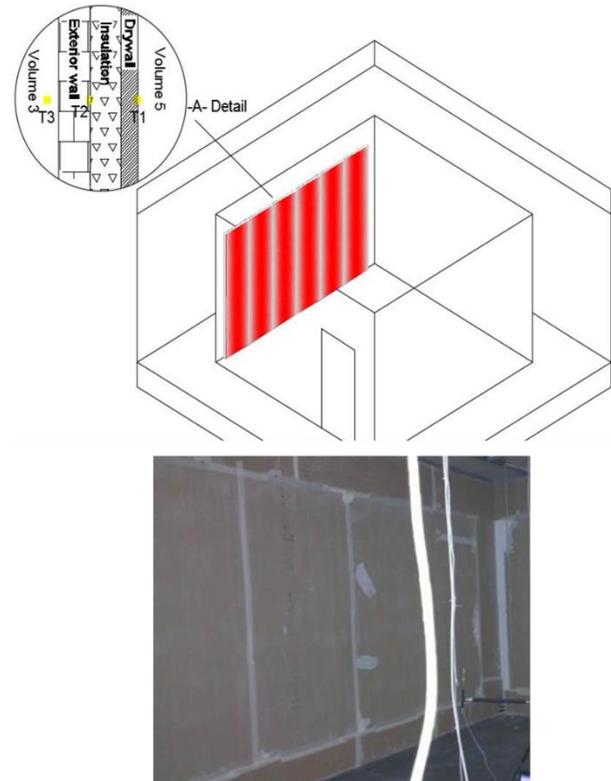
Figure 6. Arrangement of the wall mounted radiant panel

Hydraulic Circuit of WMRP

A water conditioning system was attached to the test chamber. As shown in Fig 7, inlet water accesses the hydraulic line through the buffer tank (the water temperature is maintained by means of electrical resistances), it then comes to the four-way valve. Here, the four-way valve was placed to provide a mixture through the supply and return lines. The mixed water leaves the four-way valve such that it is equal to the desired WMRP inlet temperature and enters the pump to supply the needed pressure. Then, the water comes to the three-way valve where the mass flow rate of the water is maintained at precisely appropriate flow rate. After that, the water passes through the ultrasonic flow meter, where the volumetric flow rate was measured. The data

for the flow rate control was provided from an electromagnetic flow meter. Following the flow meter, the water goes through manifolds and then the WMRP facility to activate the heat transfer mechanism. After finishing the cycle in the hydraulic line, the fluid comes to the four-way valve again through the return pipeline it is mixed with the water that comes from the buffer tank if needed (to adjust the required temperature of the fluid precisely). The pipes in the water circuit are well insulated.
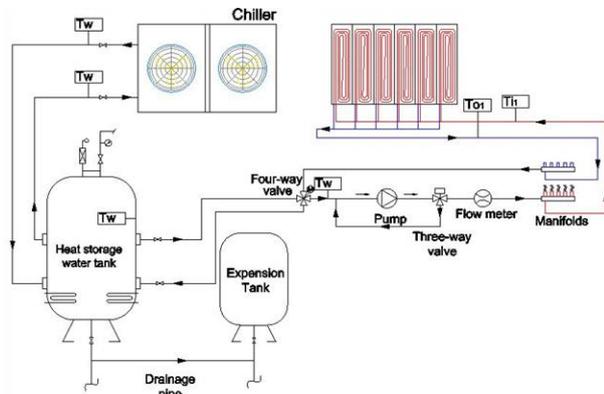

Figure 7. Hydraulic circuit of the test system

The Measurement Equipment and Experimental Method

The main objective of the experimental study was to validate the numerical simulation results. The most reliable analyses are those based on the measurements performed in the full scale test chamber which was described in detailed above. The measurements were carried out for the chosen representative WMRP, where the temperature transducers were inserted as described in the Fig. 3-b. For all cases, the supply water mass flow rate (0.04 kg/s) and the other environmental conditions were fixed at desired conditions. The measurements were carried out under steady conditions for variable outdoor (volume-2) air temperatures (T_o). The water conditioning system was turned on to achieve the desired steady state and initial conditions and heat flow throughout the heated WMRP before collecting the experimental data. Steady state conditions were ensured after about 3-4 h in which supply water temperature and water flow rates, surface temperature of the WMRP (T_{si}), outdoor surface temperature (T_{so}), WMRP backside surface temperature (T_p), outdoor air temperature (T_o) and indoor air temperature (T_i) were nearly constant, only then the tests were begun.

Indoor and surrounding volumes' air temperatures (T_i, T_o), the related surface temperatures (T_p, T_{si}, T_{so}) water mass flow rate, supply and return water temperatures were measured, controlled and stored for each measuring time interval (1 minute). Average test duration took around 8 h so that all important variables reached desired and steady state conditions. The temperatures of T_i, T_o, T_p, T_{si}, T_{so} were evaluated after the system reached a steady state condition (which was characterized by physical properties that were unchanging in time). Results corresponding to average values were stored during the periods of at least 30 min in which stable conditions were observed.

Multiple tests were carried out by varying the outdoor air temperature (T_{o1} = -3 °C, T_{o2} = 3 °C, T_{o3} = 5 °C) in Volume-2. When the air and surface temperatures bands changed less than 0.1 K/min, measurements were started. In terms of the recorded measurements, the panel backside surface temperatures (T_p) were obtained and compared with the numerical results.

RESULTS AND DISCUSSION

The optimum insulation thickness (x_{opt}) values which were calculated by two different methods (NDDM, SDDM) for different fuels in Istanbul climate, are shown in Table 5 and described below.

According to the general outcomes, optimum insulation thickness values which were calculated by NDDM are higher than the ones calculated by SDDM. Because, in SDDM indoor air temperature is used as a base temperature and heat production in the walls is neglected for the buildings heated by WMRP. Therefore, calculated optimum insulation thickness values from SDDM for the walls where the WMRP is mounted will be insufficient from the point of energy saving. On the other hand, optimum insulation thickness values were obtained by NDDM for a building heated by WMRP is lower than the ones obtained for standard building using the SDDM. This is because; backside insulation layer attached to the WMRP provides additional insulation to the walls, resulting in higher energy efficiency. Moreover, the payback periods (PBP) for natural gas and lignite obtained by NDDM were shorter than the ones obtained through SDDM. Accordingly, NDDM ensures better estimation of optimum insulation thickness than the SDDM, resulting in a shorter payback period.

Table 5. Summary results

	NDDM						SDDM					
	Xopt		PBP		DD		Xopt		PBP		DD	
	N.Gas	Lignite	N.Gas	Lignite	N.Gas	Lignite	N.Gas	Lignite	N.Gas	Lignite	N.Gas	Lignite
WMRP	3.7cm	3.3 cm	3.9yrs.	4.2yrs.	2686	2686	2 cm	1.7 cm	7.2yrs.	7.4yrs.	1865	1865
Standard Heating	5.2cm	4.9 cm	7.9yrs.	9.1yrs.	1865	1865	5.2cm	4.9 cm	7.9yrs.	9.1yrs.	1865	1865

Table 6. Validation of the numerical studies

Outer Air Temperature	U_{NDDM} (W/m².K)	T_p (°C)			Error (°C)		
		Experimental	CFD	MPR	CFD-Exp	MPR-CFD	MPR-Exp
-3 °C	0.72	15.9	15.6	15.4	-0.3	0.2	-0.5
3 °C	0.72	17.4	17.9	17.8	0.5	0.1	0.4
5 °C	0.72	18.4	18.7	18.6	0.2	0.1	0.2

Experimental Validation

The numerical model was validated with the experimental results conducted in same conditions. WMRP backside temperatures (T_p) calculated from the simulations were compared with the experimentally evaluated results. Because T_p is the most important, new parameter and affects the NDDM results, this validation of the T_p ensures the accuracy of the numerical results. However to increase the accuracy of the numerical results, comparison study was done for three different outer air temperatures (T_o).

The comparison summary of the numerical and experimental results is shown in Table 6.

According to the comparison results of the simulated and measured T_p values; average deviation is 0.3 °C (~ 1%) which is quite low. The maximum deviation (0.5 °C) was seen in the case where the outer air temperature (T_o) value is about 3 °C. Moreover MPR and CFD results are compatible to each other and average deviation is about 0.15 °C yielded.

Comparison of the NDDM and SDDM Results

Fig. 8 shows the variation of unit cost with respect to the insulation thicknesses (for the fuels of natural gas and lignite) that were obtained using the new (NDDM) and standard methods (SDDM). According to the results, as insulation thickness increases the heating load and accordingly heating cost decreases and vice-versa; when insulation thickness decreases, the heating load and heating cost increases. Furthermore, when the insulation thickness increases, cost of insulation also increases proportionally. When these two curves added together a total cost curve is obtained and the minimum value of this curve gives the optimum insulation thickness (x_{opt}). It can be seen in the Fig. 3a that for the walls where the WMRP is integrated, optimum insulation thicknesses for the usage of natural gas were obtained as 3.7 cm and 2 cm according to the NDDM and SDDM respectively. These values are 3.3 cm and 1.7 cm for the fuel of lignite (Fig 3b). Beyond these values, increasing the thickness of the insulation also increases the total cost. Table 5 shows the optimum insulation thickness results for the Istanbul for the new and standard methods described above.

With respect to the results obtained from both methods, significant difference (~85 %) in optimum insulation thicknesses is seen between the two methods. The main reason of such deviation is; heat source (WMRP) in the wall is not taken into account in the SDDM, while the NDDM considers the interface temperature (between wall and WMRP) as a base temperature rather than indoor air temperature as well as heat generation (as mentioned above). Thus, SDDM is not convenient to estimate optimum insulation thickness for the walls where the WMRP is implemented. Nevertheless, if one wishes to use the standard method, one should double the obtained insulation thickness result from the SDDM as a safety margin.


(a)

(b)
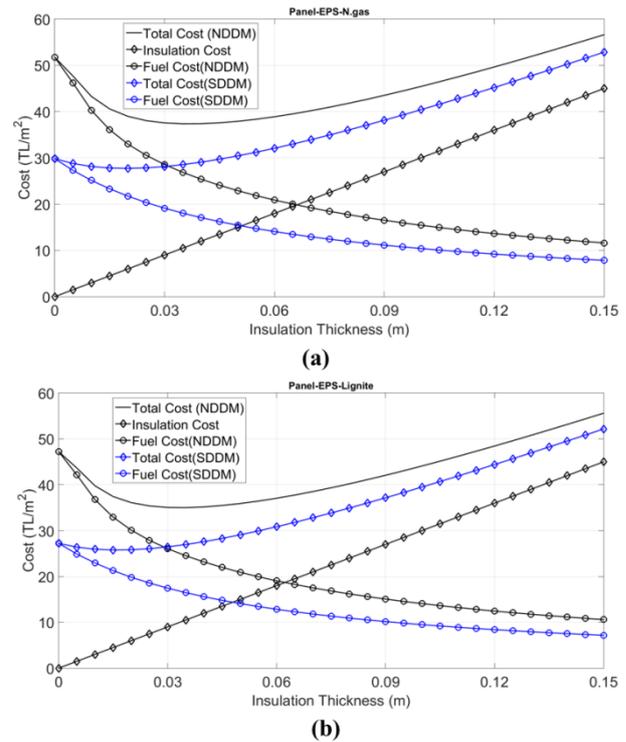
Figure 8. Comparison of the SDDM and NDDM methods for the WMRP a) Natural Gas b) Lignite

In Fig. 9, optimum insulation thickness results of SDDM for a standard building conditioned by conventional heating systems are given. As seen in the figure, optimum insulation thickness values for the fuels of natural gas and lignite are 5.2 cm and 4.9 cm respectively. If we compare the optimum insulation thickness results of conventional heating system (in SDDM) with the ones which were obtained from the case of WMRP in SDDM (corresponding to the values of 2 cm 1.6 cm), is significantly higher. Where, it can be stated that WMRP systems increase the insulation capability of buildings by means of providing extra insulation (backside insulation of the WMRP).
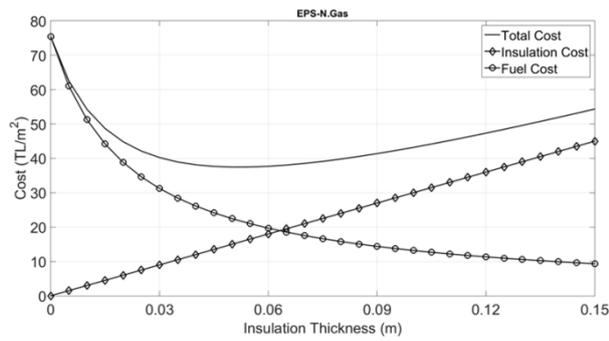
(a)
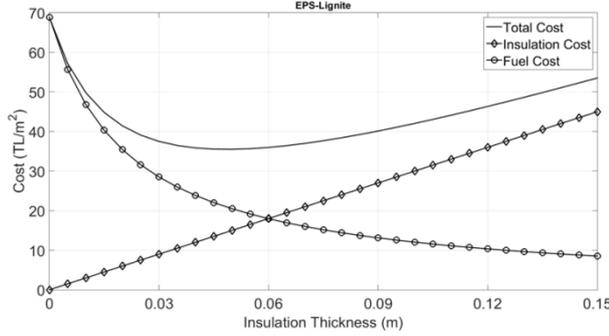
(b)

Figure 9. Optimum insulation thickness results of conventional heating systems according to the SDDM a) Natural Gas b) Lignite

Fig. 10 shows the variation of fuel consumption and net saving with respect to the insulation thickness (when the walls are insulated with EPS) which were obtained by the SDDM and NDDM for the Istanbul climate. From Figs. 10a-b it can be stated that, there is a non-linear relation between energy saving and insulation thicknesses – energy savings tend to increase quickly before the optimum point then the increment diminishes. Furthermore, when the insulation thickness increases, net savings gradually increase and reach the maximum value at the optimum thickness; following this point net savings decrease opposite of the trend of the total cost in Fig. 8. Energy savings in the new method (NDDM) is higher than the standard method (SDDM) due to the additional insulation is attached to the WMRP. In the NDDM, energy saving up to 25 TL/m^2 (average of natural gas and lignite) can be ensured with the calculated optimum insulation thickness, while the calculated insulation thickness by the SDDM provides only 7 TL/m^2. The effect of insulation thickness on fuel consumption for natural gas and lignite is shown in Figs. 10b-c. The trends are wholly opposite to the energy saving results in Fig. 10a-b. As expected, increasing the insulation thickness decreases the fuel consumption. The maximum decrease in the fuel cost is seen at the region before the optimum insulation thickness point.
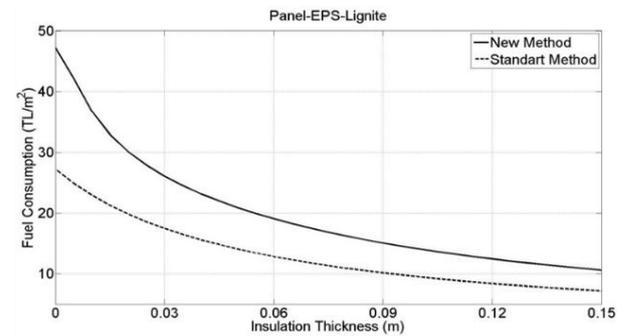
CONCLUSION

In this study optimum insulation thickness, net energy saving and payback period were calculated for the Istanbul degree-day region and for two different fuels of natural gas and lignite using the SDDM and NDDM. As
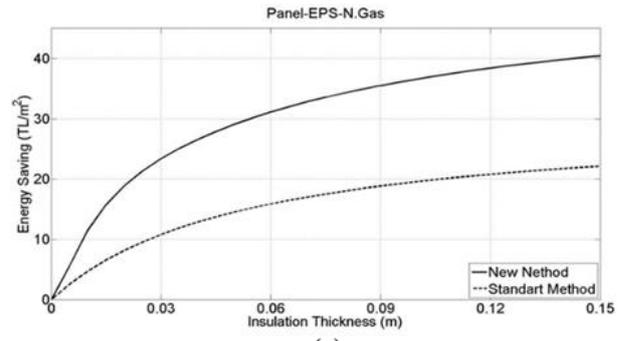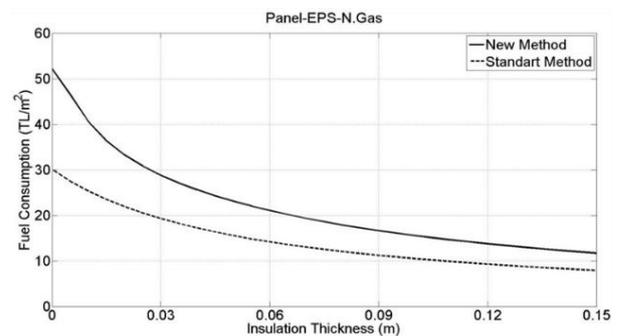

(a)
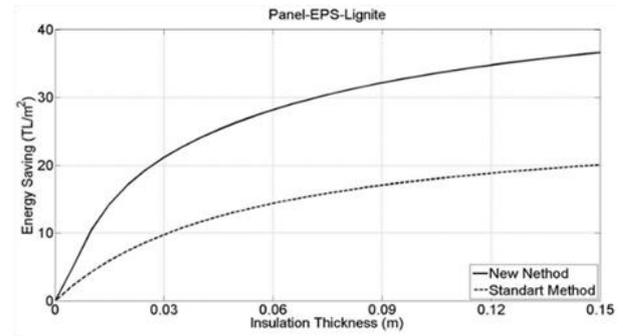
(b)

(c)

(d)

Figure 10. Total energy saving versus insulation thickness for SDDM and NDDM a) Natural Gas b) Lignite, fuel consumption comparison of the methods c) Natural Gas d) Lignite

insulation thickness increases, heating load decreases and accordingly cost of fuel decreases. At the point of optimum insulation thickness, the values of total cost of fuel and insulation material are at a minimum.

Radiant heating systems are quite convenient alternatives to the traditional HVAC systems, ensuring that efficiency requirements are met by producing more

thermally comfortable environments. The existing SDDM is used to estimate the optimum insulation thickness for the buildings where the WMRP is used. For this reason we propose new degree-day method (NDDM) in which base temperature is the interface temperature (backside temperature of the WMRP) between the WMRP and the wall structure. In this way, better estimation of insulation thickness and accordingly higher energy efficiency can be ensured. In the new method, WMRP backside temperature (T_p) was obtained from the series of the CFD analysis for different thermal transmittance coefficient (U) and outdoor air temperature (T_o) values are used to create an empirical equation for the estimation of T_p with multiple polynomial regression method. Then the obtained 3rd degree of the polynomial equation was used to calculate the T_p which is a function of U, also U is a function of x.

According to the results of NDDM; optimum thicknesses of 3.7 cm and 3.3 cm were found for the usage of natural gas and lignite respectively. These values are 2 cm and 1.7 cm for the SDDM. This result shows that, the SDDM significantly lower (85-95%) estimates the optimum insulation thicknesses and the method is not valid for the buildings where the WMRP is used. Therefore, the payback periods of SDDM are higher comparing to the NDDM. So, for the design process of radiant systems, the proposed new method is recommended since it provides an energy savings of 14 TL/m^2. In such cases, the total investment cost will be returned in the range of 3.9 - 4.2 years.

Moreover for the building where the conventional heating systems are used, SDDM was applied to calculate the optimum insulation thicknesses and the results are found as 5.2 cm and 4.9 cm respectively for the natural gas and lignite. These results confirm the fact that the WMRP systems provide higher energy efficiency comparing to the conventional systems, with respect to the additional backside insulation of the WMRP.

The proposed optimization technique and the evaluated results may lead to a general result for WMRP systems and which may be used to determine the optimum insulation for many different insulation materials and climatic conditions economically and efficiently. Future works should focus on the further numerical and experimental studies to extend our correlation taking into account of the other parameters such as effect of backside insulation of WMRP.

ACKNOWLEDGMENT

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