Available online at www.sciencedirect.com
SCIENCE DIRECT®
ELSEVIER
International Journal of Thermal Sciences 42 (2003) 311–316
International Journal of Thermal Sciences
www.elsevier.com/locate/ijts

Distribution ratio of radiant heat and its effect on cooling load

Zhiwei Lian [a,*,1], Yan Zhang [b]

[a] *Institute of Refrigeration and Cryogenics, Shanghai Jiao Tong University, Shanghai 200030, People's Republic of China*
[b] *Engineering Training Center, Shanghai Jiao Tong University, Shanghai 200030, People's Republic of China*

Received 7 July 2001; accepted 3 May 2002

Abstract

Distribution ratio of radiation heat on each inside surface of building enclosure of a room has an important effect on the accurate calculation of air conditioning loads. Cooling loads are calculated and analyzed under different distribution ratios on the building enclosure of a room with underfloor air conditioning system and with different constructions. It is found that the previous approximate methods of distribution ratio will result in serious errors when calculating the cooling loads accurately formed by radiant heat gains from indoor heat sources. The accurate method should be calculating the loads based on the actual distribution ratio to each of the inside surfaces of the room. Factors affecting the ratio of radiant heat sent out by indoor heat sources are analyzed. And some regular results are concluded with the ratio calculated under different conditions. The results can supply reference to the people concerned.

Keywords: Air conditioning; Cooling load; Radiation heat transfer; Indoor heat source; Distribution ratio

1. Introduction

It is inevitable that the problem of distribution ratio of radiant heat, Pr_i, would be encountered when air conditioning cooling loads are accurately calculated. Different building enclosures cause different cooling loads when the radiant heat goes through them at any time. Therefore, distribution ratio, Pr_i, on the building enclosures directly influences the cooling loads of the room at each hour. A systematic study on this problem has not been made till now, however. Different solutions to this problem are used domestically and abroad in calculating cooling loads of rooms. For example, as regard to indoor heat sources (including heat equipment, lighting fixtures and occupants), radiant heat is generally evenly distributed, by area, on the inside surfaces of the building enclosure [1], and as to solar radiant heat, the same method as above is used [2], and another method is also applied in the computation, such as 50% of the radiant heat is distributed on the floor and the other 50% is evenly distributed by area on the other inside surfaces [3]. What about the result differences when these methods are used to deal with the cooling loads of the same room? It is discussed as regard to indoor heat sources inside a room with underfloor air conditioning system.

2. Effect of different methods on cooling load

Different constructions of a room cause different cooling loads at each hour as regard to the same affecting factor. A computer program [4] is made to calculate cooling loads of a room through setting a set of partial differential equations and solving them to obtain response factors of the building enclosure and, through considering view factors of radiation heat transfer and considering the distribution ratio Pr_i to each inside surfaces of the building. A subprogram is made to calculate the view factors based on their basic definition and by way of principle of superposition. The program calculating the cooling loads in a room has been verified using climate chamber tests. Results show the average error between the simulation and experiment is 4.1% and the maximum one is 15.9% [4]. Table 1 shows the results of cooling loads in the occupied zone calculated by the program, when a unit step factor affecting the building enclosure of an air conditioned room with medium-weight type construction according to following radiant heat distribution method:

* Corresponding author.
E-mail addresses: zwlian@sjtu.edu.cn (Z. Lian), zhangy526@263.net (Y. Zhang).
[1] Member of E_1 Commission of IIR.

1290-0729/02/$ – see front matter
doi:10.1016/S1290-0729(02)00033-9

Nomenclature

A	surface area of heat source	m^2
E	emission power	$W\cdot m^{-2}$
F	view factor	
J	effective radiation	$W\cdot m^{-2}$
Pr	distribution ratio of radiant heat	
Q	quantity of heat	W
T	surface temperature	K

Greek letters

ε	emissivity	
σ	radiation constant of black body	$W\cdot m^{-2}\cdot K^{-4}$

Subscripts

b	black body
e	equivalence
i	inner surface of room
j	inner surface of room
r	radiation
t	total
v	vacuum tank
w	wall
0	heat source

(1) Radiant heat is evenly-distributed by area to each inside surfaces and the corresponding cooling load is expressed by CL_A;
(2) 50% of the radiant heat is distributed on the floor and the other 50% is evenly distributed by area on the other inside surfaces. The corresponding cooling load is expressed by CL_D;
(3) All of the radiant heat is distributed to an inner wall, outer wall, floor and ceiling, respectively. And the corresponding cooling load is expressed by CL_I, CL_O, CL_F, CL_C, respectively.

It can be seen from Table 1 that there are big differences among the methods. The maximum cooling load occurs when the affecting factor is only on the inner wall and the minimum on the ceiling. It is not difficult to explain through the two main factors causing the cooling load in a room with underfloor air conditioning system. One is the distribution ratio of radiant heat in occupied and non-occupied zone; The other is the coefficient of the convective heat transfer, α_C, of each inside surface. According to reference [5], as regard to the room with underfloor air conditioning system, it can be got

$$\alpha_C(\text{sidewall}) > \alpha_C(\text{ceiling}) > \alpha_C(\text{floor})$$

Of course, the latter four methods are extreme situations which are supposed for analysis. It is very commonly seen, however, for the heat equipment to stand near the inner wall in practical engineering (e.g., big computer room). Distribution ratio of the radiant heat of the equipment to the inner wall is much bigger at this time.

Now a calculation example is analyzed as follow.

According to the characteristics of a room with underfloor air conditioning system, the room is divided into two sections: occupied zone and non-occupied zone. In this way, the building enclosure of the room consists of thirteen faces (as shown in Fig. 1). The room is a kind of medium-weight type construction. Its outer wall is of 0.37 m brick and inner walls are of 0.24 m brick and floor is of 0.1 m reinforced concrete, with 0.02 m cement mortar on each inner and outer surfaces.

Faces 1, 2, 5 and 6 refer to the inside surfaces of the side walls and 7, 8 to the window and door in the occupied zone, respectively, and faces 9–13 stand for the inside surfaces of the building in non-occupied zone. The room is of medium construction with cement mortar as its inner surfaces. Its size

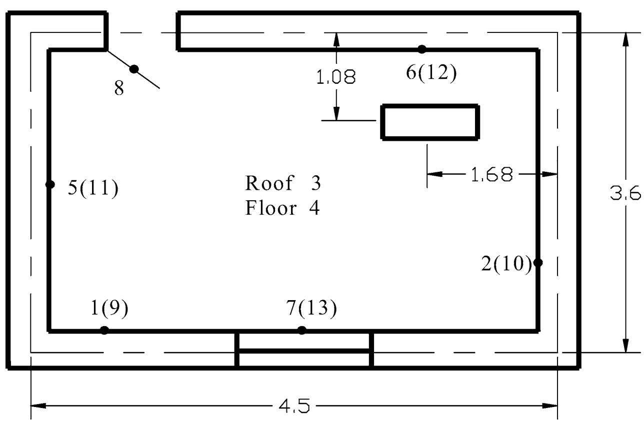

Fig. 1. Position of the heat source in the room in the calculation example.

Table 1
Effect of Pr_i on the cooling load in a room with underfloor air conditioning system

Time (h)	CL_A (W)	CL_D (W)	CL_I (W)	CL_O (W)	CL_F (W)	CL_C (W)
1	0.1418	0.1312	0.1801	0.1727	0.1164	0.0190
3	0.1842	0.1764	0.2370	0.2152	0.1639	0.0323
5	0.2108	0.2059	0.2701	0.2400	0.1981	0.0433
10	0.2512	0.2514	0.3182	0.2761	0.2519	0.0639
20	0.2887	0.2912	0.3642	0.3112	0.2915	0.0856
40	0.3120	0.3129	0.3939	0.3382	0.3144	0.0987

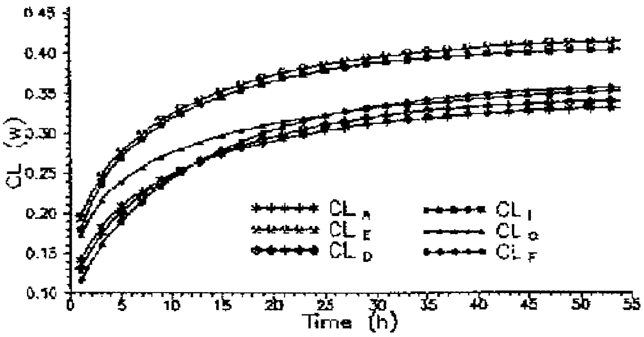

Fig. 2. Effect of different Pr_i on loads of a heavy construction room when heat source near inner wall.

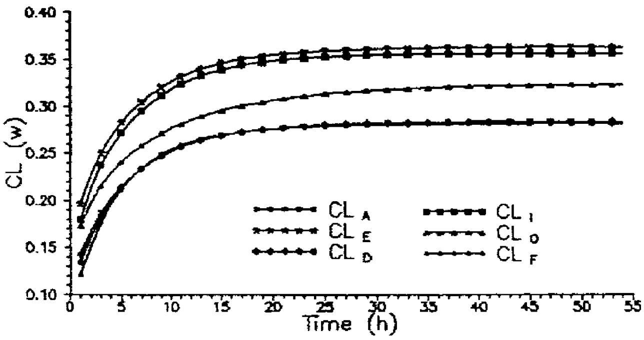

Fig. 3. Effect of different Pr_i on loads of a light construction room when heat source near the inner wall.

is 4.5 m × 3.6 m × 3.3 m (H). Dimensions of an indoor heat source are 0.5 m × 2.0 m × 2.0 m (H). Its temperature and emissivity are 60 °C and 0.94, respectively, standing on the floor of the room, shown in Fig. 1.

Distribution proportions Pr_i of the radiant heat to each inner surface are obtained from references [6] as 0.0693, 0.1310, 0.1135, 0.1817, 0.0432, 0.1987, 0.0269, 0.0735, 0.0275, 0.0350, 0.0193, 0.0599 and 0.0151. According to the proportions, cooling loads in the occupied zone of the room, CL_E, can be calculated as shown in Figs. 2 and 3 still under the effect of the unit step of radiant heat when the room is of heavy and light construction, respectively. The cooling loads are not shown in Figs. 2 and 3 when all the radiant heat affects the ceiling because they are much smaller compared with the others.

It can be seen from Figs. 2 and 3 that there are big differences between the cooling loads calculated by each method and the calculation example, no matter what kind the room is, heavy, medium or light construction. The same conclusion can be got when the position of the equipment is changed to the center of the floor and other conditions are kept unchanged (distribution ratio is still obtained from reference [6] and the figure is omitted). So the accurate method should be calculating cooling load based on actual distribution ratio to each of the inside surfaces of a room.

3. Calculation principle of the Pr_i and its main influencing factors

When an air conditioned room is affected by radiant heat, the heat can be got only through many times of heat absorption and reflection between the heat source and the inside surfaces of the building. Assumed that each surface can be thought as diffuse-gray one, and the surface of heat source is labeled as 0, the other surfaces of indoor enclosure are labeled as $1, 2, \ldots, n$, respectively (in this example $n = 13$, shown in Fig. 1). Then there are $n + 1$ of gray surfaces in total which made up a close cavum and radiation heat transfer happens.

According to the law of conservation of energy, as to surface i, following equation can be got,

$$\frac{E_{bi} - J_i}{(1-\varepsilon_i)/(A_i\varepsilon_i)} + \sum_{j=0}^{n} \frac{J_j - J_i}{(A_i F_{i,j})^{-1}} = 0 \quad (i = 0, 1, 2, \ldots, n) \tag{1}$$

Considering integer property of view factor, $\sum_{i=0}^{n} F_{i,j} = 1$, formula (1) can be written as:

$$\sum_{j=0}^{n} F_{i,j} J_j - \frac{J_j}{1-\varepsilon_i} = \frac{\varepsilon_i}{\varepsilon_i - 1} E_{bi} \quad (i = 0, 1, 2, \ldots, n) \tag{2}$$

The matrix of the formula (2) is

$$[M] \cdot [J] = [C] \tag{3}$$

where

$$[M] = \begin{bmatrix} F_{0,0} + \frac{1}{\varepsilon_0 - 1} & F_{0,1} & F_{0,2} & \cdots & F_{0,n} \\ F_{1,0} & F_{1,1} + \frac{1}{\varepsilon_1 - 1} & F_{1,2} & \cdots & F_{1,n} \\ F_{2,0} & F_{2,1} & F_{2,2} + \frac{1}{\varepsilon_2 - 1} & \cdots & F_{2,n} \\ F_{n,0} & F_{n,1} & F_{n,2} & \cdots & F_{n,n} + \frac{1}{\varepsilon_n - 1} \end{bmatrix}$$

$$[J] = [\, J_0, \quad J_1, \quad J_2, \quad \ldots, \quad J_n \,]^{\mathrm{T}}$$

$$[C] = \left[\frac{\varepsilon_0 \sigma_b T_0^4}{\varepsilon_0 - 1}, \quad \frac{\varepsilon_1 \sigma_b T^4}{\varepsilon_2 - 1}, \quad \ldots, \quad \frac{\varepsilon_n \sigma_b T^4}{\varepsilon_n} \right]^{\mathrm{T}}$$

Using iterative method through computer, J_0 (effective radiation of heat source surface) and J_i (effective radiation of each indoor surface) at different temperature can be calculated. Therefore the radiant heat that heat sources sent out can be expressed as,

$$Q_{r0} = \frac{E_{b0} - J_0}{(1-\varepsilon_0)/(A_0\varepsilon_0)} = \frac{\varepsilon_0 A_0 (\sigma_b T_0^4 - J_0)}{1 - \varepsilon_0} \tag{4}$$

Thus the radiant heat each inner surface gains is as follow,

$$Q_{r0,i} = \frac{J_0 - J_i}{(A_0 F_{0,i})^{-1}} = A_0 F_{0,i}(J_0 - J_i) \quad (i = 1, 2, \ldots, n) \tag{5}$$

The percentage of radiant heat on each surface is as

$$P_{rj} = \frac{Q_{r0,i}}{Q_{r0}} = \frac{F_{0,i}(J_0 - J_i)(1-\varepsilon_0)}{\varepsilon_0(\sigma_b T_0^4 - J_0)} \quad (i = 1, 2, \ldots, n) \tag{6}$$

Table 2
Pr_i of an occupant when facing (or backing against) the north wall/%

C/m	Pr_1	Pr_2	Pr_3	Pr_4	Pr_5	Pr_6	Pr_7	Pr_8	Pr_9	Pr_{10}	Pr_{11}	Pr_{12}	Pr_{13}	B/m
0.10	4.27	15.88	5.31	20.65	2.59	31.99	1.25	12.90	1.54	1.43	1.14	0.34	0.68	
0.83	5.88	22.81	5.57	28.87	2.64	17.73	1.73	7.15	1.79	1.72	1.14	2.17	0.79	0.50
1.73	9.48	25.21	5.85	31.37	2.70	9.75	2.78	3.93	1.95	1.92	1.15	3.05	0.86	
0.50	5.48	13.83	5.84	28.18	3.18	24.55	1.61	9.90	1.80	1.96	1.30	1.54	0.80	
0.83	6.41	15.53	6.01	31.00	3.24	19.47	1.88	7.85	1.93	2.11	1.31	2.40	0.85	0.90
1.73	10.54	17.57	6.33	33.80	3.37	10.84	3.09	4.37	2.13	2.32	1.34	3.34	0.94	
0.50	6.03	6.05	6.40	29.69	6.05	25.53	1.77	10.29	1.95	1.83	1.83	1.70	0.87	
0.83	7.10	6.42	6.59	32.68	6.42	20.80	2.08	8.39	2.11	1.90	1.90	2.65	0.93	2.03
1.73	11.73	7.02	6.96	35.75	7.02	12.07	3.44	4.87	2.36	2.01	2.01	3.70	1.05	

From the formula (6) it is seen that the main influencing factors of Pr_i can be divided into 2 kinds, one is concerned with the heat source, and the other is concerned with the room. Shape and location of the heat source are two main factors of it. As to an indoor heat source, the change of shape ordinarily indicates the change of indoor heat equipments because the shape of an occupant or lighting lamp is relatively fixed in common air conditioned rooms. The change of shape directly affects the view factor between the source and the enclosure of the room, and further affects the heat quantity each surface absorbs; The change of location of the heat source will lead to the change of distance between the source and the inside surfaces, and will further affect the heat quantity the source projects to each inside surface and affect the distribution ratio.

The characteristic of a room includes its shape, size, height and the properties of inside surfaces of the enclosure. The variation of room shape etc. as well as that of equipments also affects the distribution ratio of radiant heat Pr_i. As to the inside surfaces of building, their properties of surfaces will directly affect the heat absorption and reflection of the radiant heat they receive. So the Pr_i on them is affected consequently.

4. Pr_i of diversified indoor heat sources

4.1. Occupant

Supposing that an occupant is seated when working in an air conditioned room, and thinking of the effect of shelter of his clothes, his body is considered as a cubic heat source of 0.45 m × 0.15 m × 1.20 m (H). The effective radiant area is 1.508 m^2. When practically calculating the distribution ratio of radiant heat Pr_i, and considering a room with underfloor air conditioning system for instance, the room enclosure is divided into 13 surfaces (shown in Fig. 1). The temperature of each surface is different. The average temperature is 22 °C. Surface emissivity is assumed as 0.90. Surface emissivity of the occupant is assumed as 0.98. And his surface temperature is taken as 28.6 °C [7]. Assume the location of the occupant in the room is $B = 0.5$ m, quarter of width of the room and half of width of the room away from the east wall, respectively, and $C = 0.1$ m, quarter of width of the room and half of width of the room away from the north wall. The distribution ratios of radiant heat Pr_i on inside surfaces can be got and are listed in Table 2 when the occupant is facing (or backing against) the north wall. The results when the body is laterally facing the north wall please consult reference [6], which is omitted here because of the limited space.

From Table 2 we can conclude:

(1) Distance between the body and the wall has a great effect on Pr_i. It increases with the increasing of the distance between heat source and wall. The shorter the distance, the acuter the increase, and vice versa.
(2) Pr_i increases with heat source's approaching to the middle location of the wall, and vice versa.
(3) Pr_i will be greater when an occupant is facing (or backing against) the building enclosure than that when he is laterally facing the building enclosure.
(4) Wherever the location of the body in the room is, Pr_i on the floor is always greater. It ranges between 20% and 36% along with the changing of the location of the body. And it gets the maximum when the body is in the middle of the room. Pr_i on the ceiling is less and currently ranges 5% to 8%.
(5) Wherever the location of the body in the room is, most of Pr_i is distributed to the building enclosure or furniture in the occupied zone, accounting for 81% to 90%; and Pr_i distributed on non-occupied zone only accounts for 10% to 19%.

4.2. Lighting fixtures

Incandescent lamps and daylight lamps for lighting in an air conditioned room are currently covered by lamp-chimneys or bedded in the ceiling. To calculate easily, incandescent lamps and daylight lamps are simplified to plane heat resources of 0.20 m × 0.20 m and 0.15 m × 1.0 m respectively, according to the common size of lamp-chimneys.

When practically calculating the distribution ratio of radiant heat of incandescent lamps, its surface emissivity is supposed as 0.97 and the imaginary plane temperature is 145 °C or 60 °C [7], assuming the situation of each inside surface is the same to the forward. The calculated results

Table 3
Pr_i of radiant heat of an incandescent lamp bedded in the ceiling/%

C/m	Pr_1	Pr_2	Pr_3	Pr_4	Pr_5	Pr_6	Pr_7	Pr_8	Pr_9	Pr_{10}	Pr_{11}	Pr_{12}	Pr_{13}	$t/$
0.80	6.18	8.36	0.00	29.38	8.36	9.13	2.48	3.33	2.32	4.31	4.31	20.51	1.31	60
1.70	8.68	9.10	0.00	31.72	9.10	8.95	3.49	3.26	5.14	4.77	4.77	8.07	2.91	60
1.70	8.67	9.09	0.00	32.19	9.09	8.94	3.37	3.30	5.09	4.73	4.73	7.99	2.81	145

Table 4
Pr_i of radiant heat of a daylight lamp bedded in the ceiling/%

C/m	Pr_1	Pr_2	Pr_3	Pr_4	Pr_5	Pr_6	Pr_7	Pr_8	Pr_9	Pr_{10}	Pr_{11}	Pr_{12}	Pr_{13}	B/m
0.40	6.19	8.27	0.00	30.60	8.27	8.58	2.22	3.26	2.32	4.14	4.14	20.78	1.21	
1.30	8.56	8.99	0.00	32.96	8.99	8.82	3.07	3.35	5.21	4.57	4.57	8.17	2.70	2.18

Table 5
Pr_i of equipments whose shapes are 1.0 m × 1.0 m × 1.0 m (H)/%

C/m	Pr_1	Pr_2	Pr_3	Pr_4	Pr_5	Pr_6	Pr_7	Pr_8	Pr_9	Pr_{10}	Pr_{11}	Pr_{12}	Pr_{13}	B/m
0.40	5.17	19.31	10.77	22.38	4.36	18.33	2.01	6.78	1.89	2.91	2.07	2.96	1.05	
1.30	9.00	21.21	11.52	25.11	4.62	9.28	3.49	3.43	2.23	3.18	2.17	3.51	1.23	0.63
0.40	5.67	8.36	11.75	24.50	8.36	10.70	2.20	7.28	2.05	2.85	2.95	8.27	1.13	
1.30	9.97	9.13	12.61	27.43	9.13	10.27	3.87	3.80	2.45	3.05	3.05	3.85	1.35	1.75

indicate that the change regularity of Pr_i is similar to that of the occupant when location of lamp changes. Here lists the results when the lamp is located on the ceiling between the east wall and the west one, and is away from the north wall, quarter of the width of room and half of the width (see Table 3). It can be imagined that, if the incandescent lamps are not bedded in the ceiling, but located at some height in the room, the distribution ratio of radiant heat Pr_i on floor or furniture will be greater and that on surrounding building enclosure will be less.

When actually calculating the distribution ratio of radiant heat of a daylight lamp Pr_i, we assume that it is a plane heat resource of 37.5 °C [7], and the other situations are the same as that of incandescent lamps. The calculation results indicate that, when the location of daylight lamp changes, the change regularity of Pr_i is the same to the corresponding one of the occupant whether the lamp tube is parallel with or vertical to the north wall. Table 4 lists the results when the daylight lamp is apart from quarter of the width of the room and half of the width when we assume that it is located on the ceiling between the east and west wall and the lamp cube is vertical to the north. Also, if it is not bedded in the ceiling, the distribution ratio on the floor will increase with the reducing of the distance between it and the floor.

From Tables 3 and 4 we can conclude:

(1) The conclusion (1) and (2) educed for the occupant are also right here.
(2) Wherever the location of lamps (include incandescent lamps and daylight lamps) on the ceiling is, the distribution ratio Pr_i on the floor will always get the maximum (about 30%), however the Pr_i on the ceiling is almost zero.
(3) Wherever the location of lamps in the room is, the Pr_i usually accounts for 70% in occupied zone and 30% in non-occupied zone.
(4) Surface temperature of lighting fixtures has little effect on its Pr_i.

4.3. Equipment

The equipment in an air conditioned room may have diverse surface temperatures and emissivities and its shape and the location in the room may vary greatly. This causes great trouble in calculating its distribution ratio Pr_i. But the following calculation results indicate that its surface temperature and its emissivity have little effect on the Pr_i. Table 5 lists the distribution ratio Pr_i of the equipment on each surface when assuming the surface temperature of equipment is 60 °C, and the emissivity is 0.94, and their shapes are 1.0 m × 1.0 m × 1.0 m (H) (the temperature of inside surfaces of the building is the same with that above). When the shape is changed to 0.5 m × 2.0 m × 2.0 m (H) and the other situations are kept the same, the results are shown in Table 6. And the comparison of the results with the former is in Table 7 when assuming $B = 1.75$ m, $C = 1.30$ m, surface temperature $t = 200$ °C and emissivity $\varepsilon = 0.2$.

From Tables 5, 6 and 7 we can conclude:

(1) The conclusion (1) and (2) educed for the occupant are also right here.
(2) Because the shape of equipment varies greatly, the Pr_i on the floor may not be the maximum but is quite bigger anyway, usually 20% to 30%. However the Pr_i on the ceiling will be greater than that of other kinds of heat sources, usually 11% to 13%. When the bigger surface

Table 6
Pr_i of equipments whose shapes are $0.5\text{ m} \times 2.0\text{ m} \times 2.0\text{ m}\ (H)/\%$

C/m	Pr_1	Pr_2	Pr_3	Pr_4	Pr_5	Pr_6	Pr_7	Pr_8	Pr_9	Pr_{10}	Pr_{11}	Pr_{12}	Pr_{13}	B/m
0.65	6.93	13.10	11.35	18.71	4.32	19.87	2.69	7.35	2.75	3.50	1.93	5.99	1.51	
1.55	11.71	14.34	12.22	20.79	4.56	11.51	4.33	4.26	3.52	3.81	2.00	5.53	1.94	0.50
0.65	7.22	7.20	11.82	19.53	7.20	20.61	2.80	7.62	2.86	2.67	2.67	6.24	1.57	
1.55	11.70	7.83	12.73	21.70	7.83	12.05	4.53	4.46	3.68	2.84	2.84	5.78	2.03	1.25

Table 7
The effect of surface temperature and emissivity on $Pr_i/\%$

ε	Pr_1	Pr_2	Pr_3	Pr_4	Pr_5	Pr_6	Pr_7	Pr_8	Pr_9	Pr_{10}	Pr_{11}	Pr_{12}	Pr_{13}	$t/$
0.94	9.97	9.13	12.59	27.46	9.13	10.27	3.87	3.30	2.45	3.05	3.05	3.85	1.35	60
0.97	9.97	9.13	12.83	26.97	9.13	10.27	4.02	3.74	2.48	3.08	3.08	3.89	1.41	200
0.20	9.99	9.14	11.57	29.59	9.14	10.29	3.22	4.04	2.35	2.92	2.92	3.69	1.12	60

of the equipment is near to a certain wall, the Pr_i on this wall will reach the maximum value.

(3) The variety of shape of equipment has an effect on Pr_i on the surfaces in occupied zone and in non-occupied zone. But the Pr_i still accounts highly for 70% to 80% in occupied zone and accounts for 20% to 30% in non-occupied zone.

(4) The surface temperature and emissivity has a low effect on Pr_i , which can be ignored.

5. Conclusion

It is found that the former approximate methods of the distribution ratio of radiant heat will result in serious errors to the cooling load calculation, such as the method of evenly-distributed by area on inside surfaces of a room, when accurately calculating the loads formed by strong radiant heat from indoor heat sources, no matter what kinds the rooms are. The accurate method should be calculating the cooling loads based on the actual distribution ratio of the radiant heat to each of the inside surfaces of the room.

A series of regularity results for indoor heat sources are summarized and obtained according to calculation and analysis under the several familiar conditions of distribution ratio of radiant heat on each inside surface of a room. And these results also adopt for traditional mixed air conditioning systems. Due to the indetermination of the factors which affect this problem and the complexity itself, it is hard to present general and precise results of distribution ratio for all kinds of indoor heat sources. But the results got from the calculation program can be directly used in actual calculation of the loads after inputting the conditions of each room and each heat source.

Acknowledgements

The project is financially supported by National Natural Science Foundation of China. And the author also wants to express thanks to Ms A.L. Lian for her translation of the paper.

References

[1] P.L. Chen, et al., Theory and Method of the Calculation of Air Conditioning Loads, Tongji University Press, Shanghai, 1987 (in Chinese).
[2] T. Kusuda, N.B.S.L.D., The computer program for heating and cooling loads in buildings, NBS-BSS-69, 1976.
[3] L.B.L., DOE-2 reference manual, Lawrence Berkeley Laboratory and Los Alamos National Laboratory, LBL Report No LBL-8706, Rev. 2, 1981.
[4] Z.W. Lian, et al., Mathematical model for calculating load in a conditioned space with upward displacement, J. Refrig. 3 (1996) (in Chinese).
[5] R.Y. Gu, W.F. Wu, Simulation on the coefficient of convective heat transfer on inside surfaces of building enclosure of an air conditioned room, in: Computational Heat Transfer Research and Application, High Education Press, Beijing, 1994, pp. 192–195 (in Chinese).
[6] Z.W. Lian, Effect of the indoor heat sources on loads in a conditioned space with upward displacement, Ph.D. Dissertation, Xi'an Jiao Tong University, 1995 (in Chinese).
[7] Z.W. Lian et al., Radiant heat gains from a heat source inside a conditioned space, in: Pan. Pacific Symposium on Building and Urban Environmental Conditioning in Asia, Japan, 1995, pp. 53–59.