Improved Fluorescent Lighting Models for Building Energy Programs

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Abstract

Currently, building energy analysis programs employ models of fluorescent lighting systems which are much oversimplified and potentially inaccurate. One important factor neglected by all whole-building programs is the variation of lamp power and light output with lamp wall temperature. This means that the lighting component of zone load and artificial lighting levels are both calculated incorrectly. Additionally, the latter implies that the predicted energy saving for systems that automatically reduce artificial lighting when daylight is available is also incorrect. Recent experimental and analytical work provide a basis for correcting this deficiency. This paper reviews the recent work and proposes a new lighting heat gain model that can be adapted to either weighting factor or heat balance building energy analysis programs.

Introduction

Artificial lighting represents a significant portion of energy consumption and demand in modern buildings, both directly and through its contribution to cooling load. This would suggest careful attention to the models used for lighting induced cooling loads in building energy analysis computer programs. In fact, however, the lighting systems models currently in use in these programs are very simple, representing a state of knowledge more then 20 years old. For example, in both weighting factor based programs (LBL 1984) and heat balance programs (Hittle 1979) lighting cooling loads are customarily modeled in terms of user-supplied "upward fraction" and "percent radiant," based on an idea suggested by Kimura an Stephenson in 1968 (Kimura and Stephenson 1968).

In addition to the above shortcomings in cooling load calculations, current programs also fail to properly calculate lighting input power, and light output. It is well known that both the input power and light output vary strongly with minimum lamp wall temperature, Figure 1 (IES 1984). Yet all current whole-building simulators omit calculation of lamp temperature, instead assuming that the input power and lumen output remain constant at the "rated" condition at the peak light output point. Depending upon the number of lamps per luminaire, the air flow circuit for the luminaire and plenum and other factors, the lamp may actually operate 20°F or more above or below the rated temperature. From Figure 1 we see that the actual input power can therefore be as much as 20% lower than predicted if the temperature dependence is ignored. This means that the lighting heat gain and resulting cooling load, are being calculated incorrectly. Additionally, the calculated artificial lighting levels are off, so that the predicted energy saving for automatically controlled daylighting systems is also incorrect.

A great deal of work on lighting loads has taken place since the currently used models were formulated. Experimental work at the National Institute of Standards and Technology (NIST), sponsored by the Electric Power Research Institute and the U. S. Department of Energy, has provided new thermal performance data (Treado 1991). This work used a full-scale test cell that

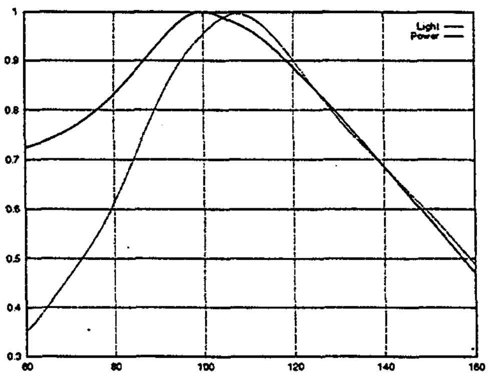

Figure 1. Lamp temperature dependence.

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could be re configured to represent a wide range of typical commercial and institutional building designs. Design variables investigated included luminaire type, air path, and floor mass and covering. Measurement allowed calculation of new sets of lighting weighting factors (Treado 1990), and work under way at NIST attempts to extract new upward/downward fractions and radiant splits. As will be seen later in this paper, the detailed temperature measurements from these tests may also allow parameter estimation for more detailed models. In addition to the NIST work, there has been a great deal of experimental effort on luminaire and lamp performance at the Lawrence Berkeley Laboratory. This work has produced, among other things, lamp performance data such as exemplified in Figure 1 for conventional and advanced luminaires and ballasts (Rubinstein, Verderber, and Siminovitch 1984).

In addition to the experimental work, more complete theoretical lighting models have been developed and incorporated in computer programs (Sowell 1972, Sowell 1989, Walton 1990). These programs allow description of room and plenum geometry, air flow paths, film coefficients, and radiative properties, so that all energy flow paths between the lamp and the exhaust air can be represented in a detailed thermal model. Also, the nonlinear lamp performance is accounted for. Consequently, these programs are able to calculate the cooling loads directly from first principles and measurable properties, rather than relying on empirical fractions and splits. However, along with the greater detail comes longer computation times, making these programs useful mainly for research; in most instances they will be too costly to use for year-long analyses of entire buildings.

The work reported here attempts to bridge the gap between the crude lighting models currently employed in whole-building simulators and the detailed lighting codes, using use the results of recent experimental work to establish needed data. The focus is on a single aspect, namely defining a *lighting heat gain model* that will predict lighting input power and light output taking account of their dependence on minimum lamp wall temperature. This model can be incorporated in both weighting factor and heat balance programs. Integration of the new model in the whole-building simulators is also discussed.

Model Development

Basis

Because lighting input power and light output depend upon minimum lamp wall temperature, the basis of the calculation must be a thermal model of the luminaire, including its interactions with the room and above-ceiling plenum. The model must account for two different schemes commonly used for lighting level control, namely discrete switching of multiple lamp circuits, and continuous dimming. To account for discrete level control in the zone, the luminaire model is assumed to have two, separately switched lamp circuits. The lamps in each circuit are either full on or full off. All lamps in a particular circuit are assumed to be at the same average temperature and the same light output. This leads to the thermal model shown in Figure 2.

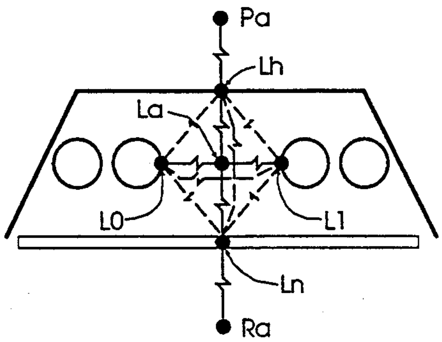

Figure 2. Nodal model of luminaire.

Note that each of the lamp nodes represents zero or more lamps. Systems with continuous dimming can then be modeled with zero power to one of the lamp nodes, and continuously variable power to the single active node in the model. Also, observe that the luminaire model can represent all similarly controlled luminaires in the zone. In summary, the general model is specialized by lumping together all lamps with similar control and thermal conditions. Lumping together means adding surface areas, lumen output, and power.

The intuition guiding development of the mathematical model is that given boundary temperatures at the plenum and room air nodes and power dissipation at the various nodes, one ought to be able to calculate temperatures at all non-boundary nodes. The basic equations to allow for this calculation are the steady state energy balances at each non-boundary node. The nonlinear relationships between lamp power, light, and temperature (Figure 1) introduce additional equations that must also be satisfied. Also, we observe that input power manifests itself as dissipation at the lamp, and at the luminaire housing where the ballast is mounted. Additionally, the light emitted at the lamp causes two thermal effects. First, it represents energy that leaves the lamp node by a mechanism that is not temperature dependent; this must be subtracted from the dissipation. Second, it is partially absorbed at the housing and lens, representing dissipation at these surfaces. The thermal model must therefore be supported by a luminous radiant transfer model that calculates net short-wave absorption at each solid node. With these ideas in mind, the math model is easily defined.

We assume that the room and plenum temperatures are available, perhaps from a previous time step. and that the lighting signal $r(t)$ is provided for each lamp circuit. This signal is viewed as the fraction of "full on" requested by the lighting controls. For switched circuits it is 0 or 1, while for dimmed circuits it can be any value between these limits. Given these inputs, the math model can be solved to give instantaneous values for lighting heat gain and lighting level in the zone. Although probably not needed external to this model, the temperatures of the various nodes will also be determined.

Lamp Power and light output

Equations 1 through 4 relate to lamp and ballast input power. Equation 1 is the relationship between lamp power in circuits *0* and *1* and the minimum lamp wall temperature, Figure 1. $P_{nom,i}$ is the total power for all lamps in each circuit. Total lamp power for both circuits is given by Equation 2. Ballast power is assumed to be k_b times the total lamp power, Equation 3. The total zone lighting power, P_{tot}, is given by Equation 4. Equations 5 and 6 give corresponding instantaneous lighting output, ϕ_l, using the function $f_\phi(T_{min})$ to represent the normalized light output curve, Figure 1.

$$P_{l,i} = P_{nom,i} r_i(t) f_p(T_{min,i}); \qquad i = 0,1 \qquad (1)$$

$$P_l = P_{l,0} + P_{l,1} \qquad (2)$$

$$P_b = k_b P_l \qquad (3)$$

$$P_{tot} = P_l + P_b \qquad (4)$$

$$\phi_{l,i} = \phi_{nom,i} r_i(t) f_\phi\left(T_{min,i}\right); \quad i = 0,1 \qquad (5)$$

$$\phi_l = \phi_{l,0} + \phi_{l,1} \qquad (6)$$

Lamp heat balance

Equation 7 is the heat balance on the lamp wall. The left side is the input lamp power less that carried away by short-wave radiation, $K\phi_{l,i}$, where K is the heat equivalent of the light. The average temperature for circuit i, $\overline{T}_{l,i}$, is used in the heat balance because the model distinguishes between average and minimum temperature of the lamp wall. In these and other heat balance equations, $(h_c A)_{a,b}$ represents the product of convective film coefficient and the heat transfer surface area between nodes a and b. Similarly $(h_r A)_{a,b}$ represents the product of the long-wave radiative coefficient and the heat transfer surface area between nodes a and b. Observe that we account for long-wave radiative transfer between the each lamp set and the lens and housing, and between the two lamp sets.

$$\begin{aligned} P_{l,i} - K\phi_{l,i} = {} & (h_c A)_{l,i}\left(\overline{T}_{l,i} - T_{la}\right) \\ & + (h_r A)_{l,i,lh}\left(\overline{T}_{l,i} - T_{lh}\right) \\ & + (h_r A)_{l,i,j}\left(\overline{T}_{l,i} - T_{l,j}\right) \\ & + (h_r A)_{l,i,lns}\left(\overline{T}_{l,i} - T_{lns}\right); \qquad i = 0,1 \end{aligned} \qquad (7)$$

Luminaire air heat balance

The luminaire air node *la* is coupled consecutively to the lens, *Lns*, lamps, *l,1* and *l,2* , and housing *lh*. Additionally, it is coupled to the room air node by bulk convection, represented by $\dot{m}C_p$, where $\dot{m}$ is the mass flow rate. This would be small for an unventilated luminaire. This leads to the heat balance shown in equation 8.

$$\begin{aligned} 0 = {} & (h_c A)_{lhi}\left(T_{la} - T_{lh}\right) + (h_c A)_{lni}\left(T_{la} - T_{lns}\right) \\ & + (h_c A)_{l,0}\left(T_{la} - \overline{T}_{l,0}\right) + (h_c A)_{l,1}\left(T_{la} - \overline{T}_{l,1}\right) \\ & + \dot{m}C_p\left(T_{la} - T_{ra}\right) \end{aligned} \qquad (8)$$

Lens and Housing Heat Balances

Short wave radiative transfer is somewhat simplified in the model. It is assumed that the lamp emits according to Equation 5, and the fraction *CU* of this is delivered to the room. The remainder is absorbed either by the lens or the luminaire housing. Lens short wave absorption is assumed to be α_l of the amount entering the room, leaving $(1-(1+\alpha_l)CU)$ for absorption by the luminaire housing. With this idea, the heat balances on the lens and housing are as given in Equations 9 and 10. Observe that the lens couples convectively with the room air, and the housing with the plenum air, while both are coupled with the luminaire air.

$$\begin{aligned} \alpha_l CU K\phi_l = {} & (h_c A)_{lno}\left(T_{lns} - T_r\right) \\ & + (h_c A)_{lni}\left(T_{lns} - T_{la}\right) \\ & + (h_r A)_{0,lns}\left(T_{lns} - \overline{T}_{l,0}\right) \\ & + (h_r A)_{1,lns}\left(T_{lns} - \overline{T}_{l,1}\right) \\ & + (h_r A)_{lns,lh}\left(T_{lns} - T_{lh}\right) \end{aligned} \qquad (9)$$

$$\begin{aligned} \left(1-(1+\alpha_l)CU\right)K\phi_l + P_b = {} & (h_c A)_{lhi}\left(T_{lh} - T_{la}\right) \\ & + (h_c A)_{lho}\left(T_{lh} - T_{pa}\right) \\ & + (h_r A)_{0,lh}\left(T_{lh} - \overline{T}_{l,0}\right) \\ & + (h_r A)_{1,lh}\left(T_{lh} - \overline{T}_{l,1}\right) \\ & + (h_r A)_{lns,lh}\left(T_{lh} - T_{lns}\right) \end{aligned} \qquad (10)$$

Minimum temperature depression

Equation 11 says that the minimum lamp wall temperature is an empirically determined amount d below the average temperature of the lamp wall.

$$T_{min,i} = \overline{T}_{l,i} - d; \quad i = 0,1 \qquad (11)$$

Long Wave Radiant Interchange

The radiant conductances in the heat balance equations, e., g., $(h_r A)_{1,lns}$ can be estimated using diffuse radiant transfer methods. Including interreflections, this can be expressed by the vector-matrix equation:

$$h_r A = A[[I - RF] - I][I - RF]^{-1} E \qquad (12)$$

In this equation all symbols are *4x4* matrices, with I being the identity matrix. A is diagonal and represent the radiating surface areas for *l0, l1, lh,* and *lns*. R is also diagonal, representing the long wave reflectances of these same surfaces. F contains the view factors between all surface pairs. E is diagonal with nonzero elements $\varepsilon_i \sigma T_{avg}^3$ where ε_i and σ are the emissivity and Stephan-Boltzmann constants respectively, and T_{avg} is an average temperature for the participating surfaces. The elements of the matrix $h_r A$ are the needed radiant conductances in the model.

Data Requirements

The model data requirements are evident from the above equations. Some of these data are design parameters of the room, e.g., nominal lighting power and the various areas. Much of the thermal data is readily available from property handbooks, or can be calculated. However, some data items, notably convective heat transfer coefficients, remain to be determined empirically.

Radiative heat transfer data for the model is readily available. The short-wave performance is characterized in the model by the Coefficient of Utilization, *CU*, provided by the luminaire manufacturer, and lens absorptivity which is well known for common lens materials. Also, long wave surface emissivity is known to be close to 0.9 for all materials commonly used in luminaires. The geometric data, i.e., radiant interchange view factors, can be calculated by well known methods (Sowell 1972, Walton 1986).

Convective heat transfer coefficients must be determined experimentally or from empirical relationships. Unfortunately, the full scale lighting test cell experiments (Mitalas and Kimura 1971, Treado 1991) have been aimed at overall measurements and have not directly produced detailed data such as required here. The work of Spitler (Spitler, Pedersen, and Fisher 1991) did measure heat transfer coefficients for room surfaces, but not for luminaire or lamp surfaces.

Ultimately, deployment of the model proposed here will require experiments aimed specifically at measurement of convective film coefficients at luminaire and lamp surfaces. Until this work can be carried out, there are two possible avenues to obtain the needed data. The obvious method is to employ the general convective correlations such as summarized in the ASHRAE Handbook (ASHRAE 1989). However, these cannot be expected to accurately represent the conditions within and around luminaires. For example, free convection correlations for horizontal cylinders are generally based on cylinders in a free field, whereas the convection patterns around lamps within a sealed luminaire are likely to be strongly affected by the luminaire housing, lens, and other lamps in close proximity. Also, when the luminaires are ventilated with room exhaust air, the complex flow patterns are unlikely to be well represented by the available forced convection correlations.

Another approach is to attempt to extract the model parameters from the NIST data using parameter identification techniques (Grewal and Andrews 1993). With this technique the model is used to predict key system variables, e.g., lamp wall temperatures, under a range of conditions, treating the coefficients as parameters. The predictions are compared to the experimental values, forming an error measure. Regression is then used to calculate the parameters so as to minimize the error measure.

The success of the parameter identification approach is not certain at the time of this writing. If the model is inadequate, e.g., some important phenomenon has been neglected, or the measured values embed significant effects of uncontrolled factors, the error measure will remain large providing little confidence in the calculated parameters. Another problem that sometimes arises in identification is that the experiment does not adequately cover the range of possible system excitations. When this happens, it is impossible to distinguish the effects of each model parameter independently so the regression fails. However, there may be reason for optimism. First, the model is based on detailed models which have been shown to agree reasonably well with overall test cell measurements (Sowell 1990). Second, the needed measurements are lamp, luminaire air, luminaire housing, and lens temperature elevations above room temperature. Since these elevations tend to be large relative to thermocouple error, these particular measurements may be among the most reliable in the experiment. In any event, it appears that this is the most hopeful approach until more direct measurements can be obtained.

Another need for experimental data is evident in the lamp performance functions, $f_p(T_{min,i})$ and $f_\phi(T_{min,i})$. Although the curves from the IES Handbook (IES 1984), Figure 1, are often used generically for lamp performance, it is well known that such data must be empirically determined for particular lamps. Actual performance curves for some lamps have been established in the lighting laboratory at LBL (Rubinstein, Verderber, and Siminovitch 1984), and in the NIST test cell (Treado 1991). More data of this kind will be needed to support the proposed model for a wide range of luminaires.

Finally, appropriate values of d, the temperature depression below the average at the coldest point on the lamps, needs to be determined empirically. It is to be expected that d will depend upon the luminaire configuration and upon the air flow path for ventilated types. Some data was obtained at NIST, but more will be required to cover all the cases.

Solution Algorithm

Due to the power and light output variation with lamp wall temperature, the model is nonlinear and requires an iterative solution. A simple algorithm based on successive substitution converges too slowly for practical use, but the Newton-Raphson method works quite well. Space does not permit a detailed presentation of such an algorithm, but we can explain it in broad terms.

Two iteration variables, $T_{min,i}$, are assigned estimated values. This allows evaluation of Equations (1-6) sequentially. The heat balances, Equations (7-10), then form a set of simultaneous linear equations which can be expressed as $UT = S$. Here U is a $5x5$ matrix representing convective and long-wave radiative transfer, T is the vector of luminaire node temperatures (See Figure 2), and S is the vector of heat loads at these nodes. Assuming constant long-wave properties, film coefficients and air mass flow through the luminaire, the U matrix is constant and can be calculated and inverted once before the iteration loop. However, S includes power dissipated at the node, e.g., lamp and ballast input power, and net short-wave absorption, all of which vary with lamp temperature, so it must be updated throughout the iteration. Once S is determined, T can be found by solving this linear set, and then $T_{min,i}$ can be calculated from Equation (11). The difference between the assumed and calculated $T_{min,i}$ forms a vector-valued function (with two elements) that becomes the "user function" in a standard Newton-Raphson procedure (Press et al. 1988).

With very little more work, the Jacobian matrix for the Newton process can be calculated at the same time the functions are being calculated. In this regard, note that all of the equations can be easily differentiated if the derivatives of the lamp performance functions are known. By representing $f_p(T_{min,i})$ and $f_\phi(T_{min,i})$ as spline fits, these derivatives are readily available.

Integration in Energy Programs

Weighting Factor Programs

In the current DOE-2 program, lighting heat gain is assumed to be independent of lamp wall temperature and the lighting cooling load can be calculated once and for all in the LOADS step of the program. With the model proposed above, the lighting heat gain is affected by the room and plenum temperatures and these temperatures vary in the subsequent SYSTEM simulation step. Consequently, this approach will no longer be suitable. This raises the question of where in the calculation sequence the lighting heat gains and loads should be calculated.

To gain insight on this question, observe that there is a somewhat parallel situation in DOE-2 with regard to the effect of zone air temperature variation on storage load. In the LOADS step, DOE-2 calculates all zone loads with the assumption of constant temperatures in the room and above-ceiling plenum. The loads thus calculated do not account for heat storage and release as zone and plenum temperatures vary in response to secondary system control actions. To account for these effects, the constant temperature loads are corrected in the System Simulation step by use of the zone air temperature weighting factors. This suggests the following strategy for calculation of lighting loads taking account of lamp wall temperature effects.

In order to provide a lighting load estimate in the LOADS step, the proposed lighting heat gain model can be solved to give lighting heat gains at the lamp surface, i.e., P_{tot}, employing *design values* for room and plenum temperatures. These heat gains are then used along with lighting weighting factors to yield *design* lighting loads on the room and plenum. The sequence of lighting heat gains at each hour must be saved, as well as the sequence of cooling loads.

Actual lighting loads are subsequently calculated during System Simulation by a four step process. First, the lighting heat gain model is solved using the previous hour plenum and room air temperatures. Second, the lighting heat gain for the current hour as calculated in the LOADS step is subtracted from the same quantity calculated in the SYSTEMS step. This generates a time series of differential lighting heat gains, *reckoned at the lamp surface*. As the third step, this series is used as input to the lighting weighting factors, yielding *differential* lighting cooling loads. Finally, these differential lighting cooling loads are added to the saved sequence of design cooling loads, giving the actual lighting loads.

There is also the question of which weighting factors to use. One approach is to continue using those determined by the current DOE-2 custom weighting factor procedure. If this is done the proposed heat gain model can be modified to calculate the heat transfer to the plenum and room in addition to the total input power so that these spaces can be separately modeled if desired.

A better approach would be to revise the custom weighting factor calculation to more accurately represent the details of heat transfer in the luminaire, room and plenum. However, there is a theoretical problem to be resolved, because the transfer function method itself is based on *superposition*, which is mathematically limited to linear problems, while the dependence of lighting power on lamp wall temperature is highly nonlinear. Some theoretical work has been done on transform methods for nonlinear systems, primarily in the controls literature. Techniques such as z-forms and describing functions allow this extension (Gibson 1963). However, these methods are not applicable here. The describing function technique, for example, is aimed at prediction of amplitude and phase shifts for sinusoidal signals of small amplitude, and consequently is suitable for stability and analysis and of control systems. In building simulators such as DOE-2, the need is to predict performance in the time domain and over wide ranges of system operation.

A more promising approach is to partition the thermal model into a *linear, algebraic* portion and a *nonlinear, dynamic* portion. The nonlinear portion can then be solved algebraically to give "forcing functions" for the linear dynamic portion, which is modeled with weighting factors. To apply this approach to the lighting model the dividing line between linear and nonlinear portions must be *the lamp surface*. That is, the processes whereby energy leaves the lamp surface and is absorbed at other surfaces and air streams are, to a good approximation, linear. The nonlinear relationship between lamp wall temperature and input power is entirely within the lamp. This means that the lighting weighting factors must be calculated for a pulse of heat gain appearing at the lamp surface. The thermal model used for calculating these weighting factors need not include the lamp nonlinearity; it is a straightforward thermal model of the plenum, luminaire and room. The input is a pulse at the lamp surface, and the output is the time sequence of cooling loads at the room and plenum air nodes. The forcing function, i.e., instantaneous lamp input power, is calculated during the simulation step based on room and plenum air temperatures and lamp nonlinearity.

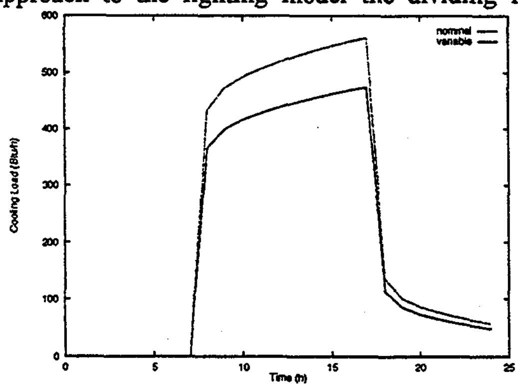

Figure 3. Lighting cooling loads.

These ideas can be implemented with straightforward modifications to the current DOE-2 code. One approach under consideration is to use either the HLITE (Walton 1990) or the LIGHTS (Sowell 1989) model more or less directly in place of the current DOE-2 custom weighting factor routines. Instead of the customary radiant fractions for lighting energy, the user will be asked to provide geometric and surface property data describing the room and plenum. The detailed model will then be used to determine cooling load response to an input pulse of heat gain at the lamp surface, and at the ballast. Resulting cooling loads will be calculated at both the plenum and room air nodes. From such responses, weighting factor sets for the plenum and room can be determined.

Heat Balance Programs

The lighting heat gain model can also be used in heat balance programs such as BLAST. In this case the room and plenum temperatures are calculated in the LOADS step, so the lighting heat gain has to be calculated only once and in this step. The only question is whether to use previous or current hour temperatures in this calculation. The latter will yield better accuracy, especially at times near light switching. However, additional iteration will be required.

Results

At this point the proposed lighting heat gain model has been implemented, but no integration with an energy program has been attempted. Consequently, the only results available are predicted lighting heat gains. These heat gains can be used in conjunction with available lighting weighting factors to yield approximate cooling loads. It is instructive to compare these predicted loads with those calculated using nominal input lighting power as the heat gain. Such a comparison is made in Figure 3.

The case shown here is for a standard 2 ft x 4 ft recessed fluorescent luminaire with 4 40W lamps operating with 220 lb./h ventilation air. Handbook correlations were used for film coefficients and radiative properties. Steady state lamp wall temperature, not shown, are predicted to be about 120°F. We see that loads are about 15% lower when lamp wall temperature effects are accounted for. More dramatic effects are predicted for unventilated 4-lamp luminaires. However, as pointed out by Treado (Treado 1991), 2 lamp luminaires run cooler so that ventilation is unnecessary, and in fact counterproductive since the lamps then operate below the peak light output point.

Conclusions

The lighting models presently used in whole-building energy analysis programs are oversimplified, resulting in inaccurate predictions of lighting levels, cooling loads, energy usage, and peak demand. Presented here is a new model that adjusts input power and light output accounting for sensitivities to lamp wall temperature. This model will improve accuracy of predicted cooling loads due to lighting, and will allow better assessment of daylighting control strategies. Integration with both weighting factor and heat balance programs is discussed.

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