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Thermal radiation in a room: numerical evaluation

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Summary This paper describes the use of the discrete transfer method for the calculation of radiative heat transfer in heated situations within buildings. The principal features and the flexibility of the method are illustrated by considering a number of representative thermal conditions found in room configurations. Results obtained by the discrete transfer method are compared with the results from the standard matrix method. The study also demonstrates how the discrete transfer method is capable of evaluating the distribution of heat fluxes on wall surfaces in complex geometrical situations. Its use has a clear advantage in situations where the standard methods require considerable user input.

Thermal radiation in a room: Numerical evaluation

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List of symbols

G	Total incident heat on the surface element (W m^{-2})
I	Intensity of radiation along a ray ($\text{W m}^{-2} \text{steradian}^{-1}$)
$\dot{q}$	Total heat flux leaving the surface element (W m^{-2})
K	Absorption coefficient
$n, n + 1$	Successive locations along a ray
q_{net}	Net heat flux on the surface (W m^{-2})
s	Distance along a ray (m)
T	Temperature (K)
ϵ	Emissivity
σ	Stefan-Boltzmann constant ($\text{W m}^{-2} \text{K}^{-4}$)

Subscripts

g	Gas
w	Wall
j	Ray number

1 Introduction

Convective and radiative heating systems are used to provide heating and related comfort conditions in domestic and industrial buildings. The two systems provide different types of comfort conditions, however, due to the nature of the mechanism by which heat is transferred to the occupants. Of the two methods, heating by radiation is the more efficient as radiative heat from the relevant sources falls directly on the occupants, so producing the required level of comfort. Thus, the room may be operated at a lower overall air temperature but the comfort conditions (as defined by the expressions of Fanger⁽¹⁾) are determined by the mean radiation temperature (MRT).

Radiation heat transfer takes place between surfaces in a room and also between any people present and their surroundings. Therefore, in radiative systems, the walls, floor, ceiling and windows receive considerable amounts of heat from the hot radiating surfaces and this engenders a greater heat loss compared with convective systems. The exact amount depends on surface temperatures and the magnitude of heat conduction through walls. In designing heating systems, it is therefore

important to determine the amount of heat received by wall surfaces.

For simple configurations, average heat losses can be calculated using standard radiative heat transfer techniques. A family of different methods exists for the calculation of radiative heat transfer in rooms. Methods based on an electrical circuit analogy and the radiosity matrix are in common usage today. Based on these techniques, a number of researchers have developed models for the calculation of radiative heat transfer in buildings. In two papers, Stefanizzi *et al.*^(2, 3) have reviewed some of these methods and have compared the relative predictive capabilities. Their studies have shown that simple methods are adequate for standard geometries but more accurate methods are required in the cases of low surface emissivity or where there are large temperature differences between surfaces.

The value of the MRT which is widely used in determining comfort conditions (Fanger^(1, 4)) is evaluated by using configuration factors and surface temperatures and there are a number of different methods that can be used in this context. Olesen *et al.*⁽⁵⁾ have reviewed and compared such methods but it is clear that all such calculation methods depend on the availability of configuration factors. Techniques used for the sizing of radiator systems (Harwell⁽⁶⁾), the evaluation of heat transfer in radiation systems and the design procedures for the associated comfort conditions (see, for example, Ling⁽⁷⁾) are also based on the value of the MRT. For complex geometrical configurations, the methods based on mean radiation temperature are inapplicable however, as view factors are unavailable. Therefore, a need exists for the incorporation of general radiation heat transfer methods into building heat transfer models.

The absorption of radiation by enclosed air is often ignored due to the fact that the absorption coefficients of participating components in air (namely CO_2 and water vapour) are very small. However, in high humidity environments, the absorption of radiation by air can be significant and should be accounted for in calculations. The calculation of radiation in emitting/absorbing media is a difficult problem. Again, for simple configurations, network methods and zone methods can be used. However, for situations where there are a number of heaters and windows at different surface temperatures, network or zone methods can be very tedious. For such

Figure 1 Ray tracing in the discrete transfer method

situations, numerical radiative transfer methods based on the Monte Carlo⁽⁸⁾, flux⁽⁹⁾ and discrete transfer methods⁽¹⁰⁾ can be employed. Radiative transfer can be calculated more accurately with such techniques and involves relatively little effort from the user. Considerable user input is required in calculating configuration factors for network and zone methods. This paper demonstrates the application of the discrete transfer method in the evaluation of wall heat fluxes in radiant heating systems. It has the advantage of enabling calculations to be made of the distribution of wall heat fluxes and the intensity of radiation at any location and is applicable in emitting/absorbing media so that the effect of CO₂ and humidity can be taken into account. Therefore, it provides more accurate heat fluxes relative to those calculated from network or zone methods. It is also applicable to any arbitrarily shaped geometry and can be used to evaluate configuration factors as a pre-processor for existing radiative calculation procedures.

2 Discrete transfer method

The discrete transfer method is based on the direct solution of the radiation transport equation

$$\frac{dI}{ds} = K_g \frac{\sigma T^4}{\pi} - K_g I \quad (1)$$

where I is the radiation intensity in the direction of Ω , s is the distance in that direction, K_g is the gas absorption coefficient, σT^4 is the black body emissive power of the gas (E_g) and σ is the Stefan-Boltzmann constant. The calculation domain is sub-divided into a three-dimensional numerical grid as shown in Figure 1. The spatial distribution of the numerical grid can be decided by the user to include details of radiators, windows, doors and surfaces with different temperatures. The solution procedure is based on a ray tracing technique. Rays emanating from the centre of each cell wall are considered. For example, a ray starting from point P, which lies on the floor of the configuration, is shown in Figure 1. The ray passes through several grid cells to reach another wall. Equation 1 may be integrated along the ray to yield the recurrence relation:

$$I_{n+1} = E_g^*(1 - e^{-K_g \delta s}) + I_n e^{-K_g \delta s} \quad (2)$$

where n and $n + 1$ designate successive locations through a cell, δs is the distance between n and $n + 1$ and $E_g^* = K_g E_g / \pi$. The rays are chosen to represent equal angular spacing emanating from the centre points on the wall cells. The recurrence relation (equation 2) is applied along the chosen representative direction Ω . E_g^* and K_g are presumed to be uniform over each cell and the intensity is calculated along the ray starting from a wall location. By calculating radiation from all the locations and by considering a sufficient number of rays, the intensity of radiation arriving at each wall cell can be calculated. The procedure is iterative. For a temperature-prescribed wall, the initial intensity of radiation leaving a wall is taken as

$$I_0 = \frac{\varepsilon \sigma T^4}{\pi} \quad (3)$$

By considering all rays arriving at a wall cell face (for example on the face where point Q is situated) and applying the relation in equation 2 along all the rays, the incident heat flux on that face can be obtained from

$$G = \sum_{all j} I_j(\Omega_j, \mathbf{n}) \delta \Omega \quad (4)$$

where I_j is the calculated intensity in a discretised angle $\delta \Omega_j$ and $\mathbf{n}$ is a unit vector normal to the wall. The I_j values are the intensities from all the rays converging to a point (number of rays selected by the user). Except for black walls, the local heat flux, $\mathcal{J}$, leaving the cell face is given by

$$\mathcal{J} = (1 - \varepsilon_w)G + \varepsilon_w E_w \quad (5)$$

where ε_w is the emissivity of the wall material. The new radiation intensity leaving the wall cell face to be used as the initial value when applying the recurrence relation in the next iteration is

$$I_0 = \frac{\mathcal{J}}{\pi} \quad (6)$$

The procedure is repeated until all the G values calculated in two successive iterations are within a prescribed tolerance (10^{-2}). The accuracy of the solution is further assessed by performing a heat balance. If the solution does not result in an acceptable heat balance, it can be improved by increasing the number of rays used per point or by refining the grid. The grid resolution and the number of rays per point are parameters that are chosen and used to achieve the heat balance. At the end of the calculation, the net radiative heat flux on a wall cell surface can be obtained from

$$q_{net} = G - \mathcal{J} \quad (7)$$

3 Test results and discussion

The principal features and the flexibility of the method are illustrated by considering a number of representative thermal conditions found in room configurations. Case 1 is a cubic geometry used by Stefanizzi *et al.*⁽³⁾. The geometry is shown in Figure 2. It is a $3 \times 3 \times 3$ m cube with all surfaces, except surface 1, at 20°C. Surface 1 is maintained at 60°C. Emissivities of all surfaces are considered to be 0.9. The present calculation procedure is applied to calculate the flux distribution of the surfaces and the results are compared with the results obtained by using the radiosity matrix method⁽¹¹⁾.

Figure 2 Configuration for Case 1

Figures 3, 4 and 5 show the wall flux distributions calculated for the configuration shown in Figure 2. A grid of 10×10 was used to represent each surface. The present method enables the user to calculate the actual flux distribution at any location on a particular wall. Most other methods assume a constant flux distribution over the entire surface. When the configuration is large and the geometry is complex, this assumption may not be valid since there can be significant differences in fluxes over a large surface. These differences can be important in determining comfort conditions. Table 1 compares the total net flux on the surfaces calculated using the present method and the radiosity matrix method⁽¹¹⁾. The values calculated by the present method are slightly different because, in the radiosity matrix method, the surface flux has been assumed to be constant over a surface.

Although the radiosity matrix method is analytically exact, its results will depend on how the surfaces are physically represented in the calculation. Therefore, the

Figure 4 Heat flux (W) on surface 4

results from the matrix method can only be used as a guide for comparisons. The present results are more accurate since the surfaces have been divided into a large number of elemental surfaces and this ensures a more accurate and detailed representation of the actual flux distribution.

Figure 3 shows the heat flux on the 60°C surface 1. The negative sign indicates that the surface is emitting radiation. Figure 4 shows the heat flux on surface 4 which is directly opposite to surface 1, while Figure 5 illustrates the heat flux distribution on surfaces 2, 3, 5 and 6. The heat fluxes on these surfaces are identical due to symmetry. Figure 5 clearly indicates a variation of heat flux over the surface with the greatest amount being received at the sides closest to surface 1. The assumption of a constant heat flux over all such surfaces and the use of the matrix method yields a flux value of 48.8 W m^{-2} over the entire surface (see Table 1) whereas the present results show a distribution ranging between $20\text{--}100 \text{ W m}^{-2}$.

Figure 3 Heat flux (W) on surface 1 (temperature 60°C)

Figure 5 Heat flux (W) on surfaces 2, 3, 5 and 6

Table 1 Comparison of the discrete transfer method and the matrix method (Case 1)

Surface	Temperature (°C)	Discrete transfer method		Matrix method	
		Total flux (W)	Flux per unit area (averaged values) (W m^{-2})	Total flux (W)	Flux per unit area (W m^{-2})
1	60	2206.2	-245.1	2221.5	-246.8
2	20	445.6	49.5	440.0	48.8
3	20	445.6	49.5	440.0	48.8
4	20	423.7	47.0	439.5	48.7
5	20	445.6	49.5	440.0	48.8
6	20	445.6	49.5	440.0	48.8

(Figure 5). The 10 W contour corresponds to 100 W m^{-2} and the 2 W contour corresponds to 20 W m^{-2} . Undoubtedly, such variations can have a noticeable effect in determining comfort conditions in practical situations.

The use of the discrete transfer method to calculate wall heat fluxes in radiating heating systems is further demonstrated in this simulation by considering two typical building room configurations. These configurations were considered by Olesen *et al.*⁽⁵⁾ and could be considered as representing residential buildings, office rooms or industrial workplaces. In Case 2 and Case 3, the surface temperatures have been assigned the values used by Olesen *et al.*⁽⁵⁾ and are specified in the subsequent discussion. Although the present method is capable of including absorption effects due to atmospheric air and water vapour, it has been neglected in the present calculations in the interest of comparing the results with the matrix method.

Case 2 (represented by the configuration shown in Figure 6) relates to a heated floor situation whilst Case 3 (depicted in Figure 12) is a heated ceiling configuration. In both cases, the large window on surface 1 is at 8°C and the other parts of this wall surface are at 18°C . All other surfaces (except the heated surface) are at 20°C . In Case 2, the floor temperature is 27°C and, in Case 3, the ceiling temperature is at 32°C .

Figure 7 Heat flux (W) on walls 2 and 5

Figures 7–11 and Figures 13–17 show the predicted heat fluxes on certain of the surfaces of the room configurations in Figures 6 and 12 respectively. A comparison of Figures 9 and 15 reveals that the heat losses through the window are higher in the case of the heated ceiling situation (Case 3). This is to be expected since the temperature of the ceiling is higher than in the heated floor situation. A further comparison of Figures 7–10 with Figures 13–16 shows that the flux distributions on walls are opposite in pattern and higher in value for the heated ceiling compared with the heated floor situation. It will be recalled that the negative sign indicates a net emission. Any other technique used for such calculations (other than the discrete transfer method) would not provide such detailed distributions resulting from different thermal conditions. Comparing the results of Case 2 and Case 3, it is observed that a 5 K increase in the temperature of the heated surface can result in a significantly

Figure 6 Configuration for Case 2

Figure 8 Heat flux (W) on wall 4

different heat flux distribution on the walls in this configuration. Tables 2 and 3 show the comparison of total net heat fluxes calculated by the present method and the matrix method. Although the overall agreement is close, it should be noted that, as mentioned earlier, the values calculated by the matrix method can only be used as a guide since greater accuracy is achieved by the discrete transfer method. In the matrix method, the errors involved in the calculation of configuration factors are

Figure 9 Heat flux (W) on wall 1 (wall with window)

Figure 10 Heat flux (W) on floor (surface 3 at 27°C)

Figure 11 Heat flux (W) on ceiling (surface 6)

compounded when the surfaces are subdivided into further elements (in this case, surface elements 1a, 1b and 1c shown on Figure 6).

The advantages of the present method in dealing with complex configurations are illustrated by analysing two further situations (Case 4 and Case 5). Case 4 is an L-shaped room of the form and dimensions considered by

Figure 12 Configuration for Case 3

Figure 13 Heat flux (W) on walls 2 and 5

Figure 14 Heat flux (W) on wall 4

Stefanizzi *et al.*⁽³⁾. The geometry is shown in Figure 18. All the surfaces are considered to be at 20°C except surface 3 which is at 10°C. The emissivities of all surfaces are taken to be 0.9. An $11 \times 5 \times 11$ computational grid was used to model this geometry and Figure 19 shows the computational grid used in the x - y plane. In the x - y plane, a non-orthogonal grid as opposed to a Cartesian grid is more suitable for this geometry as it enables the geometry to be modelled with a minimum

Figure 15 Heat flux (W) on wall 1 (wall with window)

Figure 16 Heat flux (W) on floor (surface 3)

Figure 17 Heat flux (W) on ceiling (surface 6 at 32°C)

Table 2 Comparison of the discrete transfer method and the matrix method (Case 2)

Surface (see Figure 2)	Total heat flux (W) on each surface	
	Matrix method	Discrete transfer method
1a	47.3	62.7
1b	266.9	170.5
1c	47.3	62.7
2	34.3	42.0
3	-515.7	-500.7
4	41.5	38.4
5	34.3	42.0
6	68.9	75.5

Table 3 Comparison of the discrete transfer method and the matrix method (Case 3)

Surface (see Figure 6)	Total heat flux (W) on each surface	
	Matrix method	Discrete transfer method
1a	67.1	80.5
1b	292.3	200.3
1c	67.1	75.6
2	104.7	112.1
3	168.5	169.9
4	93.6	94.9
5	104.7	112.1
6	-859.6	-853.2

number of cells. A uniform grid was used in the z direction and the calculated wall heat fluxes are shown in Figures 20–26. It can be seen that the colder surface 3 receives radiation whilst all other surfaces act as emitting surfaces. Table 4 compares the overall wall heat fluxes calculated by the discrete transfer method with the fluxes calculated by Stefanizzi *et al.*⁽³⁾ using the matrix method. The overall fluxes evaluated by the present method agree closely with those from the matrix method. This case illustrates the capability of the discrete transfer method in handling complex geometrical shapes. Although the matrix method is applicable to this particular L-shaped geometry, the setting up of the matrix requires consider-

Figure 18 Configuration for Case 4

Figure 24 Heat flux (W) on wall 2

Figure 25 Heat flux (W) on wall 5

Figure 26 Heat flux (W) on ceiling and the floor

Figure 27 Configuration for Case 5

able effort in terms of the evaluation of configuration factors. The present method does not require any additional effort other than the setting up of a suitable computational grid and this is a fairly straightforward task.

The applicability of the discrete transfer method to complex geometries which cannot be handled by other methods is further illustrated by considering Case 5. The geometry analysed here is shown in Figure 27 and includes a semi-cylindrical surface (i.e. surface 4). The lower part of surface 1 is assigned a temperature of 60°C which could be considered to be a radiator. The remainder of surface 1 and all other surfaces are assumed to be at 20°C. The emissivities of all surfaces are given the value 0.9. Figure 28 shows the computational grid used in the x - y plane. A body-fitted non-orthogonal grid is used in the x - y plane whilst a uniform grid is used in the

Figure 28 x - y grid used

Figure 29 Heat flux (W) on floor (surface 3)

z direction. The overall grid used in the simulation was $10 \times 11 \times 11$. The predicted heat fluxes on the various walls are shown in Figures 29–33. Figure 29 shows the flux on the floor (surface 3) whilst Figure 30 shows the flux on the ceiling (surface 6). A comparison of Figures 29 and 30 indicates that the floor receives more radiation as expected since the radiator is closer to the floor. Figure 31 shows the heat flux distribution on the cylindrical surface (wall 4). As it is the surface which is directly exposed to the radiator, a clear distribution is seen from top to bottom. Figure 32 shows the heat flux on walls 2 and 5 and these are identical due to symmetry. The distribution is again as expected with more flux being received towards the radiator side. Finally, Figure 33 portrays the flux on the wall with the radiator which

Figure 30 Heat flux (W) on ceiling (surface 6)

Table 5 Total heat fluxes calculated by the discrete transfer method (Case 5)

Surface	Total heat flux (W)
1	-495.4
2	90.5
3	140.5
4	124.4
5	90.5
6	54.0

acts, of course, as the emitting surface. Depicted in Table 5 are the net total heat fluxes calculated by the present method. These cannot be compared with the results of the matrix method as the matrix method is not capable of handling this geometry.

Figure 31 Heat flux (W) on wall 4

Figure 32 Heat flux (W) on walls 2 and 5

Figure 33 Heat flux (W) on wall 1

4 Conclusions

This study demonstrates the use of the discrete transfer method for the calculation of radiative heat transfer and fluxes in boxed-shaped rooms within buildings. As demonstrated however, this method is equally applicable to more arbitrarily and complex-shaped geometrical configurations so that, by its very nature, it has a clear advantage in being completely general. Standard methods of evaluating radiative heat exchanges are not readily applicable to situations such as large buildings with glass roofs, arbitrarily shaped buildings, shopping malls, airport terminal configurations etc. Also, when there are large glass areas present in the geometry, solar radiation entering the room can significantly affect the heating characteristics of the building. The discrete transfer method can be easily used to calculate heat transfer in such situations with solar gains accounted for by specification of the flux on the relevant surfaces. If required, emitting/absorbing characteristics can also be considered. Further, since the method is based on a ray tracing technique, it can be used to analyse the directional effects of solar radiation on heating characteristics. It is therefore extremely powerful and flexible and can be incorporated into general thermal analysis procedures.

As the method does not require any user inputs, such as configuration factors, it is especially appropriate to comparative studies concerning such features as alternative radiator arrangements, location of windows etc. The data generated can then be used to assess the quality of the thermal environment in terms of effectiveness and comfort. A disadvantage of the discrete transfer approach is that it requires more computer time and memory compared to other methods currently in use but it nevertheless is particularly useful in dealing with unconventional geometrical configurations where standard radiative calculation techniques are not applicable.

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