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Calculating Cooling Load Caused by Lights

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DONNES SERVANT A CALCULER LA CHARGE DE REFROIDISSEMENT DUE A L'ECLAIRAGE

SOMMAIRE

Une étude analytique de la charge de refroidissement due à l'alimentation des lumières en électricité a montré la nécessité d'une détermination expérimentale de quelques-uns des facteurs servant à calculer la charge de refroidissement. Une pièce calorimétrique de grandeur naturelle (10 x 14 x 13 pieds, d'étage à étage) a été construite et munie d'instruments à cette fin. Les résultats des essais confirment ceux de l'étude analytique, à savoir que le rapport entre le changement progressif de l'alimentation en électricité et l'augmentation subséquente de chaleur dans la pièce peut être exprimé sous forme exponentielle. Les résultats expérimentaux sont d'une utilité restreinte pour ce qui est du calcul des plans, puisqu'ils n'englobent pas de nombreux modèles de pièces, appareils d'éclairage, genres d'installations et systèmes de ventilation. Toutefois, les résultats expérimentaux ont permis d'obtenir un ensemble de données servant à calculer la charge de refroidissement d'une pièce due à l'éclairage. Le présent article présente ces données.

Calculating Cooling Load Caused by Lights

A method of computing cooling load from lights, based upon analysis of power input to lights, is presented, and experimental confirmation through use of a calorimetric facility is described. Design data are given in tabular form, and can be used in conjunction with the new transfer function method published in 1972 ASHRAE HANDBOOK OF FUNDAMENTALS.

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AN accurate estimate of the cooling load imposed by lights is essential in the design of air-conditioning systems since it is, in many cases, the major component of the room load. Calculation of this load component is not straightforward; the rate of heat gain to the air from lights can be quite different from the power supplied to them. Some of the energy emanating from lights is in the form of radiation and only affects the air after it has been absorbed by walls, floor and furniture and has warmed them to a temperature that is higher than the air temperature.

Lack of reliable data for these calculations led to an analytical study of the cooling load resulting from power input to lights.[1] This study indicated the need for an experimental check of the theoretical results and of experimental determination of some of the parameters used in these calculations. A calorimeter room was therefore built and instrumented for this purpose.[2] Subsequent results are of limited value for design calculations, however, because they do not cover a very broad range of room construction, ventilation apparatus, or installation arrangement. They have been used to derive a set of design data for calculation of room cooling load from lights. The design data and a sample calculation are now presented in the form required for the new transfer function method published in the 1972 ASHRAE HANBBOOK OF FUNDAMENTALS.

Results of the tests confirm the conclusions of the analytical study, that the relation between a step change of power input to lights and the corresponding cooling load can be described by a simple transfer function with two independent constants. Using this transfer function the cooling load at time t = n (for simplicity $n\Delta$ is denoted by n where Δ is time interval) is given by

$$q(n) = a_1 W(n-1) + a_2 W(n-2) + bq(n-1) \qquad (1)$$

where

$q(n)$ = cooling load from lights at $t = n\Delta$
$W(n)$ = power input to lights at $t = n\Delta$
a_1, a_2 and b = coefficients of transfer function, and

$$a_2 = 1 - b - a_1 \qquad (2)$$

Coefficients a_1 and b have different and distinct physical significance. The a_1 coefficient is the ratio of cooling load over power input to lights 1 hr* after the lights have been switched on, i.e. $q(t=\Delta)/W$. It depends mainly on the way in which the room ventilation system is arranged and the type of light fixtures used. The b coefficient is the ratio of cooling load at time t over load at time $(t-\Delta)$ when t is greater than 2Δ, i.e. $b = q(n)/q(n-1)$ for $n>2$. This coefficient is a measure of the rate of load decay after the lights have been switched off. The b value is a strong function of room thermal storage characteristics and is nearly independent of the type of room heat gain. Thus, the same b value is appropriate for the room transfer functions for solar heat gain, wall and roof transmission heat gain, as well as heat gain from lights.

The set of design values of a_1 given in Table I for a range of types of light fixture, ventilation rates, systems, and room furnishings, is derived from the experimental results simply by grouping the features of the room, light fixtures, and ventilation that are appropriate for a single value of a_1. The estimated precision of the a_1 value that can be obtained from Table I is about $\pm$ 0.05. If a particular case requires higher precision it must be determined by test on a full-scale prototype.

Design values of b (Table II) for different room constructions characterized by their mass per sq ft of floor area, S, and for different room air circulation intensities were computed using the procedure described in Refs 3 and 4. (In these papers the b value is called a ratio of successive terms.) This procedure takes into account heat storage of room envelope components, heat intercange between room air and room surfaces by convection, and heat interchange by radiation inside the room. In these calculations two parameters were varied: floor slab thickness, and inside surface convection coefficient. The first depends on heat storage capacity of the room; the second, on room air circulation intensity.

There was good agreement between calculated b values for S = 75 lb/sq ft of floor area and experimentally determined values (S for experimental room is approximately 75 lb/sq ft) indicating that this calculation procedure is adequate for determination of b values for typical room construction. The estimated precision of b value that can be

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$^*\Delta = 1.0$ hr is used since the Z-transfer functions, weighting factors or response factors currently used in air-conditioning calculations are based on a 1-hr time interval.

Table I. Design Values of "a_1" Coefficient

a_1	Furnishings	Air Supply & Return	Type of Light Fixture	Comment
	Features of Room Furnishings, Light Fixtures & Ventilation Arrangements			
0.45	Heavyweight, simple furnishings, no carpet	Low rate; supply & return below ceiling ($V \leq 0.5$)*	Recessed, not vented	
0.55	Ordinary furniture, no carpet. Increase a_1 by 0.05 when carpet is used	Medium to high ventilation rate; supply and return below ceiling or through ceiling grille and space ($V \geq 0.5$)	Recessed, not vented	
0.65	Ordinary furniture, with or without carpet	Medium to high ventilation rate or fan coil or induction type air conditioning terminal unit; supply through ceiling or wall diffuser; return around light fixtures and through ceiling space. ($V \geq 0.5$)	Vented	Effect on a_1 value by furnishings decreases when light fixtures are used as air supply and/or return registers
0.75 or greater	Any type of furniture	Ducted returns through light fixtures	Vented or free-hanging in air stream with ducted returns	a_1 value is equal to the fraction of light power input picked up by ventilation air at light fixtures or 0.75, whichever is greater

*V is room air supply rate in cfm/sq ft of floor area.

Table II. The "b" Values Calculated for Different Room Air Circulation Rates & Envelope Construction

Room Envelope Construction*	2-in. Wood Floor	3-in. Concrete Floor	6-in. Concrete Floor	8-in. Concrete Floor	12-in. Concrete Floor	Room Air** Circulation & Type of Supply and Return
Specific	10	40	75	120	160	
Mass lb/sq ft	0.88	0.92	0.95	0.97	0.98	Low
of floor	0.84	0.90	0.94	0.96	0.97	Medium
area	0.81	0.88	0.93	0.95	0.97	High
	0.77	0.85	0.92	0.95	0.97	Very High
	0.73	0.83	0.91	0.94	0.96	

* Floor covered with carpet and rubber pad; for a floor covered only with floor tile take next b value down the column.

**Low: Low ventilation rate—minimum required to cope with cooling load due to lights and occupants in interior zone. Supply through floor, wall or ceiling diffuser. Ceiling space not vented and $h = 0.4$ Btu/(hr)(sq ft)(F) (where h = inside surface convection coefficient used in calculation of b value).

Medium: Medium ventilation rate, supply through floor, wall or ceiling diffuser. Ceiling space not vented and $h = 0.6$ Btu/(hr)(sq ft)(F).

High: Room air circulation induced by primary air of induction unit or by fan coil unit. Return through ceiling space and $h = 0.8$ Btu/(hr)(sq ft)(F).

Very High: High room air circulation used to minimize temperature gradients in a room. Return through ceiling space and $h = 1.2$ Btu/(hr)(sq ft)(F).

selected from Table II for usual room construction is $\pm$ 0.01 for the higher values of b and $\pm$ 0.02 for the lower ones. Note that the addition of extra resistance between room air and floor slab, such as heavy carpeting, increases the b value. This was demonstrated by an experiment where 2 in. of insulation was placed between carpet and floor slab. Although the S value remained unchanged, the b value increased from 0.93 to 0.97. Thus, in special cases where the aptness of the value selected by Table II is in doubt and high precision is needed, the b value should be calculated by the procedure outlined in Refs 3 and 4 or determined by test on a full-scale prototype.

Eq (2) is based on the assumption that cooling load and power input to lights eventually become equal if the lights are long enough. This is a valid assumption for interior zones of a multi-story building; but for an exterior zone a fraction of the power input to lights is transferred to the outside and never contributes to cooling load. Transfer function coefficients must be modified, therefore, to allow for this heat loss: i.e., a_1 and a_2 must both be multiplied by a factor F, where F is the ratio of cooling load to the power input to lights at steady-state: $F = q(\infty)/W$.

For example, calculations of room thermal response factors by the procedure outlined in Refs 3 and4 indicate that for non-vented light fixtures in a typical multi-story building an empirical relation can be used

$$F = 1 - 0.02G \qquad (4)$$

for the range $0 < G < 15$, where G[Btu/(hr)(ft)(F)] is the conductance between room air and the exterior of the room per unit length of outside wall:

$$G^* = \frac{1}{L}\left\{\begin{array}{l} U_{window} \times A_{window} \\ + U_{exterior\ wall} \times A_{exterior\ wall} \\ + U_{corridor} \times A_{corridor} \end{array}\right.$$

where

L = length of exterior wall of room, ft
U = U value of room enclosure element, Btu/(sq ft)(F)
A = area of room enclosure element, sq ft

ROOM & CEILING SPACE LOADS

Cooling load caused by lights can be considered to be made up of two components when air return is through the ceiling space. One part of the load is the cooling needed to maintain constant temperature in the room; the other is the heat picked up by the ventilation air as it passes through the light fixtures or ceiling space or both. An accurate calculation of these loads requires separate transfer functions for each component. For ordinary air-conditioning cooling load calculations, however, a single transfer function can be used to calculate the total, and the components can be taken simply as constant fractions of this total. This was checked experimentally by measuring temperature of air entering the room, temperature of air as it passed through the ceiling, and temperature of air leaving the ceiling space. The ratio of the temperature changes gives the ratio of cooling load below and above the ceiling. These fractions were found to be nearly constant with time. For example, in a typical case the fraction absorbed in the ceiling space only decreased from 0.40 to 0.34 between the 2nd and 36th hour after lights were turned on.

**In cases where surrounding spaces are not air conditioned, the expression for G is augmented by additional summation terms:* $U_{separation\ between\ room\ and\ non\text{-}conditioned\ spaces} \times A_{separation}$.

Two arrangements of ventilation air return were tested at steady-state conditions for a range of air flow rates. In both cases, supply was through slots at the sides of the troffers and the return was through the center and around the tubes of the fixtures. The only difference between the two tests was that in one case space above the ceiling was used as a plenum; in the other, return air was ducted through the ceiling space and did not mix with air in the ceiling space.

A comparison of curves in Fig. 1 for return air through the ceiling space with and without ducts indicates that a significant portion of heat picked up by air (as high as 35% of power input to lights) was transferred into the room through ceiling and floor slabs when the ceiling space was used as a plenum for the return air.

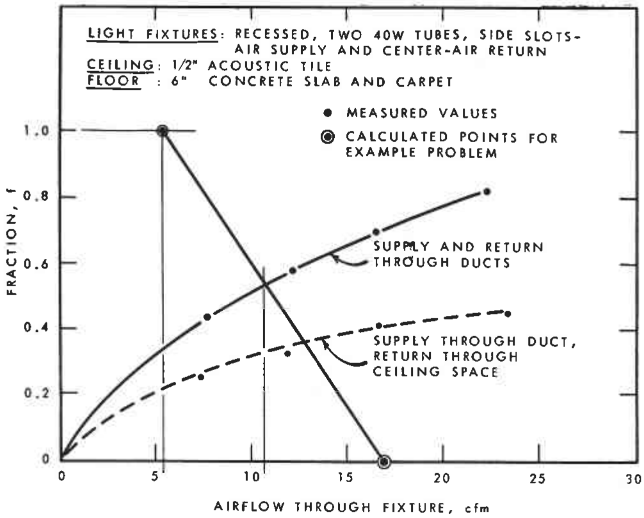

Fig. 1 Fraction of light power input that appears as cooling load in ceiling space, f, vs air flow through vented light fixtures

Where return air is ducted, data on heat pick-up and removal by ventilation air can be obtained by a relatively simple test: the heat pick-up by air flowing through a fixture is measured at various intake air temperatures and flow rates under steady-state conditions. These data are usually available from manufacturers of light fixtures. Where return air through the ceiling space is not ducted, an accurate set of data can be obtained only by testing a prototype; a rough estimate may be obtained using simple steady-state heat balance equations for ceiling space.

Example

1. Cooling Load. Calculate cooling load caused by light power input of 5 watts/sq ft of floor area. Lights are on from 7 a.m. to 6 p.m. Monday to Friday. The air-conditioning system is an all-air system, with air supplied through wall diffusers and returned through light fixtures and ducts in the ceiling space. The floor is 6-in. concrete slab covered with tile (S = 75 lb/sq ft) and the ceiling is acoustic tile. This space is an interior zone of a multi-story building and is provided with ordinary furniture.

The first step is to select the appropriate transfer function coefficients:

From Table I, $a_1 = 0.75$ is selected because light fixture and ventilation arrangement of the room is similar to the fixtures of the room given for $a_1 = 0.75$.

From Table II, b = 0.93 is selected because this value is appropriate for S = 75 lb/sq ft, i.e., non-vented ceiling space and floor covered with tile. The a_2 value is given by Eq (2), i.e.

$$a_2 = 1 - a_1 - b$$
$$= 1 - 0.75 - 0.93 = -0.68$$

The F factor is taken as 1.0 since the space under consideration is an interior zone of a building. Substitution of a_1, b and a_2 values in Eq (3) gives the following expression for cooling load caused by lights:

$$q(n) = 0.75\ W(n-1) - 0.68\ W(n-2) + 0.93\ q(n-1)$$

cooling load caused by light operation before the weekend will be essentially zero by the following Monday morning (assuming that the lights were off during the weekend). Calculations can be started, therefore, for Monday by letting q(n) = 0.0, W(n) = 0.0, W(n−1) = 0.0 and W(n−2) = 0.0 at n = 6; for time between 7 a.m. and 6 p.m. W(n) = 5.0 w/sq ft.

Thus for Monday

q(7)	=	0.75(0.0) − 0.68(0.0) + 0.93(0.0)	= 0.0 w/sq ft
q(8)	=	0.75(5) − 0.68(0.0) + 0.93(0.0)	= 3.75 w/sq ft
q(9)	=	0.75(5) − 0.68(5) + 0.93(3.75)	= 3.84 w/sq ft
q(10)	=		3.92 w/sq ft
q(11)	=		3.99 w/sq ft
q(12)	=		4.07 w/sq ft
q(13)	=		4.13 w/sq ft
q(14)	=		4.19 w/sq ft
q(15)	=		4.25 w/sq ft
q(16)	=		4.30 w/sq ft
q(17)	=		4.35 w/sq ft
q(18)	=	0.75(5) − 0.68(5) + 0.93(4.35)	= 4.40 w/sq ft
q(19)	=	0.75(0.0) − 0.68(5) + 0.93(4.40)	= 0.69 w/sq ft
q(20)	=	0.75(0.0) − 0.68(0.0) + 0.93(0.69)	= 0.64 w/sq ft

and so on. Loads for corresponding times for subsequent days are slightly higher, the maximum being q(18) = 4.5 w/sq ft.

2. Air Supply Rate & Return Air Temperature. For the space described in the previous sample calculation, determine return air temperature and air supply rate needed to remove heat generated by lights, plus an additional 2 watts/sq ft of floor area generated by occupants when supply air temperature is 55 F, room air temperature is maintained at 75 F, and there is one light fixture per 17 sq ft of floor area. Assume that the curve for ducted return in Fig. 1 is applicable for light fixtures under consideration.

The heat pick-up by the air is

$$q = 1.1\ Q(\Delta\theta)\ \text{Btuh}$$

where

Q = air supply rate, cfm
1.1 = constant, Btu/(hr)(cfm)(F)
$\Delta\theta$ = temperature rise in F deg

The airflow rate needed to remove heat generated by people (i.e. 2w = 6.8 Btuh) is

$$Q = \frac{q}{1.1\Delta\theta} = \frac{6.8}{1.1(75\text{-}55)} = 0.31\ \text{cfm/sq ft}$$
$$= 5.3\ \text{cfm/fixture}$$

and air flow rate needed to remove maximum heat load generated by lights (i.e., 4.5 w, see first part of this example) plus heat generated by occupants is

$$Q = \frac{(2+4.5)\ 3.41}{1.1(75\text{-}55)} = 1.0\ \text{cfm/sq ft}$$
$$= 17.0\ \text{cfm/fixture}$$

These air flow values can be plotted in Fig. 1 by noting that Q = 5.3 cfm/fixture and Q = 17.0 cfm/fixture are for f = 1.0 and f = 0.0, respectively, where f is that fraction of power input to lights which appears as cooling load in the ceiling space. The intersection of the straight line drawn between these calculated points and the curve for ducted return air given in Fig. 1 is the required air supply rate for this case: 10.7 cfm/fixture or 10.7/17 = 0.63 cfm/sq ft of floor area.

Return air temperature is given by

$$\theta_{return} = \theta_{supply} + \frac{q}{1.1 \times Q}$$

Thus, for q = (2+4.5)3.41 = 22.2 Btu/sq ft, θ_{supply} = 55 F and Q = 0.63 cfm/sq ft

$$\theta_{return} = 55 + \frac{22.2}{1.1 \times 0.63} = 87\ \text{F}$$

SUMMARY

The expression that relates power input to lights and cooling load for a particular room can be obtained by selecting the a_1 and b coefficients using data in Tables I and II as a guide.

The value of a_1 may be as low as 0.45 or as high as 0.90, depending on type of fixtures used, air supply system arrangement, and air supply rate and room furnishings. It is reasonable to expect that this selection procedure may give an a_1 value within about ± 0.05, and that if this degree of accuracy is not acceptable tests will be needed to determine a_1.

The b coefficient ranges from 0.8 for lightweight construction to 0.98 for very heavy construction. It depends mainly on the type of room envelope construction and on air circulation in the room. It can be selected from Table II with an estimated accuracy of ± 0.01 for the higher values and ± 0.02 for the lower ones.

The a_2 coefficient is dependent on a_1 and b, so that its precision is governed by the precision of a_1 and b. Data on light fixtures, such as heat pick-up by air when fixtures are used as supply and/or return registers vs air flow rate, should be obtained from the manufacturer of the fixtures. A careful check must be made to ensure that conditions for which the manufacturer's data apply (i.e. air flow rate and type of installation) correspond to conditions obtaining in the room under consideration.

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