# A PROPOSAL FOR AN ESTIMATION METHOD OF THERMAL LOAD AND SPACE RADIANT ENVIRONMENT FOR HVAC SYSTEM DESIGN DEVELOPED BY APPLYING DOE TECHNIQUE

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## ABSTRACT

This study aims to develop a simplified estimation method of the thermal design load and space radiant environment in order to achieve adequate and economical HVAC equipment sizing and confirmation of thermal comfort. The author proposes a new definition for the 2.5% thermal design load and corresponding operative temperature (OT). The 2.5% design load is the load which occurs at 2.5% cumulative frequency of occurrence during the summer or winter months. Hour-by-hour dynamic simulation through a year is necessary to obtain the 2.5% design load and corresponding OT. The author developed a simplified estimation method for these parameters in the case of office buildings in Tokyo through simulation analysis applying the DOE technique.

## INTRODUCTION

The most popular conventional method of thermal design load calculation for HVAC systems is to simulate dynamic thermal load using design weather data on a winter or summer day and to obtain the peak load on a periodical steady-state day. This method is based on precise load calculation, but may be responsible for oversizing of equipment because of the safe-side assumptions of design weather conditions and other variables. It is desirable to apply the concept of tolerance of the frequent occurrence of conditions exceeding the design parameters not to weather data but thermal load. The author proposes a new and more reasonable definition of design load. The design load based on the new definition is obtained by hour-by hour dynamic simulation over one year using a computer, and is not convenient for practical design procedures. The author found a solution to this problem by developing a simplified method applying the DOE technique, which is a numerical statistics method.

Confirmation of thermal comfort under severe weather conditions is very important in addition to adequate and economical equipment sizing. HVAC systems control the space air temperature and humidity. Spaces where air temperature and humidity are kept constant have been considered to be comfortable. However, even if space air temperature and humidity are kept constant, radiation from windows and other surfaces in the space will vary and then people may not feel comfortable. The radiant environment becomes worse with increase in demand for heating or cooling. It is important to estimate the space radiant environment when the thermal peak load occurs for ensuring thermal comfort in severe radiant conditions.

## DEFINITION OF 2.5% DESIGN THERMAL LOAD AND CORRESPONDING OT

New definition of the thermal design load is based on the concept that the space thermal load required to keep space air temperature at the set point value is allowed to exceed the 2.5% design load only in 2.5% of the cooling or heating hours during four months of each season. The precise value of the 2.5% thermal design load can be obtained from the following procedure. At first, the hourly dynamic thermal loads are simulated through a year and then the loads are ranked in magnitude for each season. The thermal load that ranks the order equal to 2.5% of the number of cooling or heating hours is selected as the 2.5% design load. For conventional thermal design load calculation in Japan, the engineers have popularly used TAC2.5% design outdoor air temperatures. Those design outdoor air temperatures were made based on the summer cumulative frequency of 2.5% and the winter cumulative frequency of 97.5% as the risk of exceeding. The author proposed 2.5% risk of exceeding applied to the thermal loads instead of weather data.

The OTs at points 0.5m, 2.5m and 4.5m from the window are used as indices of the radiant environment in a perimeter zone and the mean value of the OT at 5 points in the interior zone is used as an index of the interior zone radiant environment. The OT corresponding to the 2.5% thermal design load is the mean value of OTs corresponding to the thermal loads which have a cumulative frequency of occurrence of from 2% to 3%. This design load and the corresponding OT are obtained by hour-by-hour simulation over a period of one year with the space air always kept at a constant temperature and humidity during conditioning hours.

## TECHNIQUE FOR DEVELOPING A SIMPLIFIED ESTIMATION METHOD

Although the method described above enables determination of proper equipment capacity and confirmation of thermal comfort, it is not practical for design procedures. The simplified method proposed in this paper enables the engineer to estimate the thermal de-

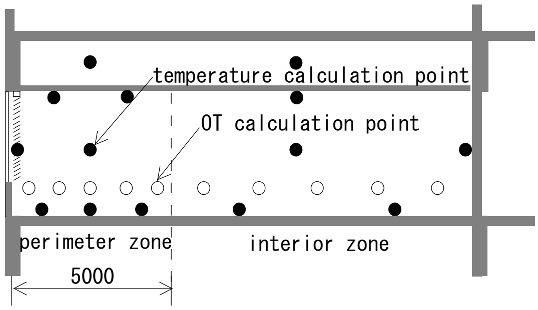



**Figure 1 The section of an office space**

sign load and corresponding OT through only several additions of mathematical parameters for office buildings in Tokyo. The results of such a calculation will agree fairly well with those obtained by precise simulation throughout a year.

There is an experimental technique generally known as the "Method for Design of Experiment (Method for DOE)". This method was developed to detect the effects of various factors by an experiment of minimum scale through optimal arrangement of such factors. The author applied the DOE technique to computer simulation analysis. In this study, orthogonal array L64 was used and the experiment was conducted numerically by computer simulation as a substitute for actual experiment. As a result, it was proven that the magnitude of various factors exerting an influence on thermal design load and corresponding OT can statistically be determined and a simplified method can be constructed.

## TYPICAL OFFICE SPACE MODEL

A typical office space model with the section shown in Figure 1 was assumed for the precise simulation analysis which is necessary for developing a simplified estimation method. The standard conditions are as follows.
Location: Tokyo
Floor: Typical floor
Orientation: South for cooling season and north for heating season
Fenestration ratio (ratio of fenestration area to building envelope area): 45%
Horizontal projection: None
Envelope insulation grade: Medium (U-factor of exterior wall is 1.6W/m$^2$K and fenestration is single glazing)
Space depth: 12m
Zoning: Space was divided into perimeter zone and interior zone
Perimeter zone depth: 5m
Maximum value of internal heat generation: Occupancy 0.2 person/m$^2$, lighting and equipment 25W/m$^2$ for cooling design conditions, 25% of these values were assumed for heating design conditions
Infiltration: Air exchange rate based on perimeter zone volume 0.2cycles/hour
Thermal capacity of internal furniture: 1.2 kJ/m$^3$K
Blind operation: Regulation depending on intensity of transmitted solar radiation and depth of sunlit floor
Heating and Air-conditioning system: Each zone was conditioned by one air handling unit (AHU)
Setting point for air temperature and humidity: 26℃ and 50% for cooling; 22℃ and 50% for heating
Heating / Air-conditioning hours: weekdays 9:00 to 18:00, Saturday 9:00 to 13:00 (warming up and pulling down: 8:00 to 9:00)
Heating / Air-conditioning period: June to September for cooling, December to March for heating
Intake outdoor airflow rate: 4 CMH/m$^2$ (outdoor air is shut off during warming up and pulling down)

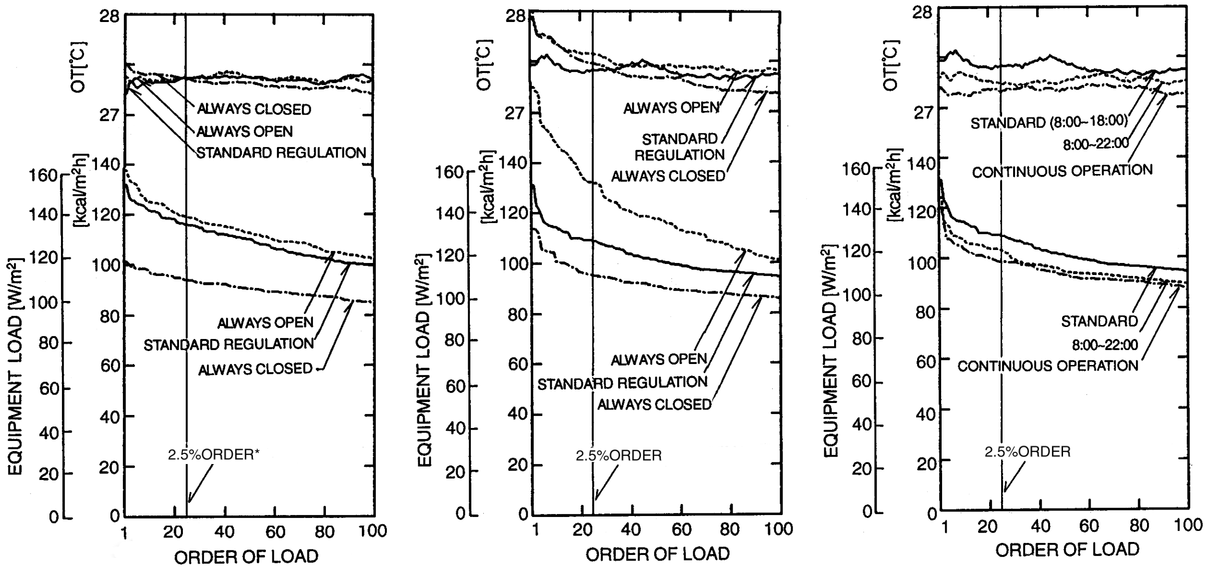
## PRECISE SIMULATION

Precise simulation analysis is necessary to develop a simplified estimation method. Air temperatures and surface temperatures in the office space were simulated by solving convective and radiant heat balance equations. 4 points of air temperature and 11 points of space surface temperature as shown in Figure 1 were treated as unknowns. The horizontal distribution of the OT was calculated at 5 points in each of the perimeter and interior zones. The thermal load of the HVAC equipment in each zone was also simulated. It was assumed that the thermal load of equipment was constant during the warming up or pulling down hour and that the space air temperature reached the set point just at the end of the hour. Through simulation under such assumptions, the thermal design load can be obtained for the case where the peak load occurs at the warming up or pulling down hour. The weather data for an average Tokyo year was used for the annual simulation.

## FACTORS AFFECTING THERMAL LOAD AND OT

Cases for precise simulation were determined by the DOE technique considering factors affecting the thermal design load and corresponding OT. The factors which influence the thermal design load and corresponding OT are classified into three types. The first type interact with each other, and the primary and interaction effects of such factors should be obtained through simulation analysis by the DOE technique. The second type, such as lighting power and occupant density, influence thermal load and OT independently and the effects can be obtained easily without consideration of interaction with other factors. The third type are not design factors but indefinite space usage factors such as blind operation, the schedules of lighting and occupants, cooling or heating hours, holidays and the amount of internal furniture.

The author selected factors of the first type and set two or four levels for each factor. The following conditions of each level are for office buildings in Tokyo.




(a) Blind regulation (south) (b) Blind regulation (east) (c) Air-conditioning hours(east)

*2.5%ORDER is the order corresponding to 2.5% seasonal cumulative frequency of occurrence

**Figure 2 Effects of space usage factors on HVAC equipment load and OT at severe cooling conditions**

A) Fenestration orientation: South, east, north and west
B) Fenestration area ratio: 30% and 60%
C) Envelope insulation grade: High and low
D) Depth of horizontal projection: 0m and 1m
E) Floor: Typical floor and top floor
F) Space depth: 8m, 12m, 16m and 20m
G) Intake outdoor airflow rate: 2 CMH/$m^2$ and 4 CMH/$m^2$
H) Design space air temperature: 26℃ and 28℃ for cooling design; 22℃ and 20℃ for heating design

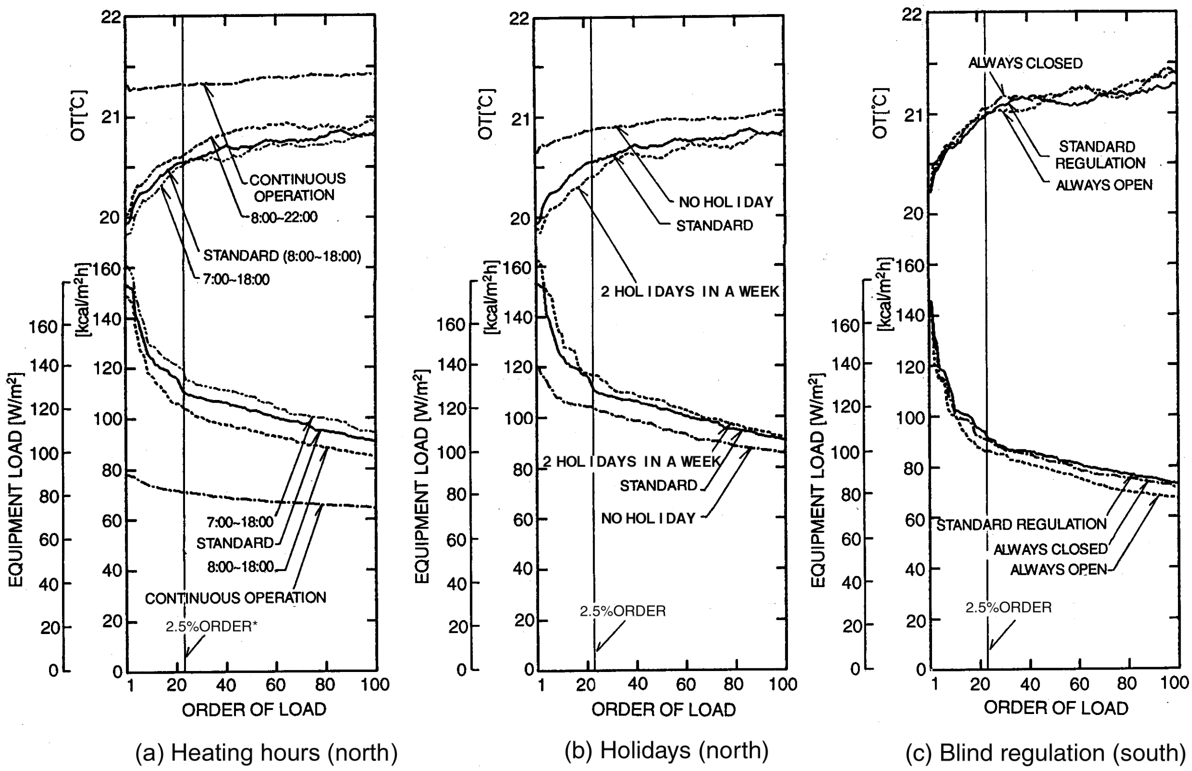
It is predicted that interaction effects exist between the following factors.
A × B) Fenestration orientation and fenestration area ratio
A × D) Fenestration orientation and depth of horizontal projection
B × D) Fenestration area ratio and depth of horizontal projection
B × E) Fenestration area ratio and floor
C × E) Envelope insulation grade and floor
G × H) Intake outdoor airflow rate and design space air temperature
The factors were distributed in an orthogonal array L64 and 64 simulations were conducted.

The following factors are of the second type and their primary effects were obtained by simulation at two levels for each factor.

I) Power input of lights and equipment: 25W/$m^2$ and 50W/$m^2$
J) Occupant density: 0.1 person/$m^2$ and 0.2 person/$m^2$

The factors of the third type is associated with space usage. The difference of the thermal design load due to the different level of these factors should be considered as the tolerance. In precise simulation analysis and a simplified estimation method, these factors are fixed as standard conditions. The author chose several factors of the third type which have rather large effects on the thermal load and OT and examined how these factors affect the thermal load and OT when the thermal load is significantly large.

The results for cooling season are shown in Figure 2. Blind regulation and air-conditioning hours are picked up as the factors of the third type. Hourly HVAC equipment loads for perimeter zone and OT at the center of the perimeter zone were obtained from simulation through a year. Hourly loads were ranked in magnitude and those within the 100th order are shown with corresponding perimeter zone OT in Figure 2. The curves of corresponding OT are smoothed by averaging 10 OTs before and behind the order. 2.5% of cooling hours during four summer months, June through September is 23 hour in the standard condition and then the 2.5% cooling design load and corresponding OT are the value at the 23rd order in Figure 2. Three cases




*2.5%ORDER is the order corresponding to 2.5% seasonal cumulative frequency of occurrence

**Figure 3 Effects of space usage factors on HVAC equipment load and OT at severe heating conditions**

is supposed for blind regulation. Those are "always open," "standard regulation" and "always closed". "Standard regulation" is the standard condition. In the case of standard blind regulation, judgment whether it is necessary to close blind is made according to intensity of transmitted solar radiation and depth of sunlit floor. "Always closed" means that the slat angle is fixed to minimize transmitted solar radiation. Figure 2(a) shows the results for south window and Figure 2(b) is the results for east window. If the blinds are always closed during the cooling period, the cooling design load may be less by about 15% than the value in the case of standard blind regulation. Figure 2(c) shows the results of the comparison in three cases of different air-conditioning hours. In a space faced east, the peak load may occur during pulling down hour in intermittent air-conditioning operation. The design load in continuous operation may be less by about 10% than standard operation.

The effects due to the different conditions of the third type factors for heating season are shown in Figure 3. Heating hours, holidays and blind regulation are picked up as the third type factors. If heating hour per day increases by 4h, the heating design load decrease by 10% and OT at a point 2.5m from the window becomes higher by 0.1K.

## RESULTS OF ANALYSIS AND PROPOSAL OF A SIMPLIFIED METHOD

The thermal design loads and corresoponding OTs in 64 cases were obtained from the L64 numerical experiment. From the analysis of variance, the factors were judged as significant at 5% for each characteristic and only the factors at this level were investigated as to whether their effects should be considered in a simplified estimation method. The small effects which can be ignored in practice were removed and the rest of the factors were reconfirmed as to whether they still had 5% significance. The mean of the value of all cases and estimated effect of each factor at each level were obtained from the analysis. For convenience of practical use, the author replaced the mean value and estimated effects by the standard value and the correcting values respectively. The standard value is the thermal design load or corresponding OT under standard conditions and the correcting values are used for correcting the standard value to obtain the thermal design load or corresponding OT for a given space. Tables 1 and 3 indicate the names of significant factors, the standard value, the 95% confidence limit and the correcting value of each factor at each level for thermal design load and corresponding OT.

The standard value $q_0$ in Table 1(a) is the cooling design load for each of perimeter zone and interior zone

**Table 1 (a) Standard values and correcting values for simple estimation of the 2.5% thermal design load (Cooling load)**

(W/m2)

| | | | | | Perimeter cooling load | | | | Interior cooling load | |
|---|---|---|---|---|---|---|---|---|---|---|
| Standard value $q_o$ ± Confidence limit | | | | | 137 (103)* ± 5 | | | | 92 (56) ± 3 | |
| Correcting value $\Delta q_k$ | With no projection | Fenestration area ratio | 30% | S W N E | − 13 | − 14 | − 40 | − 20 | − | |
| | | | 45% | S W N E | 0 | 1 | − 31 | − 7 | | |
| | | | 60% | S W N E | 13 | 17 | − 24 | 7 | | |
| | With projection | Fenestration area ratio | 30% | S W N E | − 45 | − 33 | − 42 | − 42 | | |
| | | | 45% | S W N E | − 37 | − 21 | − 40 | − 34 | | |
| | | | 60% | S W N E | − 29 | − 10 | − 36 | − 26 | | |
| | Intake OA flow rate 2 4 (CMH/m2) | | | | − 12 (0) | | 0 | | − 12 (0) | 0 |
| | Design space temp. 26 28 (℃) | | | | 0 | | − 13 | | 0 | − 9 |
| | Lights and equipment 25 50 (W/m2) | | | | 0 | | 29 | | 0 | 29 |
| | Occupants 0.1 0.2 (person/m2) | | | | − 12 (− 6) | | 0 | | − 12 (− 6) | 0 |

Note: * In case of existing latent heat besides sensible heat, both total and sensible heat rates are shown. The parenthesized values are sensible heat.

**Table 1 (b) Standard values and correcting values for simple estimation of the 2.5% thermal design load (Heating load)**

(W/m2)

| | | | | Perimeter heating load | | | | Interior heating load | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Standard value $q_o$ ± Confidence limit | | | | 124 (102)* ± 6 | | | | 93 (71) ± 5 | | | |
| Correcting value $\Delta q_k$ | Orientation | | S W N E | − 19 | − 2 | 0 | − 13 | − | | | |
| | Typical floor | Insulation ** | H M L | − 16 | 0 | 16 | | − 9 | 0 | 9 | |
| | Top floor | Insulation | H M L | 3 | 15 | 26 | | 10 | 17 | 23 | |
| | Space depth 8 12 16 20 (m) | | | 13 | 0 | − 7 | − 12 | 22 | 0 | − 10 | − 19 |
| | Intake OA flow rate 2 4 (CMH/m2) | | | − 16 (− 8) | | 0 | | − 16 (− 8) | | 0 | |
| | Design space temp. 26 28 (℃) | | | 0 | | − 15 | | 0 | | − 13 | |

Note: * In case of existing latent heat besides sensible heat, both total and sensible heat rates are shown. The parenthesized values are sensible heat.
** Insulation grade H: High, M: Medium, L: Low See Table 2 on detail of insulation grade.

**Table 2 U-factors for each envelope insulation grade**

(W/m2K)

| Insulation grade | | | High | Medium | Low |
|---|---|---|---|---|---|
| Single glazing | Fenestration area ratio | 30% | 1.0 | 2.3 | 3.7 |
| | | 45% | − | 1.6 | 3.4 |
| | | 60% | − | 0.5 | 2.8 |
| Double glazing | Fenestration area ratio | 30% | 1.9 | 3.1 | 4.5 |
| | | 45% | 1.6 | 3.3 | 5.0 |
| | | 60% | 1.0 | 3.4 | 5.7 |

**Table 3 (a) Standard values and correcting values for simple estimation of operative temperature corresponding with the 2.5% thermal design load (Cooling)**

**((Zone representative OT))**

(℃)

| Zone | | | | | Perimeter (Center point)* | Interior (Zone average) |
|---|---|---|---|---|---|---|
| Standard value $OT_o$ ± Confidence limit | | | | | 27.29 ± 0.08 | 26.92 ± 0.05 |
| Correcting value $\Delta OT$ | With no projection | Fenestration area ratio | 30% | S W N E | −.15 .11 −.22 −.11 | − |
| | | | 45% | S W N E | 0 .39 −.03 .09 | |
| | | | 60% | S W N E | .15 .66 .15 .30 | |
| | With projection | Fenestration area ratio | 30% | S W N E | −.39 −.13 −.47 −.36 | |
| | | | 45% | S W N E | −.30 .09 −.33 −.21 | |
| | | | 60% | S W N E | −.21 .31 −.21 −.06 | |
| | Floor | | Typical | Top | 0 .08 | 0 .08 |
| | Design space temp. | | 26 | 28 (℃) | 0 1.83 | 0 1.87 |
| | Lights | | 20 | 30 (W/m$^2$) | 0 .23 | 0 .25 |

* Point 2.5m from fenestration

**((Perimeter zone OT distribution))**

(℃)

| Zone | | | | | Perimeter | |
|---|---|---|---|---|---|---|
| Distance from fenestration | | | | | 0.5m | 4.5m |
| Standard value $OT_o$ ± Confidence limit | | | | | 27.70 ± 0.14 | 27.08 ± 0.06 |
| Correcting value $\Delta OT$ | With no projection | Fenestration area ratio | 30% | S W N E | −.45 −.13 −.63 −.44 | −.07 .11 −.11 −.04 |
| | | | 45% | S W N E | 0 .56 −.08 .09 | 0 .26 −.01 .08 |
| | | | 60% | S W N E | .45 1.26 .45 .62 | .07 .42 .08 .19 |
| | With projection | Fenestration area ratio | 30% | S W N E | −.77 −.45 −.94 −.76 | −.25 −.07 −.29 −.22 |
| | | | 45% | S W N E | −.50 .09 −.08 .10 | −.18 .08 −.19 −.10 |
| | | | 60% | S W N E | −.24 .57 −.22 −.07 | −.11 .24 −.10 .01 |
| | Floor | | Typical | Top | − | 0 .09 |
| | Design space temp. | | 26 | 28 (℃) | 0 1.83 | 0 1.85 |
| | Lights | | 20 | 30 (W/m$^2$) | 0 .19 | 0 .25 |

**Table 3 (b) Standard values and correcting values for simple estimation of operative temperature corresponding with the 2.5% thermal design load (Heating)**

**((Zone representative OT))**

(℃)

| Zone | | | | Perimeter (Center point)* | Interior (Zone average) |
|---|---|---|---|---|---|
| Standard value $OT_o$ ± Confidence limit | | | | 20.71 ± 0.12 | 20.93 ± 0.11 |
| Correcting value $\Delta OT$ | Orientation | | S W N E | .36 .05 0 .18 | − |
| | Typical floor | Insulation ** | H M L | .28 0 −.28 | .19 0 −.19 |
| | Top floor | Insulation | H M L | −.01 −.22 −.42 | −.15 −.27 −.39 |
| | Space depth | | 8 12 16 20 (m) | −.10 0 .11 .17 | −.16 0 .13 .24 |
| | Design space temp. | | 26 28 (℃) | 0 − 1.82 | 0 − 1.83 |

* Point 2.5m from fenestration

**((Perimeter zone OT distribution))**

(℃)

| Zone | | | | Perimeter (Center point)* | Interior (Zone average) |
|---|---|---|---|---|---|
| Standard value $OT_o$ ± Confidence limit | | | | 20.64 ± 0.13 | 20.82 ± 0.11 |
| Correcting value $\Delta OT$ | Orientation | | S W N E | .44 .04 0 .26 | .30 .05 0 .13 |
| | Typical floor | Insulation ** | H M L | .33 0 −.33 | .24 0 −.24 |
| | Top floor | Insulation | H M L | −.19 −.14 −.47 | −.09 −.26 −.42 |
| | Space depth | | 8 12 16 20 (m) | −.03 0 .09 .13 | −.13 0 .13 .21 |
| | Design space temp. | | 26 28 (℃) | 0 − 1.82 | 0 − 1.82 |

under the standard conditions. For example, the total and sensible heat of $q_0$ for perimeter zone is 137 and 103W/m$^2$, respectively. The confidence limit of 5W/m$^2$ for perimeter zone means that the error of the perimeter cooling design load obtained from this estimation method can be expected to be within 5W/m$^2$ at the possibility of 95%. For cooling design load, the factors such as depth of horizontal projection, fenestration area ratio, fenestration orientation, intake outdoor airflow rate and design space temperature are judged to be significant among all of the first type factors. On the other hand, the effects of the factors such as envelop insulation glade, floor and space depth are judged to be negligible. The correcting value $\Delta q_k$ is the value expected to be the difference of the cooling design load between the standard condition and the condition of each level for the factor $k$. Table 1(a) shows the standard value and the correcting value for the heating design load. 6 factors, i.e. fenestration orientation, floor, envelop insulation glade, space depth, intake outdoor airflow rate and design space temperature are selected as the significant factors among all of the first type factors. Table 2 gives the information of building envelop insulation glade.

Table 3 shows the standard value and the correcting value for OT corresponding to the cooling or heating design load of each zone. The representative OT for perimeter zone is OT at the center (2.5m distant from the window) of the zone and that for interior zone is the mean of OTs at 5 point in the zone. OTs in perimeter zone are significantly affected by the distance from the window and OTs at 0.5m and 4.5m distant from the window are also shown in Table 3

In the simplified estimation method, the thermal design load q [W/m$^2$] and corresponding OT [℃]can be obtained by the following equations.

$$q=q_0+\sum_k \Delta q_k \qquad \cdots(1)$$

$$OT=OT_0+\sum_k \Delta OT_k \qquad \cdots(2)$$

Where,

$q_0$ : the standard thermal design load [W/m$^2$]
$\Delta q_k$ : correcting thermal load for factor k [W/m$^2$]
$OT_0$ : the standard OT [℃]
$\Delta OT_k$: correcting OT for factor k [K]
$k$ : the factor affecting the thermal load and OT

Values of $q_0$, $\Delta q_k$, $OT_0$ and $\Delta OT_k$ can be obtained from Tables 1 and 3.

(Example)
The cooling design load and corresponding OT for perimeter zone under a given space conditions can be obtained as the following procedure.

Conditions;
A) fenestration Orientation: South
B) fenestration area ratio: 60%
C) envelop insulation glade: Medium
D) horizontal projection: None
E) floor: Top floor
F) space depth: 16m
G) intake outdoor airflow rate: 2CMH/m$^2$
H) design space temperature: 26℃ for cooling
I) lighting and equipment: 50W/m$^2$(Lighting 20W/m$^2$)
J) occupancy: 0.1person/m$^2$

(1) Estimation of the cooling design load (total of sensible and latent heat)
From Table 1(a),
standard value: 137 W/m$^2$
correcting value:
13 W/m$^2$ for fenestration orientation of south and fenestration area ratio of 60% with no projection
-12 W/m$^2$ for intake outdoor airflow rate of 2 CMH/m$^2$
0 W/m$^2$ for design space temperature of 26℃
29 W/m$^2$ for lighting and equipment of 50W/m$^2$
-12 W/m$^2$ for occupancy of 0.1 person/m$^2$
the cooling design load
= 37+13-12+0+29-12=155 W/m$^2$
Correction doesn't need for the factors such as envelop insulation glade, floor and space depth. The error of the estimated cooling design load is probably within 5W/m$^2$.

(2) Estimation of OT at the center of the perimeter zone corresponding to the cooling design load
From Table 3(a),
standard value: 27.29℃
correcting value:
0.15K for fenestration orientation of south and fenestration area ratio of 60% with no projection
0.08K for top floor
0 K for design space temperature of 26℃
0 K for lighting of 20W/m$^2$
OT at the center of the perimeter zone
= 27.29+0.15+0.08+0+0=27.52℃
Correction doesn't need for the factors such as envelop insulation glade, space depth, intake outdoor airflow rate and occupancy. The error of estimated OT is probably within 0.08K.
Similarly,
OT at the point from 0.5m distant from the window=27.70+0.45+0+0+0=28.15℃
OT at the point from 4.5m distant from the window=27.08+0.07+0.09+0+0=27.24℃

In this paper, the simplified estimation method is shown for office buildings in Tokyo. For other regions, this method can be applicable preparing the proper standard value and correcting value for the thermal load and OT through simulation applying DOE technique and analysis of variance. In such simulation, the conditions assumed for each level of the factors should be suitable to the regional climate and also weather data for each region should be used.

## CONCLUSION

1) The author tried to apply the concept of tolerance of the frequency occurrence of conditions exceeding the design parameters not to weather data but to thermal load and gave a new definition for the 2.5% thermal design load. This is obtained from hour-by-hour simulation throughout a year.

2) For confirmation of thermal comfort in a conditioned space, the author proposed the concept that severe radiant environments at the time when the thermal peak load occurs should be checked in the design of HVAC systems.

3) A technique of developing a simplified estimation method through precise simulation analysis was presented. In addition, a simplified estimation method for the 2.5% thermal design load and corresoponding OT are proposed for office buildings in Tokyo. This simplified estimation method can support to find speedy solution not for HVAC system design but also for planning of typical floor and building envelop. Besides, the effects of building factors on the thermal load and radiant environment can be easily understood by making reference to the correcting value of the thermal load and OT.

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