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A numerical investigation on effects of ceiling and floor surface temperatures and room dimensions on the Nusselt number for a floor heating system ☆

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Abstract

In this study, the effect of ceiling and floor surface temperatures and room dimensions on the Nusselt number over the floor of a floor heating system has been investigated numerically. The variation of the Nusselt number with Rayleigh number has been analyzed under constant wall temperature condition for different ceiling temperatures (10–25 °C) and room dimensions. It has been seen that when the room dimensions and temperature difference between the ceiling and interior air are increased, the Nusselt number over the floor increases as well. The numerical results have been compared with the correlations given in the literature. It has been seen that the correlations available in the literature are valid only for given thermal conditions and room dimensions. The results calculated from the correlations which do not consider the effects of ceiling and floor surface temperatures deviate up to 35% than the results of this numerical study carried out for different ceiling and floor surface temperatures and room dimensions. Therefore, a new correlation for Nusselt number over the floor, which contain the influence of thermal conditions and all of room dimensions must be discovered.

Keywords: Floor heating; Natural convection; Nusselt number; Rayleigh number

1. Introduction

Floor heating systems have been used since ancient times due to their advantages compared to other heating systems. A floor heating system for large volumes and high spaces such as hangars, gymnasiums, churches, mosques, etc. seems a preferable alternative for human physiology as the temperature gradient on vertical direction in a room heating from floor is negative. The cost of this heating system is also very reasonable for large volumes and high spaces. In floor heating systems, a more comfortable environment can be obtained, because the temperature distribution in indoor air is more homogenous than that of the other heating systems. Floor heating systems are affected less on cold days when sudden temperature drops occur because heat is accumulated on the floor [1].

In the literature, the studies are mainly focused on operation, simulation and modelling of the floor heating systems. Badran and Hamdan [2] made a theoretical and experimental study for under-floor heating system using solar collectors

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Nomenclature

Gr	Grashof number ($Gr = g.\beta.\Delta T.L^3/v^2$)
H	Height (m)
h	Average convective heat transfer coefficient (W/m^2K)
L	Floor characteristic length (m)
k	Conductive heat transfer coefficient (W/mK)
Nu	Nusselt number ($Nu = hL/k$)
Ra	Rayleigh number ($Ra = Gr.Pr$)
T	Temperature (°C or K)
z	Height above sea level (m)
g	Local gravitational acceleration (m/s^2)
v	Kinematic viscosity (m^2/s)
β	Volumetric coefficient of thermal expansion (1/K)
Pr	Prandtl number

Subscripts

c	Ceiling
f	Floor
i	Indoor
w	Wall

and solar ponds. Rekstad et al. [3] presented a new approach of temperature control and energy metering in low temperature heating systems. They stated that the parameters which are necessary for temperature control could be extracted when the heat transfer coefficient and the heat capacity of the floor construction were known. Cho and Zaheer-uddin [4] explored a predictive control strategy as a means of improving the energy efficiency of intermittently heated radiant floor heating systems. They conducted both computer simulations and experiments to assess and compare the energy performance of the predictive control strategy with an existing conventional control strategy. Athienitis and Chen [5] investigated the influence of the cover layer, thermal mass thickness, and incident solar radiation on floor temperature distribution and on energy consumption in floor heating systems. Cho and Zaheer-uddin [6] tested and evaluated two flow-control schemes in an experimental facility consisting of two identical $3 \times 4.4 \times 3.8$ m rooms. Zaheer-uddin et al. [7] made a numerical study on dynamic modelling and optimal control of an embedded-piping floor heating system in order to minimize the energy input to the boiler. Khalifa [8] presented an extensive review of studies about the natural convective heat transfer coefficient on isolated vertical and horizontal surfaces with special interest in their application to building geometries. Ritter and Kilkis [9] gave an approximate panel surface heat output algorithm as a function of size and orientation of the heated space and outdoor exposure and demonstrated with an example. Khalifa and Marshall [10] carried out experimental studies to investigate the heat transfer coefficients on interior building surfaces using a real-sized indoor test cell of which dimensions are $2.95 \times 2.35 \times 2.08$ m (Length × width × height), and they developed some correlations. Li et al. [11] investigated the heat transfer coefficient experimentally in an occupied office room of which dimensions are $3.4 \times 2.6 \times 4$ m under normal working conditions. They gave a correlation related to floor convective heat transfer coefficient that is valid in low temperature difference between the floor surface and indoor air. Min et al. [12] conducted experimental studies for three different rooms heated floor and for 10^9–10^{11} of Rayleigh number ranges. Awbi [13] investigated natural convective heat transfer coefficients of a heated wall, floor and ceiling which have been calculated using "computational fluid dynamics" for $3 \times 2.3 \times 3$ dimensioned room corresponding Grashof number ranges of 9×10^8–1×10^{11}. The heat transfer over the floor in a floor heating system was investigated by Alamdari and Hammond [14] and CIBSE [15] and a correlation was given for the Nusselt number as a parameter of the Grashof number. Awbi and Hatton [16] recommended correlations related to natural convective heat transfer coefficients for heated wall, ceiling, and floor surfaces in enclosures. Table 1 presents the correlations given in the literature relevant to the Nusselt number or convective heat transfer coefficient over the floor in the floor heating system. The studies were conducted at different dimensions and thermal conditions. Obtained

Table 1
The correlations for inner surfaces of floor-heated room given in literature

Correlations	Conditions	Reference
$h=\left(1-2.22x10^{-5}z\right)^{2.627}.\left(\frac{4.96}{L}\right)^{0.08}.2.12\left(T_f-T_i\right)^{0.31}$	Analytical study	[9]
$h=3.08.\left(T_f-T_i\right)^{0.25}$	3.4×2.6×4 m office room (experimental)	[11]
$Nu=0.33Ra^{0.33}$ $h=\frac{2.16\left(T_f-T_i\right)^{0.31}}{L^{0.08}}$	3.6×2.4×7.35)m (experimental)	[8,12]
$Nu=0.269.Gr^{0,308}$	3×2.3×3 m, ΔT=5–35 °C Gr=9×10^{8}–1×10^{11} (experimental)	[13]
$Nu=\left[\left(0.52.Gr^{1/4}\right)^{6}+\left(0.126.Gr^{1/3}\right)^{6}\right]^{1/6}$	$0<Gr<\infty$ laminar/turbulent	[14]
$Nu=0.132.Gr^{1/3}$	$10^{8}<Gr<10^{10}$	[15]
$h=\frac{2.175}{L^{0.076}}.\left(T_f-T_i\right)^{0.308}$	$9\times10^{8}<Gr<7\times10^{10}$	[16]

equations are only depend on the floor thermal conditions and dimensions. The equations do not cover the effect of wall and ceiling surface temperatures and dimensions. The equations given for the floor convective heat transfer coefficient in references [9], [11] and [16] only depend on the temperature difference over the floor. The equations given for the floor Nusselt number in references [12], [13], [14] and [15] are only functions of the floor Rayleigh number or Grashof number. It is seen that these equations given in the literature do not contain any parameter related to the wall and ceiling surface temperatures and dimensions. As a result of this, when the room thermal conditions and dimensions change, the equations will not give reasonable results. Karadağ [17] found that maximum deviation between numerical results for different wall thermal conditions (insulated ceiling) and the results calculated from correlations available in literature is 35 percent. Therefore, the primary objective of this numerical study is to determine the effect of the ceiling surface temperatures and dimensions on Nusselt number over the floor in the floor heating system.

2. Computational method

A general-purpose commercial computational fluid dynamics (CFD) package FLUENT has been used for the study. It is one of the most widely used commercial codes for simulating engineering fluid flow and heat transfer problems due to its accuracy, robustness and convenience [18–20]. It uses a control-volume-based technique to convert the governing equations to algebraic equations that can be solved numerically. This control volume technique consists of integrating the governing equations about each

Table 2
The Nusselt numbers at different grid size or numbers for variable grid distribution (The room dimension is 2×1.75×2 m and the ratio of the most small grid size to the most large grid size is 0.01)

Grid number	Mean grid size (cm x cm x cm)	Nu
7200	10×10×10	427.058
13,750	8×8×8	436.598
31,581	6×6×6	442.998
56,000	5×5×5	450.360
110,000	4×4×4	454.214

Table 3
Physical properties of air [26]

T (K)	ρ (kg/m^3)	C_p (kj/kgK)	μ (Pas)	k (W/mK)
250	1.4128	1.0053	1.599×10^{-5}	0.0222
300	1.1774	1.0057	1.846×10^{-5}	0.0262
350	0.9980	1.0090	2.075×10^{-5}	0.0300
400	0.8820	1.0140	2.286×10^{-5}	0.0336

control volume, yielding discrete equations that conserve each quantity on a control-volume basis and is used in many natural and mixed convection problems by many researchers [21–23].

The details of the numerical solutions carried out in this study are given below:

1. A three dimensional figure of the floor-heated room has been plotted and meshed appropriately by GAMBIT 1.3 generator associated with the solver. In order to ensure the grid-independence solutions, a series of trial calculation was conducted for two types grid distribution; constant and variable grid distribution. It was seen that the constant grid distribution was not suitable because of that the results were different as depending on the grid size. So, the variable grid distribution of which size is very small near the surfaces but more large towards the center (the ratio of the most small grid size to the most large grid size is 0.01) was used in the present study. Table 2 gives the Nusselt numbers at different grid size or numbers for variable grid distribution (the dimension of room is $2\times1.75\times2$ m). It is seen that difference between the results of the grid numbers of 56,000 ($5\times5\times5$ cm) and 110,000 ($4\times4\times4$ cm) is 0.85%. Therefore, the grid number is chosen as 110,000 (the average grid size is $4\times4\times4$ cm). Further, the results were checked by equalizing conduction heat transfer in boundary layer region with convection heat transfer in free flow region. It was observed that the equality was verified.
2. Boundary conditions of surfaces have been defined. They can be defined at four different types in FLUENT: Constant temperature, constant heat flux, convection and convection-radiation. In this study, a constant temperature boundary condition was selected. In order to find out the variation of the floor convective heat transfer coefficient with respect to the temperatures over the floor and ceiling, a constant wall temperature was selected as 15 °C and the solutions have been conducted for different temperatures of floor and ceiling. The effect of the temperature over the wall was investigated by the same authors in the other study [24].
3. The total heat transfer from the surfaces is by radiation and convection. As the purpose of the current study is to investigate convective heat transfer coefficient over the surfaces for different thermal conditions, only the convection heat transfer was calculated. Therefore, the heat transfer by radiation over the surfaces was not taken account. The radiative heat transfer can be calculated simply by using surfaces temperatures and radiative characteristics. The heat transferred by radiation should be added to those by convective in order to obtain total heat. The relation between radiative and convective heat transfer coefficient over the floor in floor heating systems were investigated by the present authors [25].

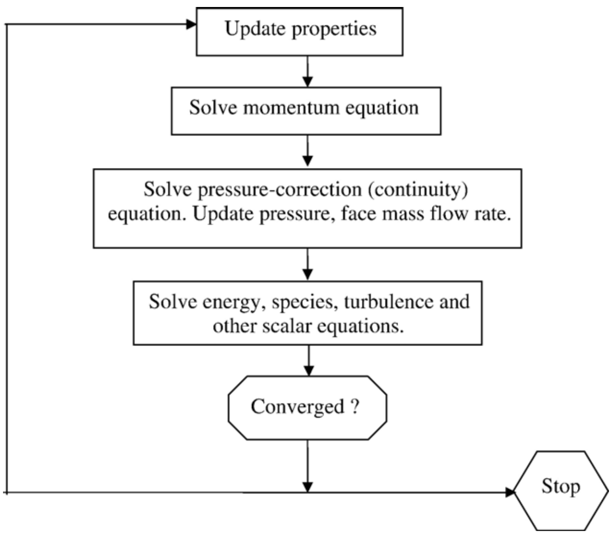

Fig. 1. Overview of the Segregated Solution Method [21].

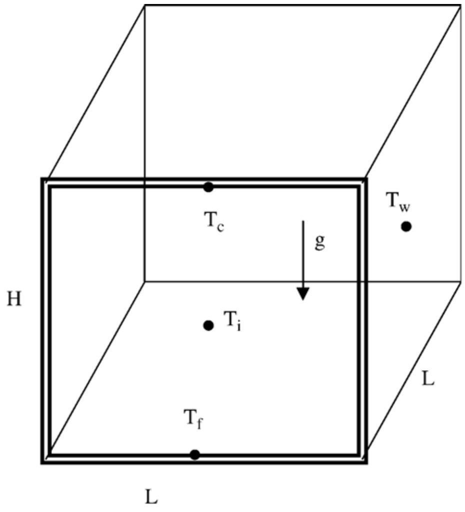

Fig. 2. Schematic drawing of the room heated from floor for the cases studied.

4. The physical properties of the air (specific heat, dynamic viscosity, thermal conductivity, and density) have been entered into the program from the tables given by Holman [26] for different temperatures. The interpolation method for interval temperature values has been used. The values of the physical properties entered in to the program are given in Table 3.
5. FLUENT uses two different solution methods named ‘‘Segregated’’ and ‘‘Coupled’’. The results of different analysis done with both methods have been compared and it has been seen that there is not a significant difference among the results [17]. However, the solution has been obtained quickly in the Segregated method. Corvaro and Paroncini [27] investigated the natural convection heat transfer in a square cavity heated from below and cooled by the sidewalls numerically and experimentally. They used the Segregated method in numerical solution and compared the results with experimental data. They found that the numerical results were suited to the experimental data. Thus, the Segregated solution method has been used in this study. In the Segregated method, the governing equations are solved sequentially (i.e., segregated from one another). Because the governing equations are non-linear (and coupled), several iterations of the solution loop must be performed before a converged solution is obtained. The flow chart of the method is illustrated in Fig. 1.
6. The standard k–ε turbulence flow model has been used in the study. The studies done in literature show that this model is appropriate for the solution of free convection. Omri and Galanis [28] calculated the turbulent flow of air in a 2D cavity with $Ra = 1.58 \times 10^9$ using the Standard and Re-Normalization Group (RNG) k–ε models and three different near wall treatment. Turbulence quantities predicted by the k–ε model was in very good agreement with large eddy simulation. They explored the numerical predictions were in good agreement with experimental values [29].
7. The residual values have been taken 10^{-3} for momentum and turbulence, 10^{-6} for energy as the result of preliminary runs made for determining appropriate residual values.

Table 4
Thermal boundary conditions and dimensions in which this numerical study was performed (The wall temperature is 15 °C)

No	Dimensions (m × m × m)	Temperatures (°C)		No	Dimensions (m × m × m)	Temperatures (°C)	
		Ceiling	Floor			Ceiling	Floor
1	1 × 1 × 1	10–25	28–57	11	4 × 2.5 × 4	10–25	28–57
2	1 × 1.75 × 1			12	4 × 3.25 × 4		
3	1 × 2.5 × 1			13	6 × 1 × 6		
4	1 × 3.25 × 1			14	6 × 1.75 × 6		
5	2 × 1 × 2			15	6 × 2.5 × 6		
6	2 × 1.75 × 2			16	6 × 3.25 × 6		
7	2 × 2.5 × 2			17	3 × 2.5 × 5		
8	2 × 3.25 × 2			18	3 × 3.25 × 5		
9	4 × 1 × 4			19	4 × 2.5 × 6		
10	4 × 1.75 × 4			20	4 × 3.25 × 6		

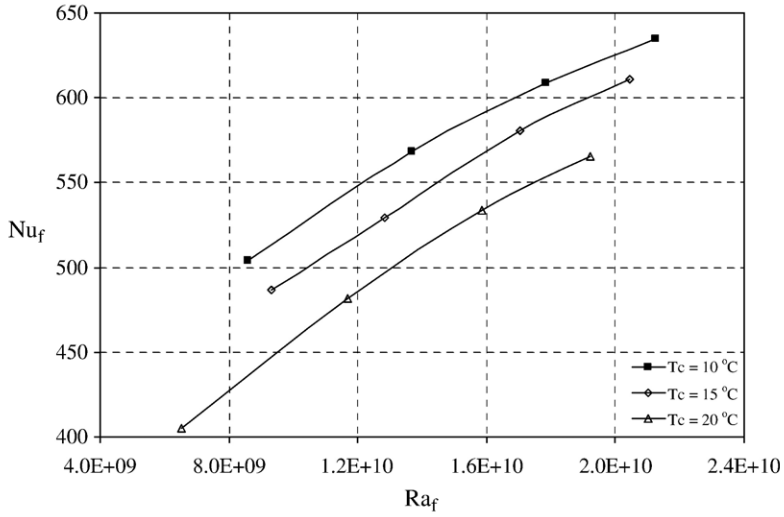

Fig. 3. Variation of Nusselt number with Rayleigh number over the floor for different ceiling temperatures ($L \times H \times L = 2 \times 1.75 \times 2$) ($Nu_f = h_f.L/k$, $Ra_f = g.\beta.(T_f - T_i).Pr.L^3/\nu^2$).

A schematic presentation of the heated floor room is given in Fig. 2. The boundary conditions and dimensions in which this numerical study has been performed are given in Table 4. The wall temperature are assumed to be constant ($T_W = 15$ °C) in the present study. The floor temperatures are selected within extensive ranges so that the results can be used at different applications of the floor heating system. It should be noted that the maximum floor temperature to be acceptable is 29 °C for houses [10,30] and 38.4 °C for coops [17].

3. Results and discussion

Numerical solutions were carried out for different ceiling and floor surface temperatures and room dimensions, which are specified in Table 4, in order to determine the effects of the thermal conditions on the Nusselt number over the floor. Figs. 3–5 show the effect of ceiling temperatures on the floor Nusselt number for three different room dimensions. It can be seen from the figures that as the ceiling temperature increases, the temperature difference between the indoor air and ceiling surface ($T_i - T_c$) decreases, and for this reason the Nusselt number over the floor decreases as well. This shows that the air flow around the ceiling affects the Nusselt

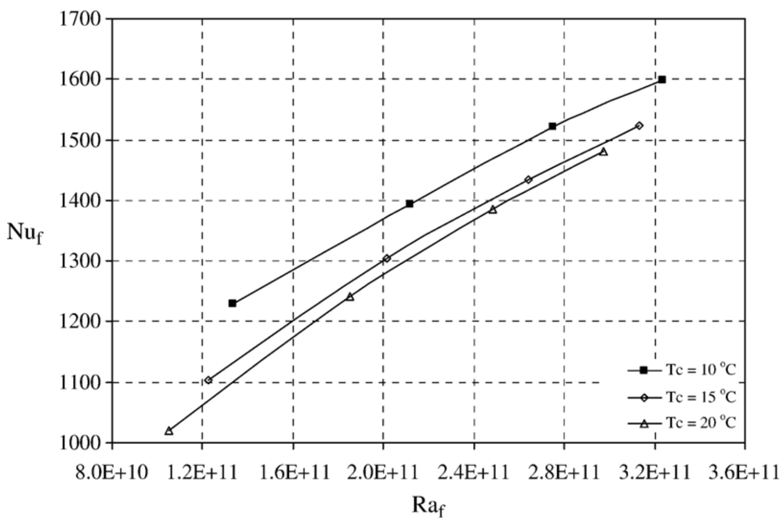

Fig. 4. Variation of Nusselt number over the floor with Rayleigh number for different ceiling temperatures ($L \times H \times L = 4 \times 2.5 \times 6$) ($Nu_f = h_f.L/k$, $Ra_f = g.\beta.(T_f - T_i).Pr.L^3/\nu^2$).

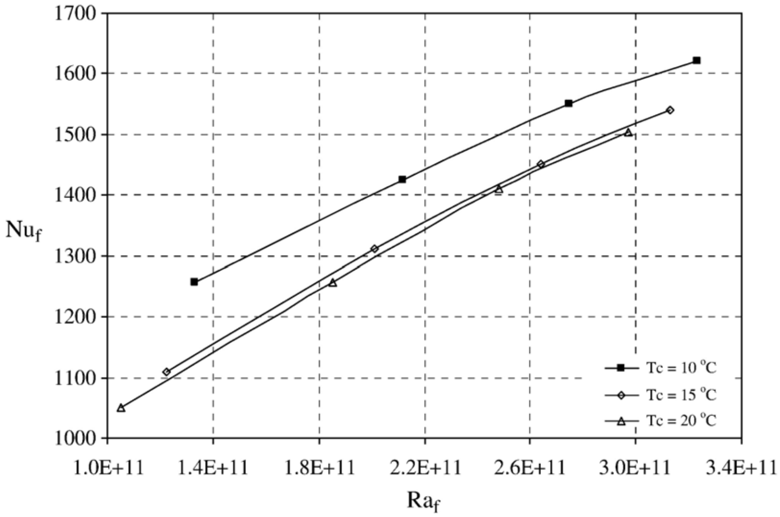

Fig. 5. Variation of Nusselt number over the floor with Rayleigh number for different ceiling temperatures ($L \times H \times L = 4 \times 3.25 \times 6$) ($Nu_f = h_f.L/k$, $Ra_f = g.\beta.(T_f - T_i).Pr.L^3/\nu^2$).

number over the floor. Figs. 6 and 7 show the variation of the Nusselt number with Rayleigh number over the floor at constant ceiling temperature of 10 °C for different room dimensions. From the Fig. 6, it is deduced that the correlation of the Nusselt number related to Rayleigh number over the floor is different depending on the wall height. Similarly, Fig. 7 shows that the correlation varies with respect to the floor characteristic length as well. As shown in the figures when the wall height and the floor characteristic length increase, the Nusselt number over the floor also increases. This is due to the effect of the air flow over the wall and ceiling on the floor Nusselt number. By considering Figs. 3–5 and Figs. 6–7 together, it can be seen that the floor Nusselt number does not depend on the Rayleigh number over the floor alone, but also the other surfaces. However, with respect to the equations given in the literature, the floor Nusselt numbers are only dependent on the floor Rayleigh numbers. In other words, there is no other parameter concerned with wall and ceiling surface temperatures and dimensions in the equations. Therefore, a new equation related to floor Nusselt number which contains new parameters such as ceiling and wall Rayleigh numbers besides the floor Rayleigh number should be discovered in order to decrease the maximum deviation.

In Figs. 8 and 9, the numerical results obtained from the present study for different ceiling and floor surface temperatures and room dimensions are compared with the results of the correlations given in literature [9,11,12,15,16]. As can be seen in Fig. 8, the

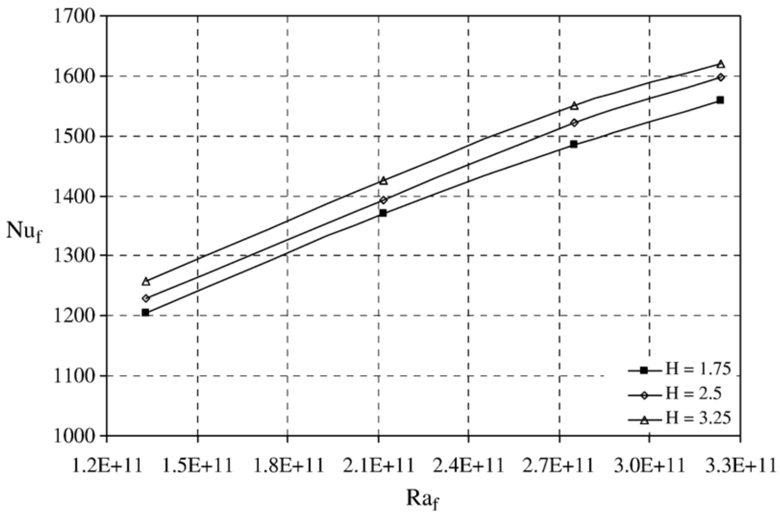

Fig. 6. Variation of Nusselt number over the floor with Rayleigh number for different wall heights ($L \times L = 4 \times 6$ and $T_c = 10$ °C) ($Nu_f = h_f.L/k$, $Ra_f = g.\beta.(T_f - T_i).Pr.L^3/\nu^2$).

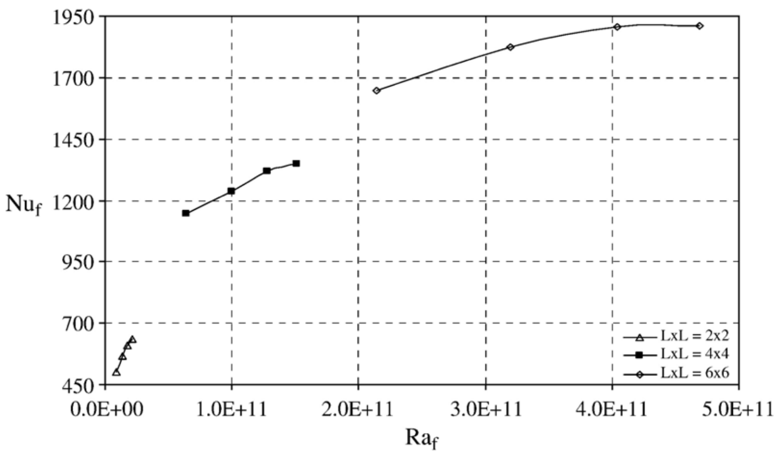

Fig. 7. Variation of Nusselt number over the floor with Rayleigh number for different floor dimensions (H=1.75 and T_c=10 °C) ($Nu_f=h_f$.L/k, $Ra_f=g.\beta.(T_f-T_i).Pr.L^3/\nu^2$).

relation between the floor Nusselt number and floor Rayleigh number in the current study which is obtained for 25 °C ceiling temperature and $L \times L = 2 \times 2$ is linear resembling the correlations given in the literature. As the temperature difference over the ceiling is chosen very small, the effect of ceiling conditions on the floor Nusselt number has been decreased in this situation. However, in Fig. 9, it is not linear as a different from those of the literature. The values obtained for different ceiling and floor surface temperatures and room dimensions have been used in Fig. 9. It is seen that the floor Nusselt number is not only dependent on the floor Rayleigh number but also on the ceiling temperatures and room dimensions. The results of correlations in literature given in Table 1 and those obtained in the current study are different from each other due to the consideration of different thermal conditions and dimensions.

4. Conclusions

In this study, the effect of ceiling and floor surface temperatures and room dimensions on the Nusselt number over the floor in a floor heating system was investigated numerically. Comparison of the results obtained in this study with the results given in literature revealed that the temperature differences over the ceiling, floor and wall surfaces, and room dimensions should be taken into account in the calculation of the Nusselt number on the floor. It was also seen that the correlations given in the literature are valid only for given specific thermal conditions and room dimensions.

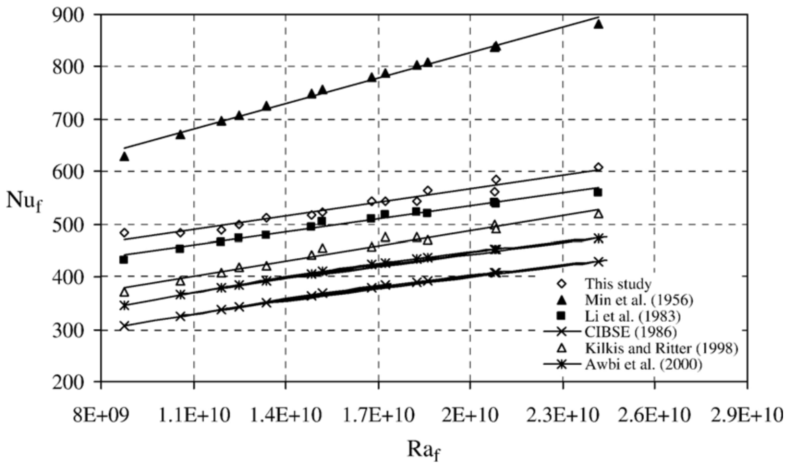

Fig. 8. Comparison of numerical results obtained from present study for different wall heights and small ceiling dimensions with the results of correlations available in literature (T_c=25 °C, $L \times L = 2 \times 2$ and H=1–3.25) ($Nu_f=h_f$.L/k, $Ra_f=g.\beta.(T_f-T_i).Pr.L^3/\nu^2$).

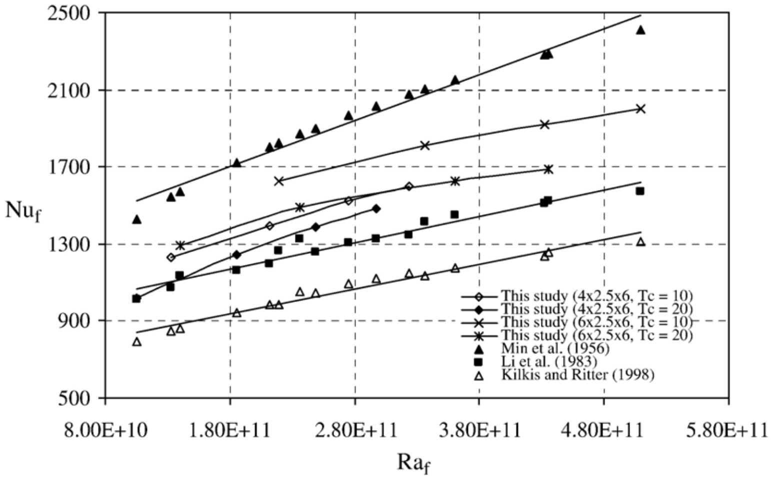

Fig. 9. Comparison of numerical results obtained from present study for two different room dimensions and ceiling temperatures with the results of correlations available in literature (T_c=10 and 20 °C, LxHxL=4×2.5×6 and 6×2.5×6) ($Nu_f=h_f \cdot L/k$, $Ra_f=g.\beta.(T_f-T_i).Pr.L^3/\nu^2$).

A maximum 35% deviation between the numerical results obtained in this work and the results of correlations given in literature was observed. This is due to the effects of temperature difference over the ceiling and wall surfaces and room dimensions.

The correlations given for the Nusselt number over the floor should contain the effects of temperature differences over the floor, wall and ceiling surfaces, and room dimensions. When the Rayleigh number over the floor, wall, and ceiling increases, the Nusselt number over the floor increases too. Therefore, a new correlation for Nusselt number over the floor should be determined as a function of Rayleigh number over the floor, wall and ceiling surfaces ($Nu_f=f(Ra_f, Ra_w, Ra_c)$).

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