

A MATHEMATICAL MODEL FOR CALCULATING
TRANSIENT HEATING OR COOLING
LOADS FROM LIGHTING

by

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NOMENCLATURE

Symbol	Significance	Units
A, B, a_1, a_2, b_1	Constants	-----
A	Area	ft ²
C	Thermal capacitance	Btu/°R
c_p	Specific heat capacity	Btu/lbm-°R
d	Fluorescent tube diameter	ft
d	Differential	-----
e	Base e constant=2.71828...	-----
F_{jk}	Radiation exchange view factor	-----
$\mathcal{F}_{jk}$	Script-F radiation exchange factor	-----
H	Total convection coefficient	Btu/hr-°R
h	Convection coefficient	Btu/ft ² -hr-°R
J	Radiosity: total radiation leaving a unit surface area	Btu/ft ² -hr
K	Total thermal conductance	Btu/hr-°R
k	Thermal conductivity	Btu/ft-hr-°R
L	Characteristic length parameter	ft
N	Number of full layers per slab	-----
n	Number of surfaces per enclosure	-----
P	Power in relation to lighting	Btu/hr
P_T	Total lighting energy in a period	Btu
q	Heat	Btu
q_R	Heat from lighting to room air	Btu
q_T	Total heat from lighting to both room and return air	Btu
S1, S2, S3	Constants	-----

Symbol	Significance	Units
$\overline{S_j S_k}$	Total radiation exchange factor	-----
T	Temperature	$^{\circ}\text{R}$
t	A point in time	hrs
$\overline{T}$	Mean effective temperature	$^{\circ}\text{R}$
T^*	Temperature of entering air	$^{\circ}\text{R}$
T^+	The new temperature at the end of a time period	$^{\circ}\text{R}$
U	Thermal conductance	$\text{Btu/ft}^2\text{-hr-}^{\circ}\text{R}$
V	Air velocity	ft/hr
W	Thickness of a slab	ft
Z	Constant	-----
GREEK		
α	Absorptance	-----
Δt	General time period	hrs
$\Delta t'$	Lightweight surface time period	hrs
ϵ	Emittance	-----
ρ	Reflectance	-----
ρ	Density	lbm/ft^3
Σ	Summation	-----
σ	Stephan-Boltzman constant	-----
τ	Transmittance	-----
DIMENSIONLESS NUMBERS		
Gr	Grashoff number	-----
Nu	Nusselt number	-----
Pr	Prandtl number	-----
Re	Reynolds number	-----

Symbol	Significance	Units
SUBSCRIPTS		
a	Refers to an air mass	-----
B	Refers to the luminaire ballast	-----
D	Refers to duct air	-----
d	Refers to the fluorescent tube diameter	-----
g	Refers to window glass	-----
i	Refers to inside	-----
j,k	General surface subscripts	-----
jk	Two subscripts signify "from j to k"	-----
o	Refers to outside	-----
R	Refers to room air	-----
s	Refers to short wavelength radiation	-----
t	Refers to the fluorescent tube	-----

CHAPTER I

INTRODUCTION

Because of the energy crisis, many areas of building design must be re-examined. One such area concerns the heat transferred to a building from electric lighting fixtures (luminaires). Lighting is the source of the majority of the air conditioning load in the interior of many office buildings and is of major concern in almost all buildings. Therefore, a more accurate analysis of the heat transferred from lighting would improve the estimation of air conditioning loads.

All of the energy consumed by lighting, P_T , is converted to heat, but not all of this energy immediately raises the temperature of the room or return air. Some of the energy does quickly enter the room or return air by convection from luminaire surfaces. But a portion may be first transmitted to plenum air or surfaces above a suspended ceiling. The radiant part of this energy must first be absorbed by a surface, heat the surface, and then be convected to room or return air. As a surface is heated, energy is stored according to the thermal properties of the surface and then released as the body cools. This results in a time delay between the release of energy from the luminaire and its appearance as heat to the room air, q_R , or heat to the total circulating air, q_T . The effect of this time delay varies with room construction, lighting and ventilation design, furnishings, and other conditions.

In 1978, the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) issued a request for proposals for a more thorough analysis of the heat transfer from luminaires. Part of the request reads as follows:

Data on heat release from lighting fixtures operating within a building or structure has been available only in recent years and then only for a partial sample of available lighting fixtures. In addition, the available data is not directly suitable in its application for use in building load calculations by either manual or computer methods. The data to be developed in this project is vital in order to permit effective input into building load calculations of the heat released by lighting fixtures.

In the Spring of 1979, ASHRAE awarded this project, RP-231, to the Mechanical Engineering department of KSU with Dr. H. D. Ball as principal Investigator. The project requires the completion of several tasks:

1. a) A literature search for works on heat release from lighting.
b) A survey of lighting manufacturers for heat release data and determination of the frequency of installation of different lighting fixtures.
2. The development of a mathematical model to predict heat release from all types of fluorescent fixtures, both to the room and to the plenum, and comparison of its results with actual tests.
3. Establishment of a plan for parametric study of many types of lighting and fixtures. The plan must be approved by ASHRAE.
4. Carry out this plan using the mathematical model.
5. Development of easily used tabular or graphical conden-

sations of the results.

OBJECTIVE

The objective of the present study is to accomplish the second task. This thesis describes the development of a mathematical model which simulates the transient and steady state heat transmission from fluorescent luminaires. The literature search section of the first task was accomplished as part of the background and analysis for this study and the results of the lighting survey are presented in Appendix A.

CHAPTER II

LITERATURE REVIEW

One nice thing about this project is the steady state results are automatically correct if the transient analysis is accurate. It is unfortunate that the reverse is not also true, for nearly all previously published works concerning heat transfer from lighting deal with steady state analysis or testing.

Before 1955, there was little work done in the area of heat transfer from luminaires. The most extensive article was by Sturrock (69) in 1938. He discusses the heating aspects of incandescent lamps and describes the testing of several methods of removing heat before it enters the room space. He also shows the results of transient heating tests on room spaces with different lighting levels. Other papers located that concern lighting heat transfer and predate 1955 have little results from experiments or analysis (17,18,40,41,43,60). Instead, they are descriptions of ideas to obtain more lumens per watt or ducting air through lighting fixtures.

Changes came with the introduction of suspended ceilings, recessed fluorescent lamps, and vented lighting fixtures. Between 1955 and 1965, many authors published articles describing the benefits of these advances (4,7,22,59,61). They discuss many varieties of fluorescent luminaires, both vented and static, and go into theories and estimates of the reduction in room cooling loads possible using these fixtures. Spurred on by

these possibilities, several groups began testing the heat transfer characteristics of fluorescent troffers. In 1962, Mueller and Benson (53) tested a variety of supply and return fixtures using a box calorimeter. They determined the amount of heat ducted away, the amount that went into the plenum, and the portion that entered the room over a range of return rates. Sylvester(70) ran similar tests on a specific troffer two years later. Ballman, Bradley, and Hoelscher (3) describe the use of a box calorimeter and relate some of their results and calculations of heat to room, plenum, and return air at different flow rates. Fisher, Roehr, and Smith (23) describe their findings after testing an experimental troffer with a simple box calorimeter to determine the amount of heat to return air at different flow rates. Dally and Ballman (16) ran more ambitious tests using complete calorimeter rooms with changes in the air supply and return systems and the lighting load. In all five of these papers, the results of steady state tests are presented and generally similar conclusions about the amount of heat going into the room space and return air are reached.

In 1965, the Illuminating Engineering Society (IES) published a five part review of work done in the area of lighting and air conditioning (39). The first article gives background material on heat production in fluorescent, incandescent, and mercury lamps and the fraction of heat radiated and convected. The next part describes how the different types of fixtures transfer energy. The third section is focused on the interaction between lighting and air conditioning systems and gives

estimates of the percentage of the energy entering room, plenum, and return air for different systems. The last two articles describe methods for controlling lighting heat. The review also contains an extensive list of references on lighting and air conditioning. This review combines most of the work done up to 1965 dealing with heat from lighting and was incorporated into the IES Lighting Handbook (33).

Henderson (26), in 1966, described the use of heat balance equations to calculate steady state heat transfer from lights. He used nine detailed examples to illustrate how this method could be used under various conditions. This type of analysis appears in many other articles written since then, mostly for heating and air conditioning journals in an easily understood format (20,44,45,64).

Much more testing has also been done concerning the heat transfer from lighting. Boyer (8,9,10) conducted a series of tests using a calorimeter room with changeable air conditioning systems and wrote a few papers dealing with the results. In one, he discusses heat transfer in interior office zones (8). In another, Boyer (9) wrote about regions near the building perimeter. In a third paper, he writes on the radiant transfer characteristics of room surfaces (10). With these papers, he gives a clearer understanding of the relationship between air conditioning and heat from lighting.

In Japan, Kimura (36) headed a group that tested the heat transfer characteristics on many luminaire styles. Using a small calorimeter room, they tested the heat removal by vented

air at different flow rates. They also used a lighting fixture built with components that were shifted, removed, or replaced to vary its design. Changes in lamp construction or the number of lamp tubes affected the heat transfer more than changes in the air flow path.

In 1968, Nottage and Park (55) used a calorimeter room to conduct extensive tests on a lighting system. They determined the amount of heat going into the room, plenum, and return air as others had done. But they also measured temperatures and heat transfer rates at numerous points on the luminaire.

Nevins, Straub, and Ball (54) published a paper on factors affecting heat transfer in luminaires. They used a box calorimeter to measure the amount of heat to plenum air and to return air at different flow rates. They also changed the size and type of the fixture. They concluded that air flow routing is important, but that similar designs will have similar heat transfer characteristics. The type of calorimeter they used is now recommended by both ASHRAE and IES for testing luminaires.

In 1970, O'Brien and Krumland (56) described their work with mathematical modeling of luminaires. They measured the heat and light transmission of two luminaire designs. Then they developed a mathematical model to simulate the thermal exchange of the test luminaires. They modeled a luminaire as separate surfaces and used heat balance equations to calculate the temperature of each surface. The model was also used to predict radiant output. Overall, their model matched the actual results, except for some differences concerning radiant transfer.

The most detailed theoretical analysis of heat transfer from lighting was by Sowell and O'Brien (65) in 1973. They attempted to develop an accurate mathematical model and compared the results with the data of the Nottage and Park (55) tests. They divided the model into as many nodes as Nottage and Park (55) had temperature measurement points. Sowell and O'Brien (65) also had a complicated radiation analysis. The results showed good prediction of general trends, but there were also some major discrepancies.

Except for Sturrock (69), all works discussed so far have dealt with steady state analysis. The only recent literature concerning time dependent heat transfer from lighting comes from the Canadian sponsored group of Mitalas, Kimura, and Stephenson.

In 1968, Kimura and Stephenson (37) described a theoretical analysis of the transient aspects of lighting heat transfer. They used a simple model, having one heat storing surface, only a few surface nodes, and an arbitrary estimate of how much heat is extracted from the fixture by the return air. The transient nature was controlled by the floor slab with the response factor method developed by Stephenson and Mitalas (67,68). Kimura and Stephenson (37) concluded that transient cooling loads could be modeled by Eq. (1) using only two experimentally determined constants, A and B.

$$q_T(t)/P_T = 1 - Ae^{-Bt} \quad (1)$$

To obtain values for these constants, they constructed a calorimeter room in which several types of light fixtures and ventilation arrangements could be tested. In 1971, Mitalas and

Kimura (51) described the construction and some initial testing of the room. Mitalas (48,49) published the final results of the project in 1973. Mitalas (48) ran tests with changes in ventilation flow rates, furnishing, carpeting, and air supply and return systems. From the results, he decided the exponential decay model of Eq. (1) was satisfactory. Then he approximated Eq. (1) with an abbreviated series expression which allowed hourly changes of lighting power. The equation is:

$$q_T(t) = a_1 P_T(t-\Delta t) + a_2 P_T(t-2\Delta t) + b_1 q_T(t-\Delta t) \quad (2)$$

The constant a_1 is determined from their test results. The values for b_1 are obtained using the mathematical model of Kimura and Stephenson (37). The other constant is given by:

$$a_2 = 1 - b_1 - a_1 \quad (3)$$

This method was a major advance in predicting the total heat load from lights because it accounted somewhat for the transient nature of the problem. But in the test room, only a few of the many possible building conditions could be considered, thus use of the model is limited. Also, comparison of actual test results with model predictions shows that Eq. (2) overestimates the total heat load for several hours after the lights are turned on.

This thesis draws its theory mostly from the works of Mitalas, Stephenson, and Kimura (37,48,49,51) and the paper by O'Brien and Krumland(56) and the one by Sowell and O'Brien (65). Results are compared against the test room transient results of Mitalas (48). Comparisons will also be made with steady state box and room calorimeter test results from other papers.

CHAPTER III

DEVELOPMENT OF THE MODEL

PRIMARY ANALYSIS

General Model Requirements

The purpose of this thesis is to design a mathematical model which may be used to predict the transient values of heat loads q_T and q_R . The model must also be flexible enough to use in a parametric study of room and lighting conditions. To begin the analysis, the results to be obtained and the factors that might affect these results must be specified.

The model has to predict when and where the energy input to the lights becomes part of q_T . To fit in with the present ASHRAE calculation procedures, these predictions must be made at least once an hour (1). In order that engineers can estimate the necessary supply air temperature and flow rate, q_R must be separable from q_T . The temperatures in the light fixture, the plenum space, and at the ceiling surface are needed to further validate the model. The radiant output of the lights is needed as another check.

The model has to be adaptable to any room configuration that might be used. Variables and conditions that could affect the heat transfer from lights are:

1. Room spatial dimensions.
2. Construction used in walls, floors, and ceilings.
3. Conditions in adjacent spaces.

4. Windows, furniture, carpeting, and draperies.
5. Room, plenum, lamp, and supply air temperatures.
6. Construction, lens material, coatings, ballast, and the number of fluorescent tubes in luminaires.
7. Variations in lamp tube efficiency due to temperatures.
8. The ratio of lighting energy to floor area.
9. Supply and return ventilation arrangements.
10. Surface emittances and absorptances.
11. Leakage of air through suspended ceilings.
12. The amount of time lighting is on and off.

Changes in any of these areas can be accounted for by the model described in this thesis.

The mathematical model will simulate the conditions in a single building section similar to that shown in Fig. 1. This building section is divided into three main parts:

1. Room - the basic occupied section of the model.
2. Plenum - the space above a suspended ceiling.
3. Luminaire - the lamp tubes, ballast, and lamp housing.

This specific building section is used because it is similar to a portion of many common office buildings. Most other types of buildings can also be simulated by using methods described in this thesis.

General Heat Transfer Considerations

The heat exchange analysis in this model is assumed to apply to both heating and cooling operations. The values of q_R and q_T can be linearly added in with other cooling or heating calculations. It is also assumed that the values of q_T and q_R

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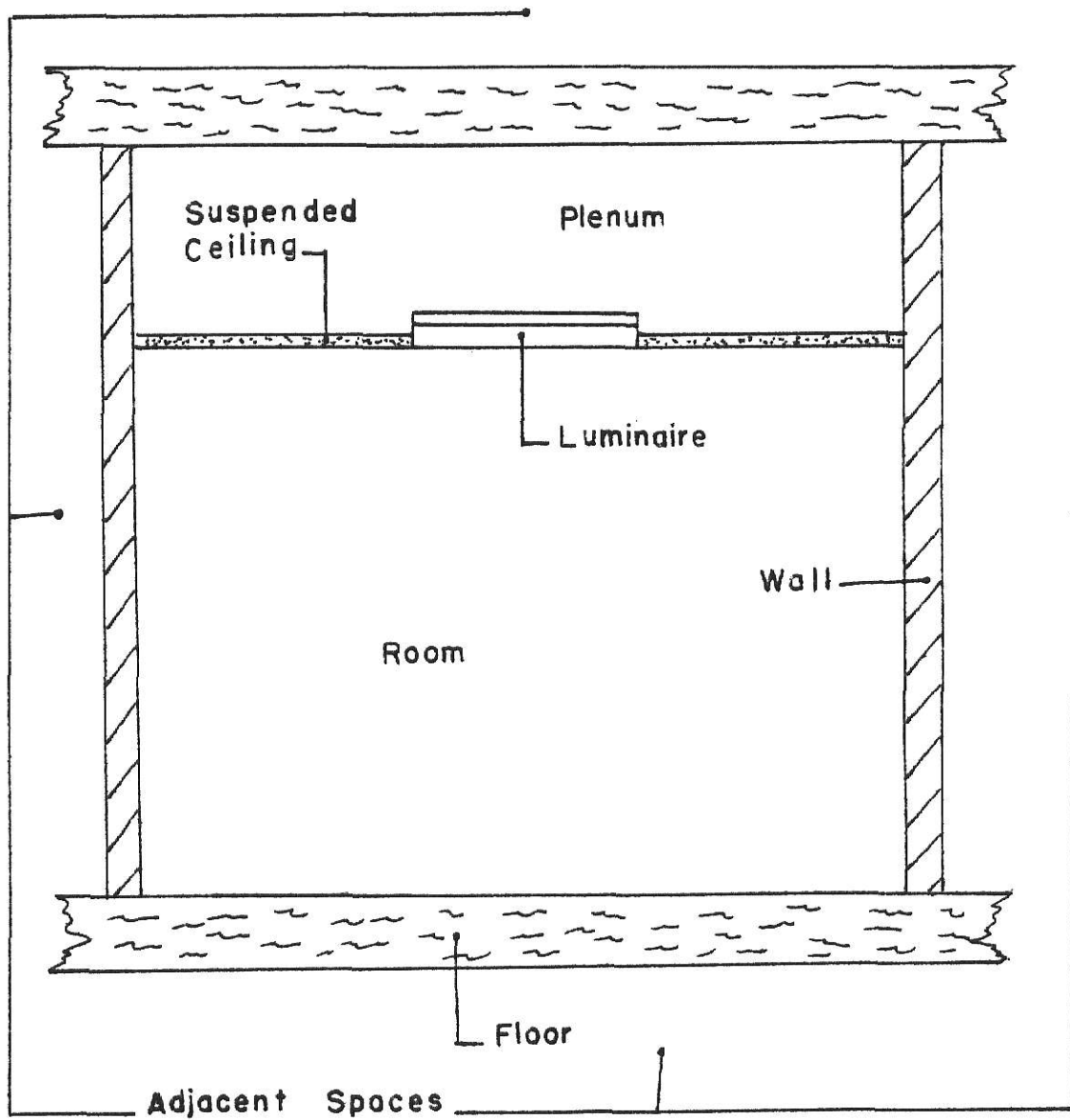


Figure 1. The Model Building Section.

are not affected by the size of other heat transmissions. These simplifications are made in the present ASHRAE calculation procedures for lighting and other heat sources (1). It is possible that the true relationships between different heat sources are not this simple, but to model interrelationships between different heat sources would be extremely complicated, and certainly beyond the scope of the present study.

Most of the initial concepts that form a basis for the mathematical model apply equally to the room, the plenum, and the luminaire. All three sections of the building model are divided into many surfaces or nodes. In each section, the heat transfer between surfaces is calculated using Euler's transient finite difference method. And the same basic methods for radiation, convection, and conduction heat transmission are used in each section. These basic, shared ideas will be developed first.

The room, plenum and luminaire are each divided into several surfaces or nodes. Typically, the room section has separate nodes representing the floor, the suspended ceiling, two pairs of walls, the lighting fixture surfaces visible in the room, furnishings, and the room air. In the plenum, there are nodes for the suspended ceiling, the floor bottom, two pairs of walls, ducting, luminaire surfaces, plenum air, and duct air. The luminaire divisions might be lamp tubes, the lens, the ballast, several strips of housing, and the lamp air. These surface divisions are shown in Figs. 2, 3, and 4.

In the room, plenum and luminaire, each surface represented by a node has all properties evenly distributed over its entire

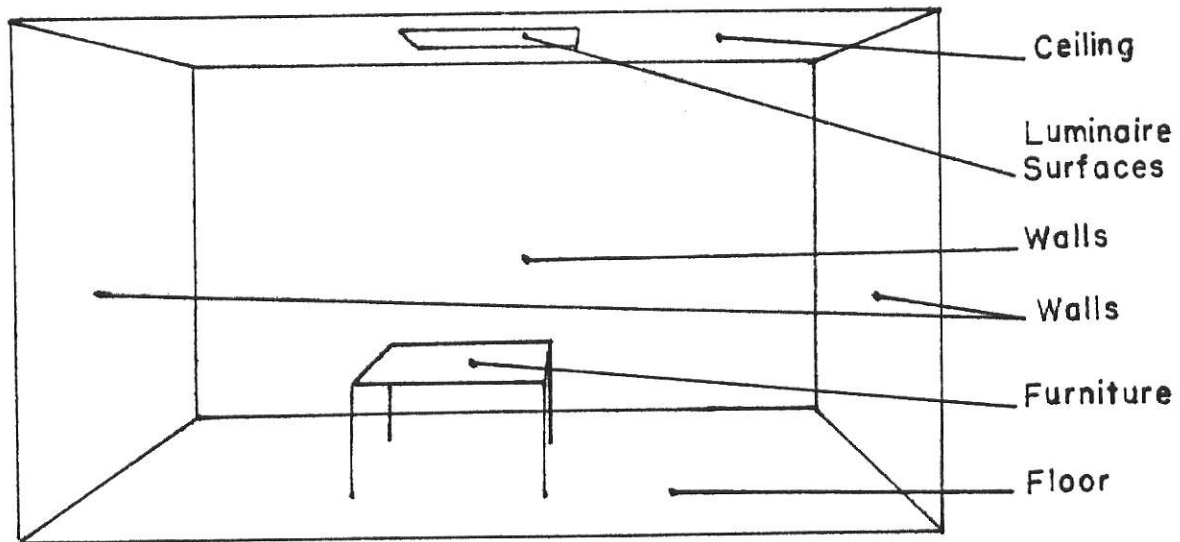


Figure 2. Room Nodes.

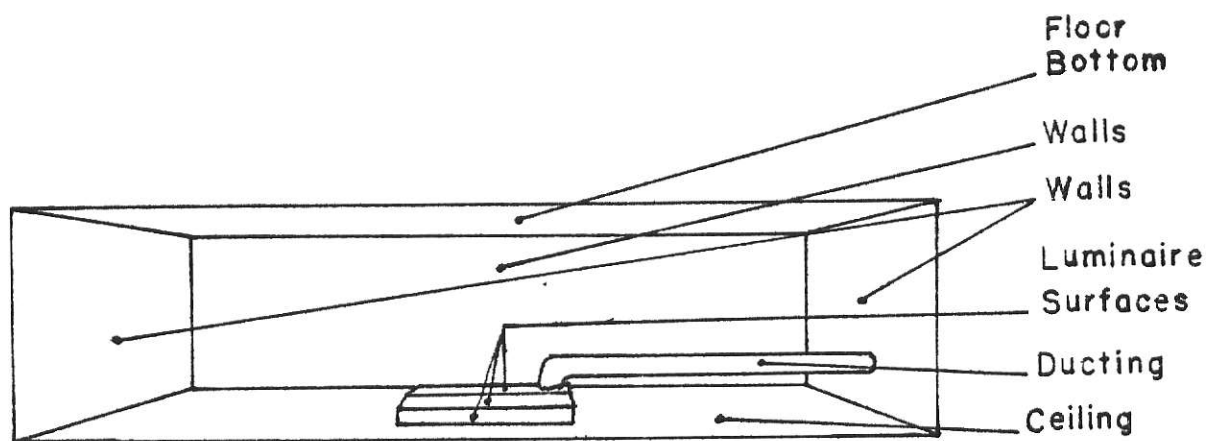


Figure 3. Plenum Nodes.

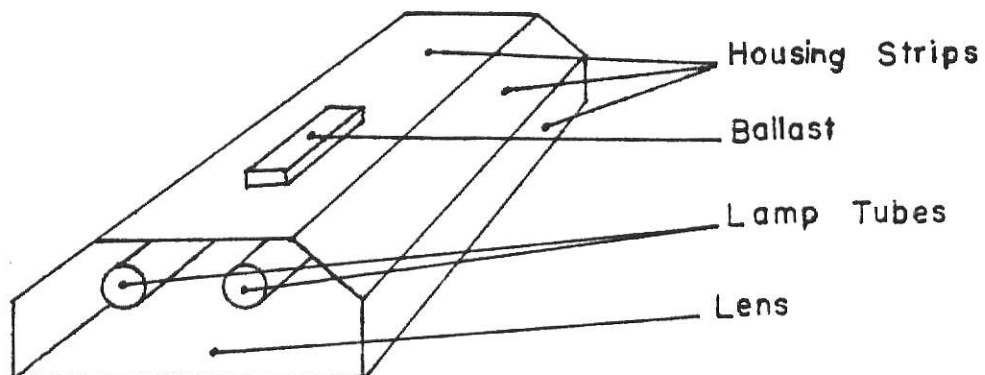


Figure 4. Luminaire Nodes.

area. The properties of emittance, conductivity, specific heat capacity, temperature, thickness, and density are assumed equal at all points on a surface. Radiant, convective, and conductive heat transmissions are also assumed uniform over each surface. Further, the properties of temperature, density, and specific heat capacity are also assumed uniform for room, plenum, luminaire, and duct air masses. These simplifications may need revision for certain cases, but they will be used as a starting point.

The calculation of heat transfer between the nodes in each section is done by using explicit finite difference methods. First, a time interval, Δt , is chosen, during which all properties and heat transfer rates are considered constant. Then the thermal energies transferred into or out of each node by radiation, convection, and conduction during Δt are summed to give the net heat gain for each node. The temperature rise for a node during Δt is equal to the net heat gain for the node divided by its thermal capacitance. The relations are:

$$q_{kj} = q_{kj}^{\text{Radiant}} + q_{kj}^{\text{Convective}} + q_{kj}^{\text{Conductive}} \quad (4)$$

$$q_j = \sum_k q_k \quad (5)$$

$$T_j^+ = T_j + q_j / C_j \quad (6)$$

After all of the new temperatures are found, the properties that depend upon time or temperature are recalculated. The new heat transfer rates are then computed and the process repeated. In this manner, the mathematical model is marched along by increments of Δt .

This is a standard predictor procedure, using Euler inte-

gration. An improvement in model accuracy could be made by using a predictor-corrector algorithm. Whether or not the improvement in accuracy would be sufficient to justify the additional computation cost (which would be substantial) will be investigated in a later phase of the project.

The size of the time period, Δt , affects the accuracy because nonlinear heat transmission and heat storage are better approximated when a smaller time period is used. But the time period cannot be reduced just for better accuracy, because a smaller time period requires more calculations.

Care must also be taken to insure stability when using transient finite difference methods. If the time period is too large or the heat capacitance of a node too small, temperatures may fluctuate wildly from period to period. To have a stable system, inequality (7) must be satisfied at all nodes.¹

$$\Delta t < \{C_j / (\sum_k K_{kj} + \sum_k H_{kj})\} \quad (7)$$

DETAILED HEAT TRANSFER ANALYSIS

Radiation Heat Transfer

Radiation transfer is calculated using the total exchange factor method of Hottel and Sarofim (28). The relation is:

$$q_{jk} = \overline{S_j S_k} \Delta t \sigma (T_j^4 - T_k^4) \quad (8)$$

¹There should also be some nonlinear radiation factor in the denominator of inequality (7). This would insure stability, but would also require a Δt smaller than necessary (19). Because radiant exchange within a building section is small, inequality (7) is sufficient. Although radiation from lighting is large, its transfer does not depend upon surface temperature, thus it does not affect stability (24).

$\overline{S_j S_k}$ are called the total exchange factors. They are functions of room geometry and surface emittances only (28). Because surface temperatures are the only quantities in Eq (8) that change over time, the calculations for radiation exchange are simple. A description of the method used to find the $\overline{S_j S_k}$ factors is given in Appendix B.

Hottel and Sarofim's total exchange factor method is well known, but it is a gray, diffuse model and thus requires several assumptions to be made (63):

1. The temperature of any surface is considered uniform over its whole area.
2. Radiation is uniformly distributed over a surface.
3. Air is a nonparticipating medium, equal to a vacuum.
4. Surfaces are gray: $\epsilon_j = \alpha_j = 1 - \rho_j$, and these properties are constant for the wavelength range under study.
5. Surfaces are diffuse, reflecting and emitting radiation equally in all directions.

Fortunately, none of these simplifications will produce large errors when this method is used to model a normal building environment. The temperature and radiation for each surface have already been assumed uniform in order to use the nodal approximation method. At normal building temperatures, air is a non-absorbing, non-emitting medium (66). Carbon dioxide and water vapor make small contributions, but these are insignificant at room temperatures. For carpet, wood, paint, glass, plastic, and other non-metallic surfaces, the emittances and absorptances are nearly equal. Such surfaces also closely fit

diffuse approximations (66).

A search of the literature dealing with the modeling of radiation transfer showed that a diffuse, gray model would be accurate enough. Several computer studies have been made comparing diffuse analysis against models that account for specular and spectral properties. A few papers show that sizable differences could occur between diffuse and more complicated models (5,6). But other computer studies and all experimental results reviewed show that diffuse models are as accurate as other types when at similar temperatures and in the presence of at least one diffuse surface (21,27,29,62,71,72).

One problem with this diffuse, gray analysis is that real surface emittances are not constant for different wavelengths. Surfaces at high temperatures have different emittances and emit a larger portion of radiation at shorter wavelengths than they do at lower temperatures. Also, lens materials such as glass and acrylic are transparent to short wavelength radiation, but opaque to longer wavelengths. But for most room materials, radiative properties can be assumed uniform at wavelengths on either side of the near infrared - far infrared division without producing large errors. At room temperatures, surfaces emit most of their energy in the far infrared range. But, when fluorescent lights are on, they emit substantial energy at visible wavelengths. To account for this, there are two sets of equations for radiation exchange; one is for long wavelengths, and the other is for short wavelength radiation from lighting.

Convection Heat Transfer

The transfer of heat by convection is modeled by using similar equations in the room, the plenum, and the luminaire. In all cases, heat is transferred from solid surfaces to adjacent air spaces. The equation for convective transfer from a surface to an air mass is:

$$q_{ja} = A_j h_j \Delta t (T_j - T_a) \quad (9)$$

The value of the convection coefficient, h_j , depends upon many things. It is affected by the orientation and shape of the surface or object concerned, the temperature difference between the surface and the air, and the velocity of the airflow around the surface. All of these factors came under consideration when estimating convection coefficients.

Conduction Heat Transfer

The equations used to calculate heat transmission by conduction in room, plenum, and lamp surfaces are not as general as the other equations used so far. This is because of the need to account for heat storage by surfaces. Some surfaces have little thermal capacitance, while other surfaces are heavy enough to store considerable energy. Thus there are two general groups of heat conduction equations.

Lightweight surfaces such as luminaire surfaces, ducting, suspended ceilings, and windows have little thermal capacitance, thus they do not store enough energy to affect the transient nature of the problem. Still, conduction through these surfaces is important. The values of their surface temperatures are also important. Therefore, these surfaces are treated with simple finite difference or steady state methods. Because each type of

surface is different, slightly different methods are used in each case. The details are given separately in later sections of this chapter.

The other group is comprised of heavy surfaces such as floors, walls, and large furnishings. These surfaces may need much time before they approach equilibrium with their surroundings. This is the main reason for time delayed heat release in buildings. Therefore, an accurate method is needed for calculating the rates of heat storage and release from these surfaces.

The method used must meet several requirements besides accurately estimating heat storage: it should give good estimates of surface temperatures for radiation transfer analysis; it must be amenable to changing energy inputs; the calculations should not require too much computer time. The two methods which seem to fit these constraints are the transfer function method and the transient finite difference method.

The transfer function method is a convenient method of using a computer to calculate the transient thermal characteristics of surfaces (50,67,68). With this method, the heat going into a surface is modeled by a time series of pulses. These pulses are then multiplied by response factors, which account for the transient nature of heat release. The weighted pulses of past time intervals are then summed to produce the total heat release during the present time interval. The response factor method is easy to use, but all heat transfer relations are assumed to be linear.

The explicit finite difference method is also a handy meth-

method of solving dynamic heat storage problems (15,19,25). The partial differential equations of heat conduction are approximated using volumes and temperature differences of finite size. The accuracy of the model improves as the size of the volume units decrease. This method may require more computer time than the transfer function method, but the calculation of heat storage is only a small part of the present problem (50). When the two methods were compared, they appeared equally accurate (15, 35).

The explicit finite difference method was selected for three reasons:

1. It can be used easily if convection coefficients vary, while the transfer function method becomes more complicated (50).
2. It does not require the assumption that all heat transfer relations are linear.
3. Similar finite difference methods are already used in this mathematical model, thus offering a possibility of simplified equations when the model is made into a computer program.

With this finite difference method, a surface is modeled as shown in Fig. 5. The surface is divided into several layers of equal thickness, the number of layers depending upon the thermal capacitance and conductivity of the material. Each layer is modeled as a node in the middle of the layer. The temperature of the node is the mean temperature of the layer. The flow of heat is assumed to be one dimensional. The conductance between

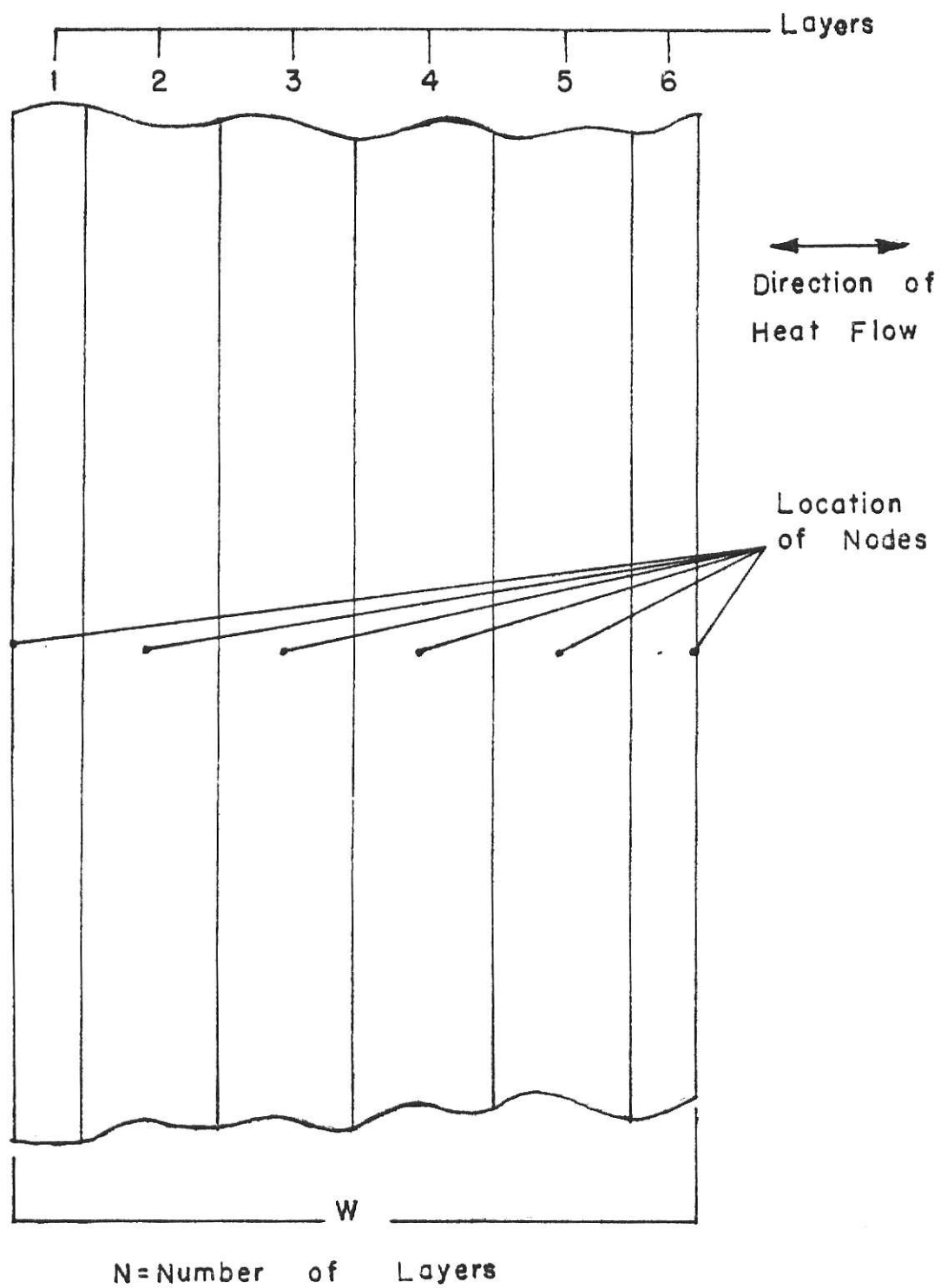


Figure 5. The Heavy Surface Model.

two nodes is:

$$K_{jk} = kA(N - 1)/W \quad (10)$$

The thermal capacitance of an interior layer is:

$$C = Ac_p W/(N - 1) \quad (11)$$

The expression for heat conduction between any two nodes is:

$$q_{jk} = K_{jk} \Delta t (T_j - T_k) \quad (12)$$

The temperature of each node is incremented by using Eqs. (4), (5), and (6).

The surface layers are modeled half as thick and with half the thermal capacitance of the interior layers and the node is located at the surface. This leaves the conductance to the next node unchanged. The reason for the half layers at the surface is to give the correct temperature for convective and radiant heat transfer. The correct steady state heat transfer rate is also obtained by using the smaller surface layers. While the interior nodes transfer heat only by conduction, the surface nodes also exchange energy by radiation and convection according to Eqs. (8) and (9). Because surface layers have only half the thermal capacitance of the other nodes, while the sum of the conductances to the surface nodes has not been reduced by half, system stability must be checked only at the surface layers for this type of model.

SPECIFIC COMPONENT ANALYSIS

So far, the basic theories and equations used in this mathematical model have been formulated. Several assumptions were made to simplify the analysis. The model has also been divided

into three parts: the room, the plenum, and the luminaire. And the basic equations for radiant, convective, and conductive heat transfer have been stated. All further analysis is built around these foundations. The room, plenum, and luminaire sections each have different conditions and components that require special approaches. The rest of the mathematical modeling in this thesis describes each component individually.

The Room Section

The first part of the building model to be considered in detail is the room. Accurate models must be designed to simulate room air, conditions in adjacent areas, carpeting, furniture, draperies, and windows. Another area that must be analyzed is the calculation of convection coefficients.

The room air is assumed to be maintained at a given temperature by a perfect air conditioning and heating system. It is not necessary to include calculations for heat storage or worry about stability considerations for the room air. For now, the supply air and the room air are assumed to be perfectly mixed as soon as the supply air enters the room. The supply air temperature is equal to the room air temperature less the quotient obtained when the room load, q_R , is divided by the heat capacitance of the supply air.

The situations existing in adjacent rooms or air spaces are major variables. In a model of the type shown in Fig. 1, it is often assumed that the adjoining spaces on all four sides as well as above and below are identical to the modeled building section. This permits the assumption that the center layers of

the walls are adiabatic and that energy leaving through the floor above equals that which enters through the floor below.

For the present study, a different method is used. Equations are supplied which describe the heat absorbed by the outer half layer of the walls, of the floor, and of the ceiling. To model a room surrounded by six similar areas, the outer layer heat absorption values are set equal to the amount of heat transferred at the corresponding inner layers. But this method is flexible, because the outer layer heat inputs can be set to simulate any desired conditions in adjacent spaces.

Carpeting can alter the transient thermal response of a room and must be accounted for by the mathematical model. This is done by modeling it as another layer above the floor as shown in Fig. 6. This layer is equal to a half layer with its node located at the top surface. The finite difference equations are similar to the regular ones. The conductance to the top floor node is calculated using the thickness and conductivity of the carpeting.

Tables, chairs, counters, shelving, and other furnishings can affect q_R . Such items are modeled as slabs composed of several layers and treated by finite difference methods. An example of a table and its model is shown in Fig. 7. The conductance between nodes depends upon an estimate of the average conductivity and thickness of the item. The convection and radiation energies transferred to this type of model are assumed uniform and each surface receives half of this heat. No conductance to the floor through legs or stands has been considered.

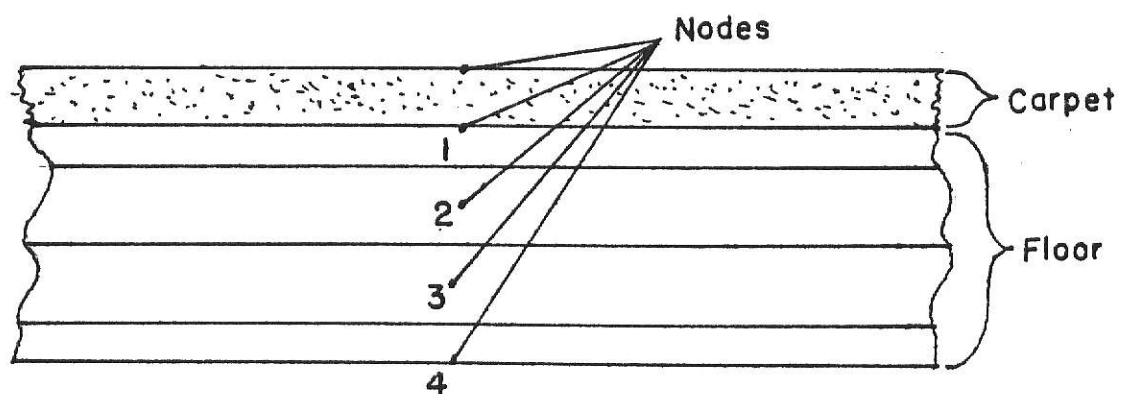


Figure 6. The Carpet Model.

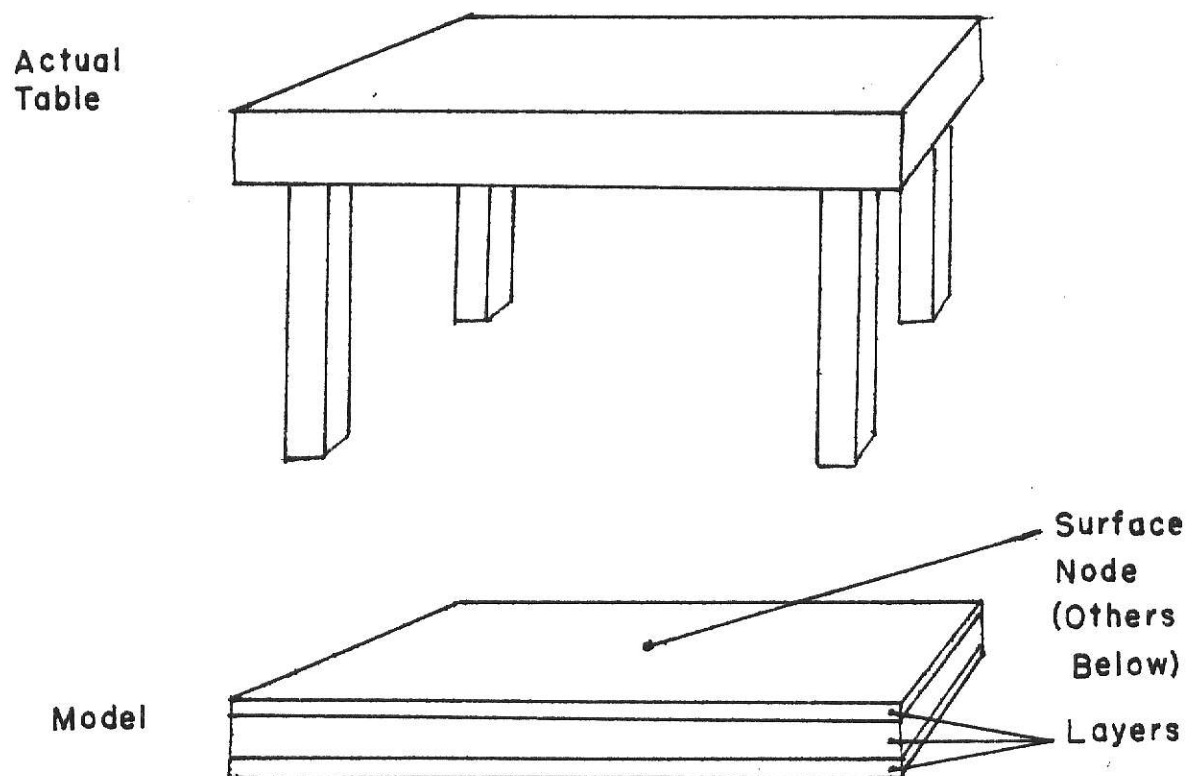


Figure 7. A Table Model.

This is a simple model of furnishings, but q_R is mostly affected by changes in furniture mass, surface area, and material properties, and these items are accounted for by using this model.

Draperies and other lightweight furnishings are treated differently. Such items do not have enough thermal capacitance to be stable over a reasonable time period. It is assumed that these surfaces quickly reach thermal equilibrium, so that the radiation which they absorb is convected to the room air instantaneously. Thus, their temperatures are set equal to the values needed to convect this energy to the air. The equation is:

$$T_j = T_a + q_j / (A_j h_j \Delta t) \quad (13)$$

where q_j equals the radiation to lightweight surface j that is convected to the room air.

Windows are another type of lightweight room surface. Glass is transparent to short wavelength radiation, but opaque to long wavelengths. To make the total radiation exchange factor calculations work correctly, the emittance must equal one minus the reflectance. Therefore, short wavelength emittance for transparent surfaces is given by:

$$\epsilon_s = 1 - \rho_s = \tau_s + \alpha_s \quad (14)$$

This emittance is used in radiation calculations to find the amount of short wavelength energy that goes into the glass. The fraction of this energy that is transmitted through the glass is equal to τ_s divided by the new ϵ_s . The rest is absorbed by the glass. The long wavelength calculations are all normal. The sum of the absorbed energies is assumed to be convected away, with half going outside and half being convected to the room

air. The window is assumed to be at equilibrium between the inside and the outside air temperatures. The temperature drop from one side of the glass to the other is so small that the window temperature is assumed to be the same on both sides.

Then the equation for the glass temperature is:

$$T_g = (T_o h_o + T_i h_i) / (h_o + h_i) \quad (15)$$

The estimation of convection coefficients is one of the most uncertain areas of this mathematical model. A literature search showed that most equations for buoyant convection coefficients were obtained from experiments using larger temperature differences than are common in rooms (11,14,47,58,73). But most of these models are similar, the formula being:

$$h = A(|T_j - T_a|)^B \quad (16)$$

where: $0.28 \leq A \leq 0.48$ for heat flow up

$0.22 \leq A \leq 0.38$ for vertical walls

$0.15 \leq A \leq 0.22$ for heat flow down

$0.25 \leq B \leq 0.33$

One paper was located that showed the results of experiments done with low temperature differences (73). The values for A are 0.83, 0.50, and 0.38 respectively. The value for B is 0.12, except for heat flow down, where B is zero. It was decided to use these last values for A and B in the initial modeling at least.

Another factor that can change the convection coefficients is the air movement. The method used here is to estimate a mean air velocity, which depends upon ventilation rate and supply

outlet design (46). This value is then used in a standard convection coefficient equation for laminar flow, which is:

$$Nu = 0.664Re^{.5}Pr^{.33} \quad (17)$$

and assuming the air is at 80 F, the equation for a forced convection factor becomes:

$$h = 0.0115(V/L)^{.5} \quad (18)$$

These forced convection factors are small at room air velocities. One method of accounting for both buoyant and forced convection is to sum the values from Eqs. (16) and (18) for the total convection coefficient (2). This combined equation method is how room convection coefficients are approximated in the mathematical model.

The Plenum Section

If the building section has a suspended ceiling, the heat transfer for plenum components must be analyzed. The luminaires are often recessed into the plenum, causing much of the thermal energy from lighting to be first transferred to plenum surfaces or the plenum air mass. After these masses are heated, they transfer increasing amounts of heat through the suspended ceiling and the floor above until the entire system reaches steady state. Also, the use of vented plenums can cause much of q_T to be picked up in the plenum. And the operating conditions of the luminaires, and thus their efficiencies, depend upon plenum temperatures. For these reasons, the analysis of plenum components is just as detailed as the analysis of room components.

Several sections of the plenum analysis are modeled as they are for the room. Conduction transfer through walls is modeled

using the same method. The same equations are used for long wavelength radiation transfer. Because the lights shine into the room only, the short wavelength analysis is not needed in the plenum. The convection coefficients for plenum surfaces are calculated with the same equations as are used in the room. Values for the mean plenum air velocity are assumed to be the same as the room if the plenum is vented, but zero if a static plenum is modeled.

Most of the differences between the methods used to model plenum components and room components result because of stability considerations. From stability analysis of several types of walls and floor slabs, a time interval, Δt , of 0.5 hours seemed appropriate. But similar calculations on suspended ceilings, luminaire surfaces, and ducting resulted in maximum time periods of 0.025 to 0.05 hours. This meant ten to twenty times as many iterations would be needed. And it was felt that these surfaces are critical enough that steady state analysis would not be sufficiently accurate.

To obtain accurate results and still keep computer time to a minimum, the computer program is divided into two main sections. One part is for heavier surfaces and the other is for nodes with small heat capacitances and is processed five to ten times as often as the first part. During the shorter time interval, $\Delta t'$, the temperatures at heavy surface nodes are constant, while the lightweight nodal temperatures change. Energy exchange from lightweight to heavy surfaces is summed for each smaller interval, then put into the standard time period section

of the program. There was still a large increase in the amount of computer time used.

Much of the heat in the plenum eventually passes through the suspended ceiling to the room. To correctly analyze convection and radiation from the ceiling surfaces, a good idea of their temperatures is needed. The ceiling is therefore modeled as shown in Fig. 8. The model is a two layer slab with nodes on the surfaces and finite difference methods apply. By doing this, the surface temperatures and the heat flux through the ceiling are correct.

The temperature of the plenum air mass changes considerably and this affects both q_R and q_T . But the plenum air mass is often too small to be stable even when using the shorter time intervals. Initially, the plenum air was given an even smaller time period and the air temperature was changed several times while the temperatures of other plenum surfaces remained constant. But this required more computer time and a more complicated program.

What this actually yielded was an approximate numerical integration with air temperature as the only variable in the equation during the interval. The actual integration could be done easily and take much less computer time. As the plenum air temperature changes during $\Delta t'$, it passes through a mean value, $\bar{T}_a$. The plenum surfaces would transfer the same amount of heat to the plenum air at this temperature as would be transferred using an exact solution.

The air can now use the same time period as other light-

weight plenum surfaces. The equation for convection transfer to the plenum air from all plenum surfaces is:

$$q_a = \sum_j A_j h_j \Delta t' (T_j - \bar{T}_a) \quad (19)$$

The equations for $\bar{T}_a$ are:

$$S1 = \sum_j A_j h_j; \quad S2 = \sum_j A_j h_j T_j; \quad S3 = S2/S1 \quad (20)$$

$$Z = S1 \Delta t' / C_a \quad (21)$$

$$\bar{T}_a = S3 + (T_a - S3)(1 - e^{-Z})/Z \quad (22)$$

These equations work equally well for both vented and static plenums. The heat capacitance of the air in Btu/°R equals 0.018 times the volume of the air in ft³. If the plenum is static, the air volume equals the plenum volume. If the plenum is vented, the air volume is either the plenum volume or the volumetric flow rate per time period, whichever is larger. Derivation and use of this mean air temperature method is described in Appendix C.

Besides luminaire surfaces, the only other items in the plenum of any concern are ducts. The ducting will usually have a small surface area and very little heat capacitance. Therefore, steady state analysis can be used without much error. The model used for ducting is shown in Fig. 9. The total thermal transmittance (U-value) for the ducting is obtained from test results from Lauvray (38). The outer convection coefficient is calculated by Eqs. (16) and (18) with A equal to 0.22 and B equal to 0.33. These values are used because there will probably be a sizable temperature difference. The conductivity is a property of the duct material. The relation for the total thermal transmittance is:

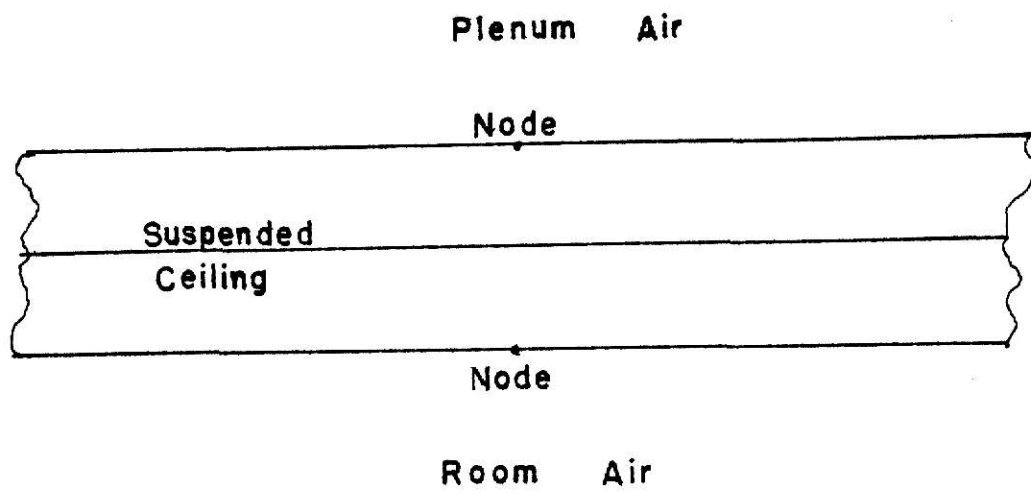


Figure 8. Suspended Ceiling Model

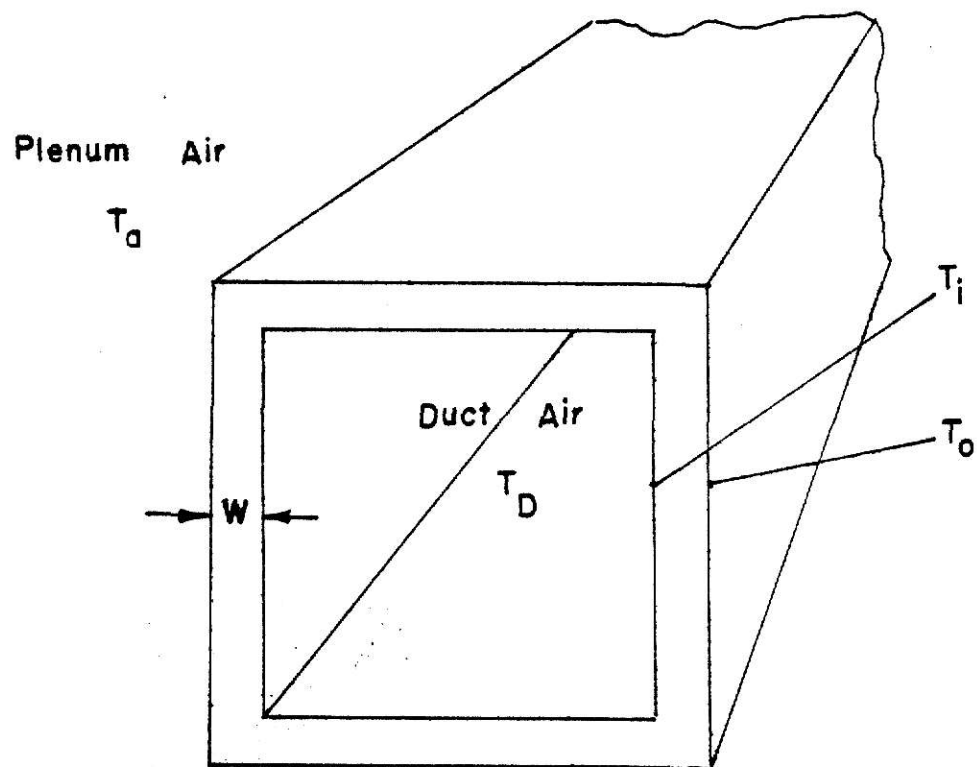


Figure 9. Ducting Model

$$UA_i = 1/\{(A_o h_o) + (W/k\bar{A}) + (1/A_i h_i)\} \quad (23)$$

where: $\bar{A} = (A_i + A_o)/2$

From this, the unknown, inside surface convection coefficient is:

$$h_i = (1/A_i)/\{(1/UA_i) - (W/k\bar{A}) - (1/h_o A_o)\} \quad (24)$$

The expressions for the temperatures at the inside and outside duct surfaces are:

$$T_i = \bar{T}_D + (T_a - \bar{T}_D)(U/h_i) \quad (25)$$

$$T_o = T_a + (\bar{T}_D - T_a)(UA_i)/(h_o A_o) \quad (26)$$

$\bar{T}_D$ is the mean duct air temperature as it passes through the duct. This temperature is obtained with analysis similar to that used for mean plenum air temperature. The mean duct air temperature is:

$$\bar{T}_D = T_i + (T_D^* - T_i)(1 - e^{-Z})/Z \quad (27)$$

where: $Z = h_i A_i / (0.018 \times \text{air flow rate})$

The amount of heat convected to the duct air is:

$$q_{iD} = h_i A_i \Delta t' (T_i - T_D) \quad (28)$$

Radiation and convection from the outside duct surface to other plenum nodes are handled normally. Because most ducting has low values of thermal conductance, all net radiant energy absorbed by the ducting is assumed to be convected to the plenum air.

The Luminaire Section

Because the luminaires are the source of all energy in the model, they get more attention than other objects. Luminaires are divided into many nodes, as shown back in Figs. 2, 3, and 4. Nodes visible in the plenum exchange radiant and convective en-

ergy using the same equations as other plenum surfaces. The convection coefficients are calculated using the same equations as for ducting. The calculations for heat transfer by surfaces visible in the room are the same as for other room surfaces.

Within the luminaire, there is both long and short wavelength radiation transfer. Long wavelength transfer is handled normally, using Eq. (8). The emittances and areas of the surfaces are included in the $\overline{S_j S_k}$ factors, as shown in Appendix B. But short wavelength radiation originates only from the fluorescent tubes and depends only upon the efficiency of the fluorescent tubes and the power to the lights. From this, the expression for the exchange of short wavelength radiation from light producing surface j to surface k is:

$$q_k = P_s \overline{S_j S_k} \Delta t' / (\epsilon_j A_j) \quad (29)$$

Note that the emittances and total exchange factors are for short wavelength radiation.

Short wavelength radiation to the lens is treated much as it is for windows. The emittance is according to Eq. (14). The amount of energy transmitted is also proportional to τ_s divided by the modified ϵ_s . But the rest of the energy is absorbed by the lens, not immediately convected to adjacent air masses.

The radiation transmitted through the lens is the amount of short wavelength radiation that enters the room. This becomes the value for P_s in Eq. (29), which is then used to calculate short wavelength radiation transfer within the room. The luminaire lens is the emitting surface for these calculations. The lens is modeled as diffuse, but directional properties of the

luminaire can be accounted for by altering the view factors when the short wavelength total exchange factors are calculated.

Because of the smaller dimensions and large temperature differences, the convection occurring in the luminaire cannot be modeled using the same equations as are used for large surfaces. Eq. (9) is still used, but the convection coefficients to the air are hard to obtain. Most of the work done with convection in small spaces has dealt with surface to surface transfer. In this model, because of the complicated geometries of the luminaire and the unknown flow paths, the heat transfer is calculated from surface to air.

Because the fluorescent tubes are the most important surfaces in the luminaire, they are studied first. With natural convection, the relation modeling the heat transfer from a horizontal tube is:

$$Nu_d = 0.53(Gr_d Pr)^{.25} \quad (30)$$

and using air at 100°F, the convection coefficient reduces to:

$$h_t = 0.25\{(T_t - T_a)/d\}^{.25} \quad (31)$$

With vented luminaires, there is both buoyant and forced flow factors affecting the convection process. Three papers were reviewed on this subject, and from these, the best model is (32,52,57):

$$Nu_d = \{1 + (Gr_d/Re_d^2)\}^{.25}(Re_d^{.5}/2) \quad (32)$$

and with a reference temperature of 100°F, this gives:

$$h_t = 0.0097\{(V^2/d^2) + 207(T_t - T_a)/d\}^{.25} \quad (33)$$

In Eq. (33), V is the velocity of the air through the luminaire. This velocity is approximated by the luminaire flow rate

divided by the luminaire cross section area.

No relations were found for convection in complicated enclosures. But convection is basically dependant upon the fluid motion of the boundary layer. With static luminaires, the air movement is controlled by the temperature difference between the air and the lamp tubes, because they are the hottest surfaces. For vented luminaires, the controlling factor is the flow velocity. The convection coefficients for the other interior luminaire surfaces are therefore set equal to the lamp tube convection factors. The program will be tested to check the consequences of these assumptions.

The remaining mode of heat transfer in the luminaire is conduction. Each surface is assumed to have a node right in its center, as shown in Fig. 10. The thermal resistance between nodes is equal to the resistance from one node to the mutual boundary plus the resistance from the other node to the boundary. The conductance is the inverse of the total resistance. Finite difference methods are used in this case, also. The total heat conducted into one node is:

$$q_j = \sum_k K_{jk} \Delta t' (T_k - T_j) \quad (34)$$

One node that depends greatly on conduction transfer is the ballast. The ballast is often enclosed by other surfaces and also receives about 20% of the total energy to the luminaire. If the conductances to the surfaces that surround the ballast are incorrect, the ballast temperature is likely to be very abnormal. Therefore, these conductances are adjusted until the correct ballast temperature results.

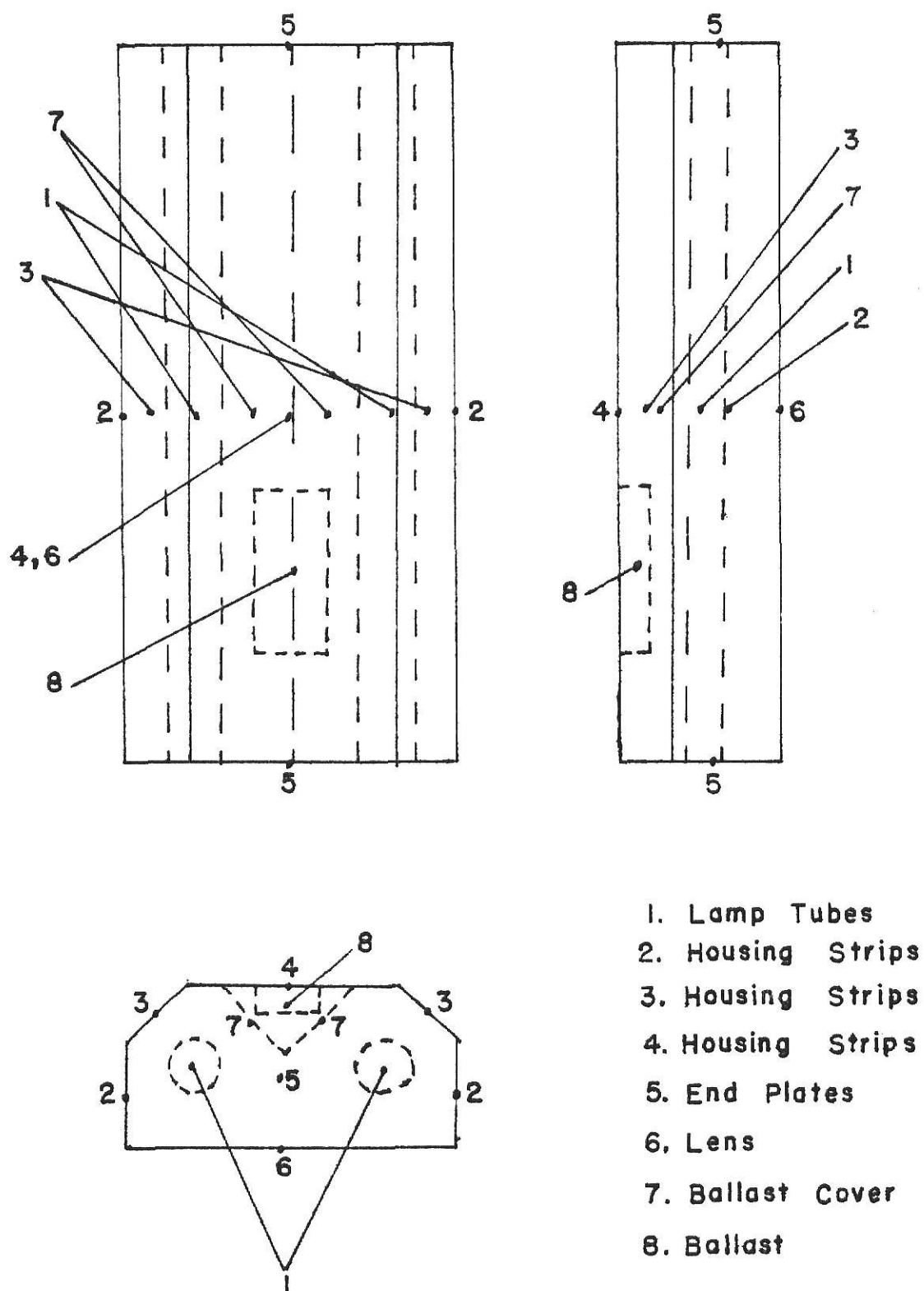


Figure 10. Luminaire Nodes.

The temperature changes in the luminaire are straightforward. The changes of the surface temperatures are calculated using Eqs. (4), (5), and (6). The calculation of the change in lamp air temperature is done with the same method as is used for plenum air.

The last items in the lamp to be modeled are the operating characteristics of the fluorescent lamp tubes. The total energy consumed by a fluorescent tube varies with the tube wall temperature, as shown in Fig. 11. The amount of light produced also varies with tube wall temperature. These two curves will be modeled with mathematical equations that fit in the operating range. A fraction of the total input to the luminaire (about 20% with 48 inch tubes) is lost to the ballast. Of the remaining energy, around 25% leaves the fluorescent tube as short wavelength radiation. The remaining energy is absorbed by the lamp tube wall, which heats up and radiates, convects, and conducts the energy away (33). The expressions used for lighting analysis are similar to these:

$$P = P_{\text{rated}}(0.8 + 0.2 \sin\{0.0393(T_t - 520)\}) \quad (35)$$

$$P_B = 0.2P \quad (36)$$

$$P_{\text{total}} = P + P_B \quad (37)$$

$$P_s = 0.1P_{\text{rated}}(1 + \sin\{0.397(T_t - 525)^{.5} - 0.89\}) \quad (38)$$

All of the constants and the shape of the curves may vary with the type of the fluorescent tube.

Most of the components and interactions important to the transient analysis of lighting heat transfer have now been mathematically approximated. Some things have been missed or by-

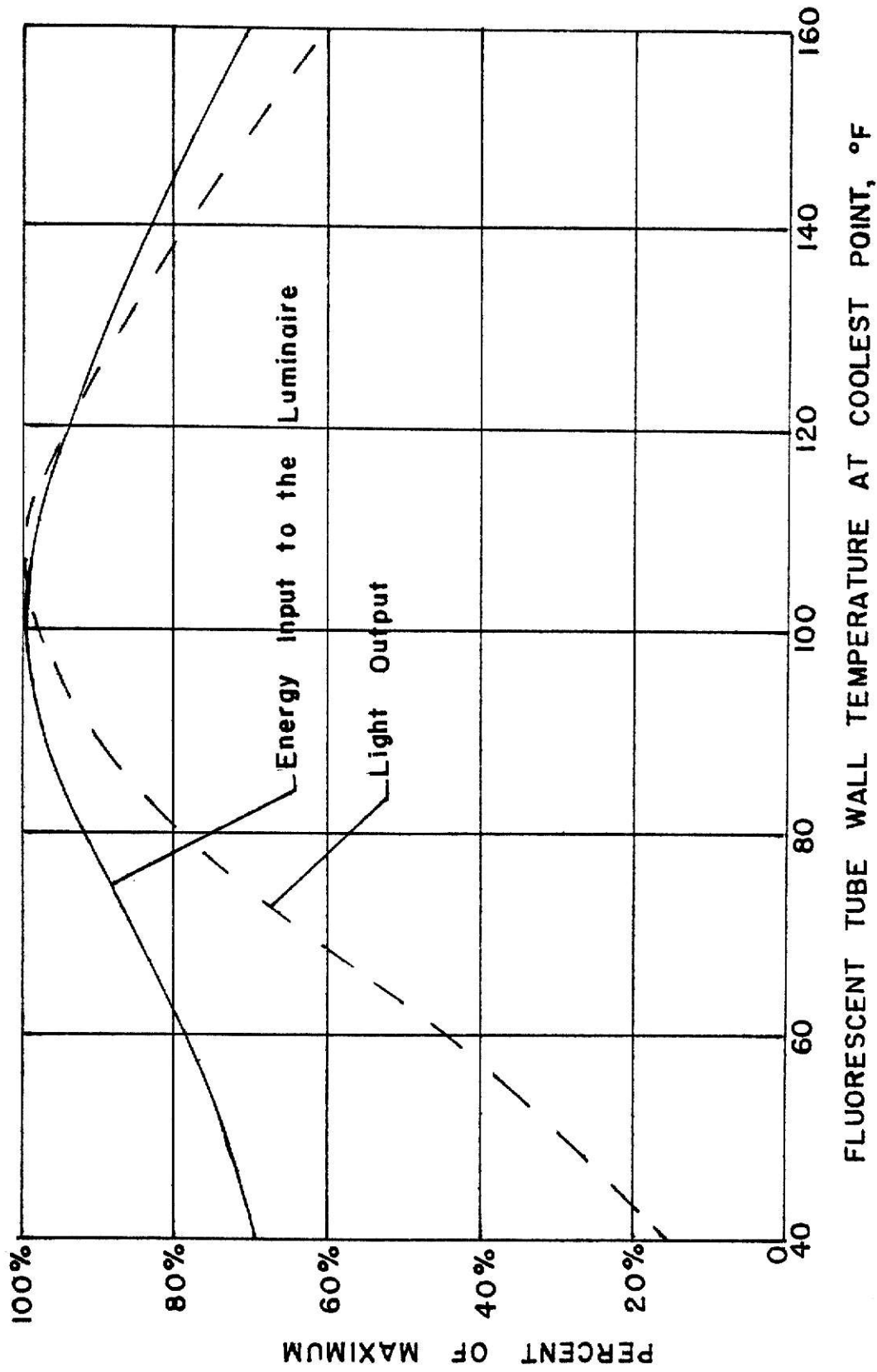


Figure 11. Typical Fluorescent Tube Operating Characteristics as Functions of the Fluorescent Tube Temperature.

passed by this model, but nearly any consideration can be modeled by building upon methods described in this thesis. A FORTRAN computer program was built which uses these equations and methods to describe the transient heat transfer within a building section. The program in Appendix D was designed around these equations, but also includes some changes that were inserted after comparing the model results with experimental results.

CHAPTER IV

COMPARISON OF MODEL PREDICTIONS

Once the computer program was running correctly, comparisons between the model predictions and data from previously published experiments were made. The only transient experimental results found in the literature concerning a full-sized calorimeter room with fluorescent luminaires were those of Mitalas and Kimura (48,51). The physical characteristics of this calorimeter room were used as input to the computer program and comparisons were made for several different ventilation arrangements and flow rates and for furnishing changes. Several other papers were used to compare temperatures and heat release from specific nodes for Mitalas (48) did not publish such data (8,10,36,54,55). These later papers are all concerned with steady state analysis only, thus the model was run until the values under consideration also reached steady state before these comparisons were made.

MODELING MITALAS' CALORIMETER ROOM

Sketches of the calorimeter room used by Mitalas and Kimura (48,51) are shown in Figs. 12 and 13 which are taken from reference (51). The calorimeter room was intended to simulate part of a large office space in a multi-story building with identical offices above and below. The room dimensions were 10 by 14 by 9 feet high. Above this was a suspended ceiling and a

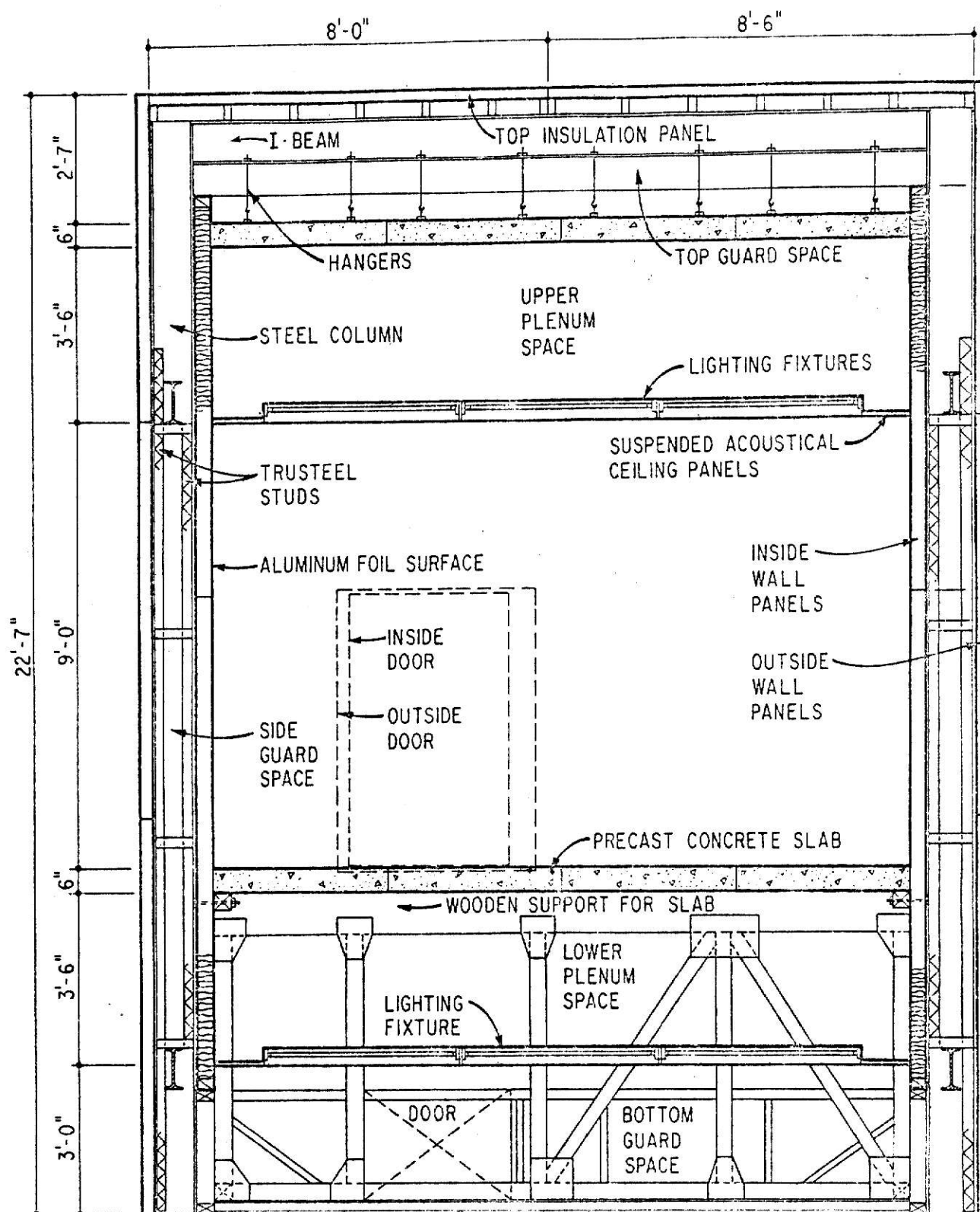


Figure 12. Side View of the Experimental Calorimeter Room of Mitalas and Kimura (51).

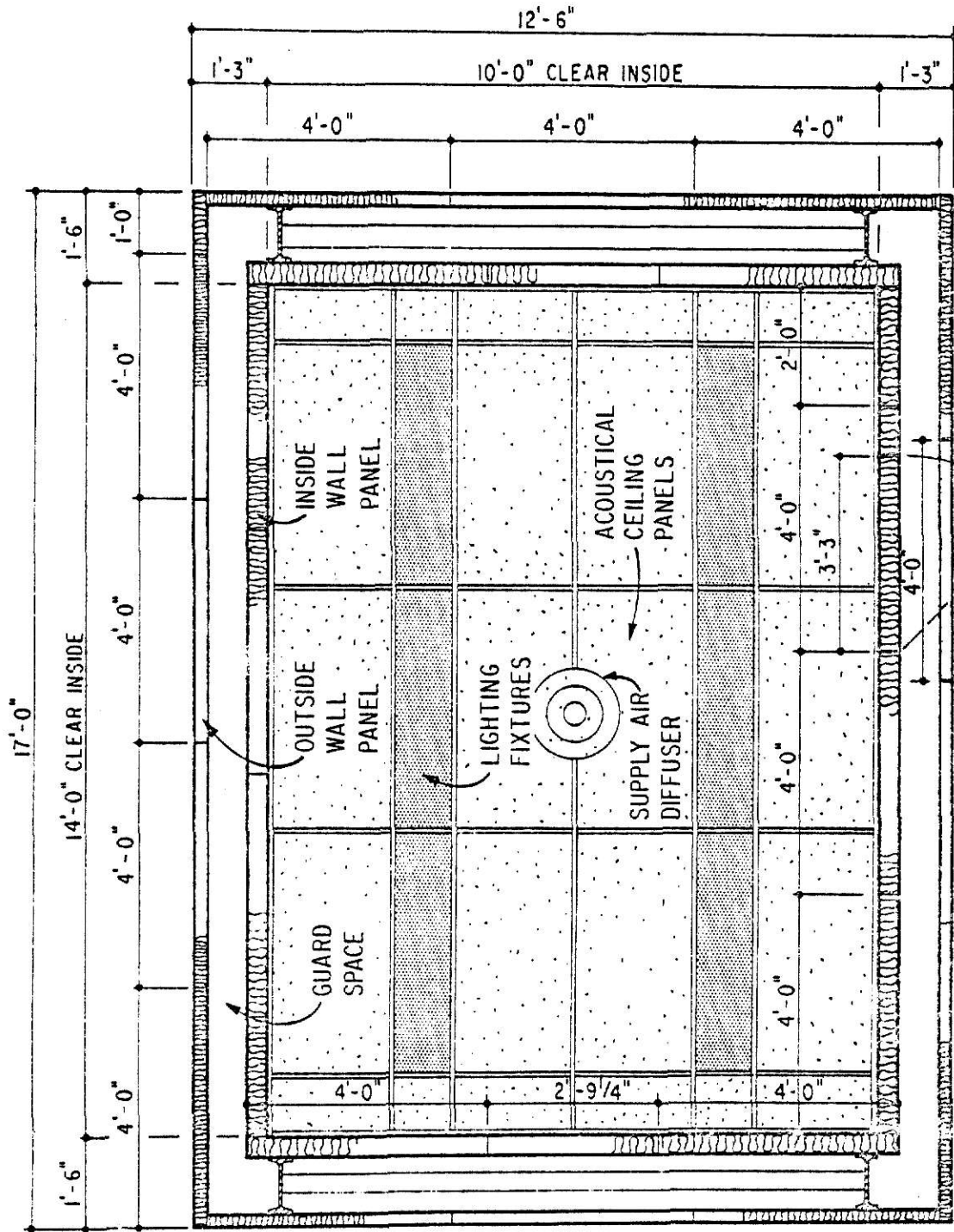


Figure 13. Top View of the Experimental Calorimeter Room of Mitalas and Kimura (51).

3.5-foot plenum space. There was a 6-inch concrete slab for the floor and another such slab above the plenum. This top slab had a layer of heated water on it to keep its surface at the same temperature as the room floor. The room was lighted by six recessed luminaires, each containing two 40 watt fluorescent lamp tubes. There were six more luminaires below the floor slab to simulate a similar plenum space below. The walls were 4-inch thick styrofoam slabs covered with thin plywood. The insides of the walls were covered with highly reflective aluminum foil and the outsides of the walls were in contact with air at the same temperature as the inside wall surface temperature. This construction was done in order to imitate a much larger office space where most of the lighting energy reaches the floor.

The input data were based on a simplified model of the calorimeter room as shown in Fig. 14. Instead of modeling the extra floor slab above the plenum and the lighting below the room floor, the heat entering from under the floor was assumed equal to the heat going up from the plenum. The outside of the walls exchange heat by convection with air at the same temperature as the inside wall surface temperature. It is assumed that the aluminum foil on the inside of the walls reflected about 80% of the short wavelength radiation that struck it, most of which was absorbed by the floor. Because the total exchange method is a diffuse analysis, corrections were made to the short wavelength $\overline{S_j S_k}$ factors to account for the specular walls. No corrections were made for long wavelength radiation because all of the room surfaces are at nearly the same temperature. The ducting, lumi-

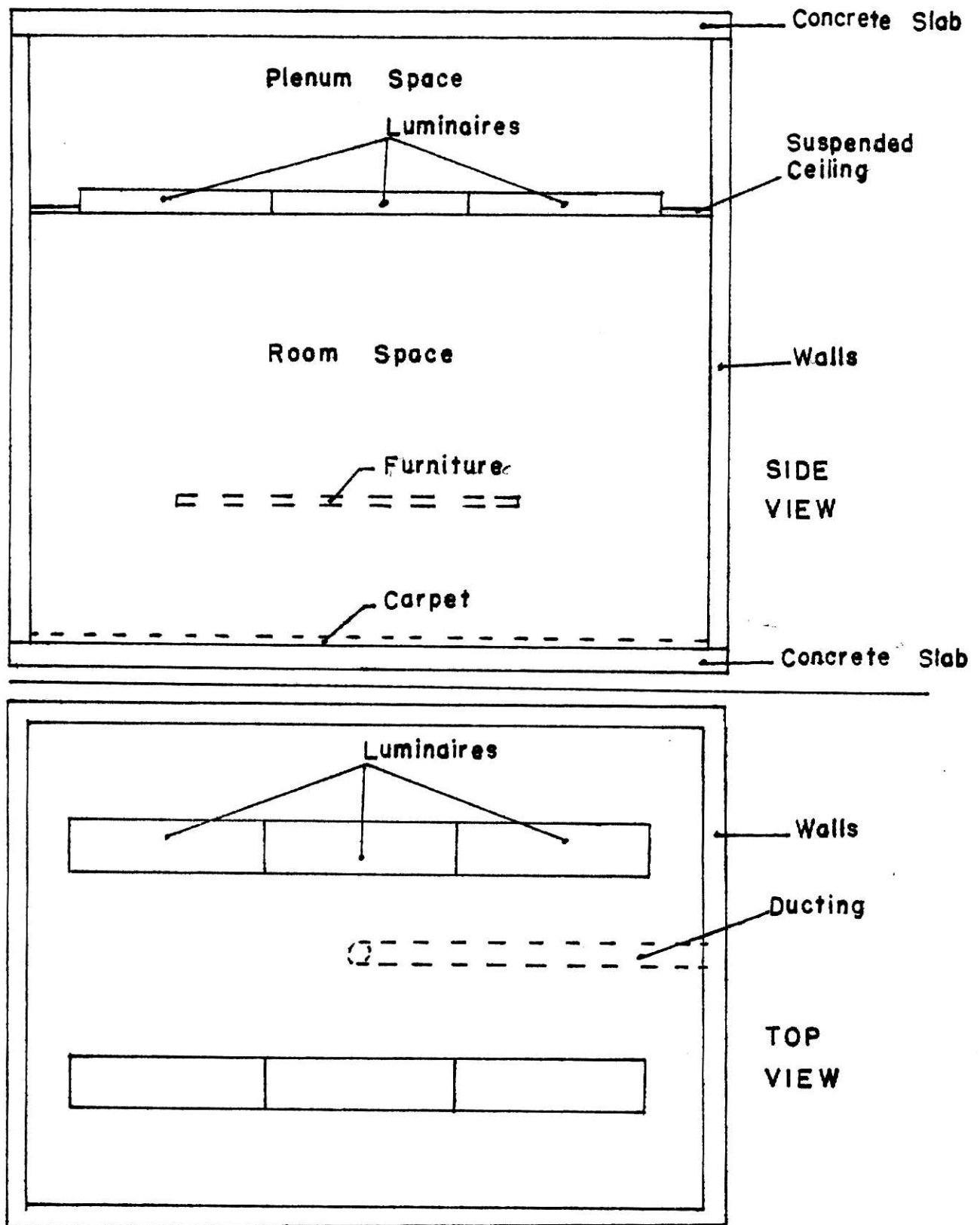


Figure 14. Model of the Experimental Calorimeter Room.

naire design, carpeting, and furniture are not described in detail by Mitalas (48), thus the modeling of these components is based only on estimates or guesses.

INITIAL MODEL COMPARISONS

After all of the input programming was done, trial runs were made to check surface temperatures, energy transfer, and other parameters to make sure the model was functioning reasonably. Three 20-hour runs were made, using an air flow rate of 100 cfm (17 cfm per luminaire) and the results for some of the important items are shown in Table 1.

Table 1.

RESULTS OF INITIAL STEADY STATE TRIAL RUNS

Parameter	Type of Ventilation		
	Static Plenum	Return Plenum	Return Lamp & Return Plenum
Lamp Tube Temperature	134°F	131°F	116°F
Lens Temperature	99°F	97°F	91°F
Housing Temperature	100°F	96°F	88°F
Lamp Air Temperature	109°F	106°F	93°F
Ballast Temperature	141°F	138°F	131°F
Heat to Room Air	85.3% of P_T	62.4% of P_T	41.3% of P_T
Heat to Return Air	0.0% of P_T	30.0% of P_T	45.7% of P_T

Comparing these values with data published by Nottage and Park (55), Lim (42), Nevins, Straub, and Ball (54), and Fisher, Roehr, and Smith (23) indicated that most of the predicted temperatures are within reasonable values and that the general

trends between different types of ventilation arrangements are also satisfactory. The predicted ballast temperatures were uniformly low, therefore the thermal conductances between the ballast node and other luminaire nodes were decreased until predicted ballast temperatures reached 150°F to 165°F at steady state. These changes caused very little differences in any of the other values, except for a slight decrease in the predicted temperatures of the nodes in contact with the ballast. The predicted fractions of lighting input energy convected to room air or exhausted from the plenum compare fairly well with values from tables in the ASHRAE Handbook 1977 Fundamentals (1).

The transient behavior of the model was examined next. Figs. 15 and 16 are the predicted transient temperature curves for several surfaces. In Fig. 15, it can be seen that the temperatures of the luminaire surfaces level off after only two hours, except for the ballast, which takes about four hours to level off. The heavier surface temperatures continue to rise even after ten hours, as shown by the plenum air temperature in Fig. 15 and by the other curves in Fig. 16 (Note the difference in temperature scale between Figs. 15 and 16). After a few hours, the luminaire surface temperatures change much less than the other surface temperatures, thus it appeared that transient calculations for these nodes could be discontinued except during four hours after the lighting is switched. A test was made to examine this idea, but a 10% lower value for q_T was obtained after 16 hours when constant luminaire temperatures were used, so the idea was abandoned.

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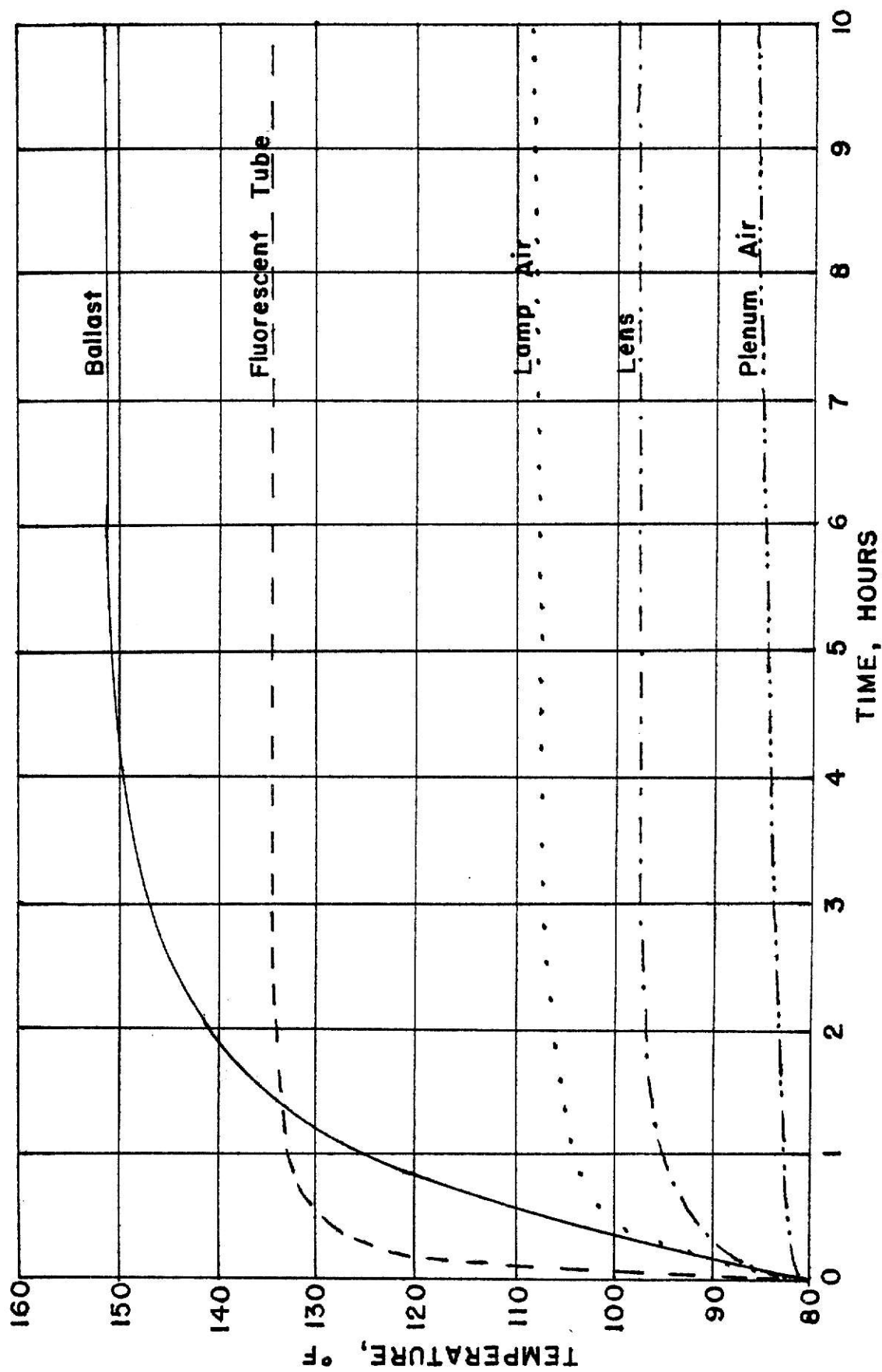


Figure 15. Predicted Transient Luminaire Surface Temperatures.

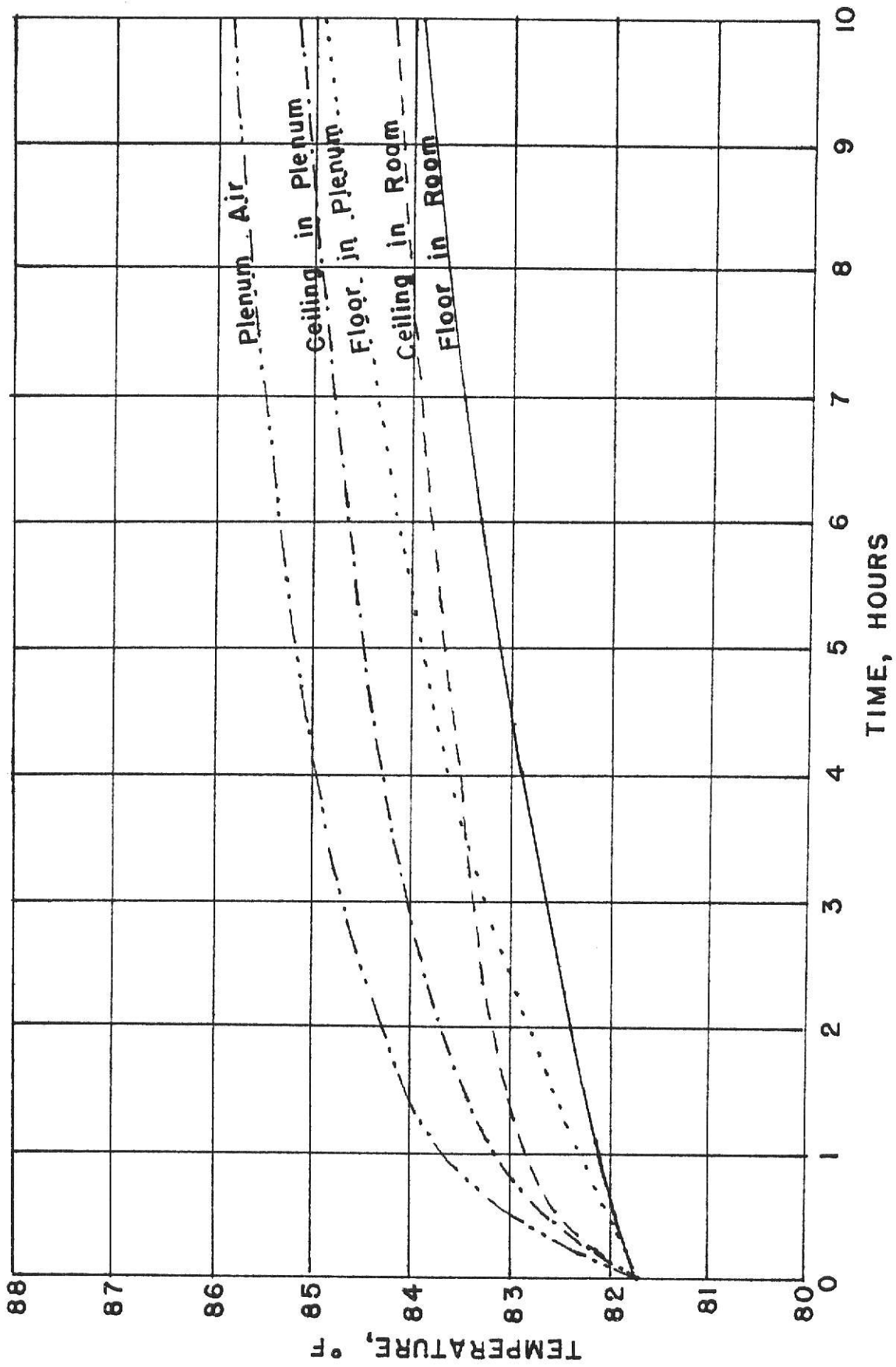


Figure 16. Predicted Transient Heavy Surface Temperatures.

The effects of changes in Δt , the standard time period, and $\Delta t'$, the lightweight surface time period, were also investigated. While changes in $\Delta t'$ were unimportant, it can be seen in Fig. 17 that using a smaller value for Δt caused q_T to change significantly during the first two hours after lighting power is changed. In subsequent programming, half sized values of Δt were used following lighting changes.

More detailed test runs were made to determine the variations that could occur if changes were made in ventilation arrangements and air flow rates, if different values were assumed for the convection coefficients, and if carpeting or furniture were present or not. Three basic ventilation arrangements were chosen:

Case A - Return through a side wall grill

Case B - Return through the ceiling plenum

Case C - Return through the luminaire then the plenum

In all three cases the supply air was from a ceiling diffuser. The standard flow rate was 100 cfm and the standard convection coefficients were obtained from the corresponding equations specified earlier in this thesis. For the standard runs, no carpet or furniture was modeled. The room air temperature was set at 81.7°F because this was what two graphs of the test results from Mitalas (48) showed. The results of computer runs on these three standard cases are shown in Fig. 18. The supply air temperature is dependent upon the amount of heat from lighting that enters the room air before it is ventilated away, and for a time period Δt , this is given by q_R . The total cooling or heat-

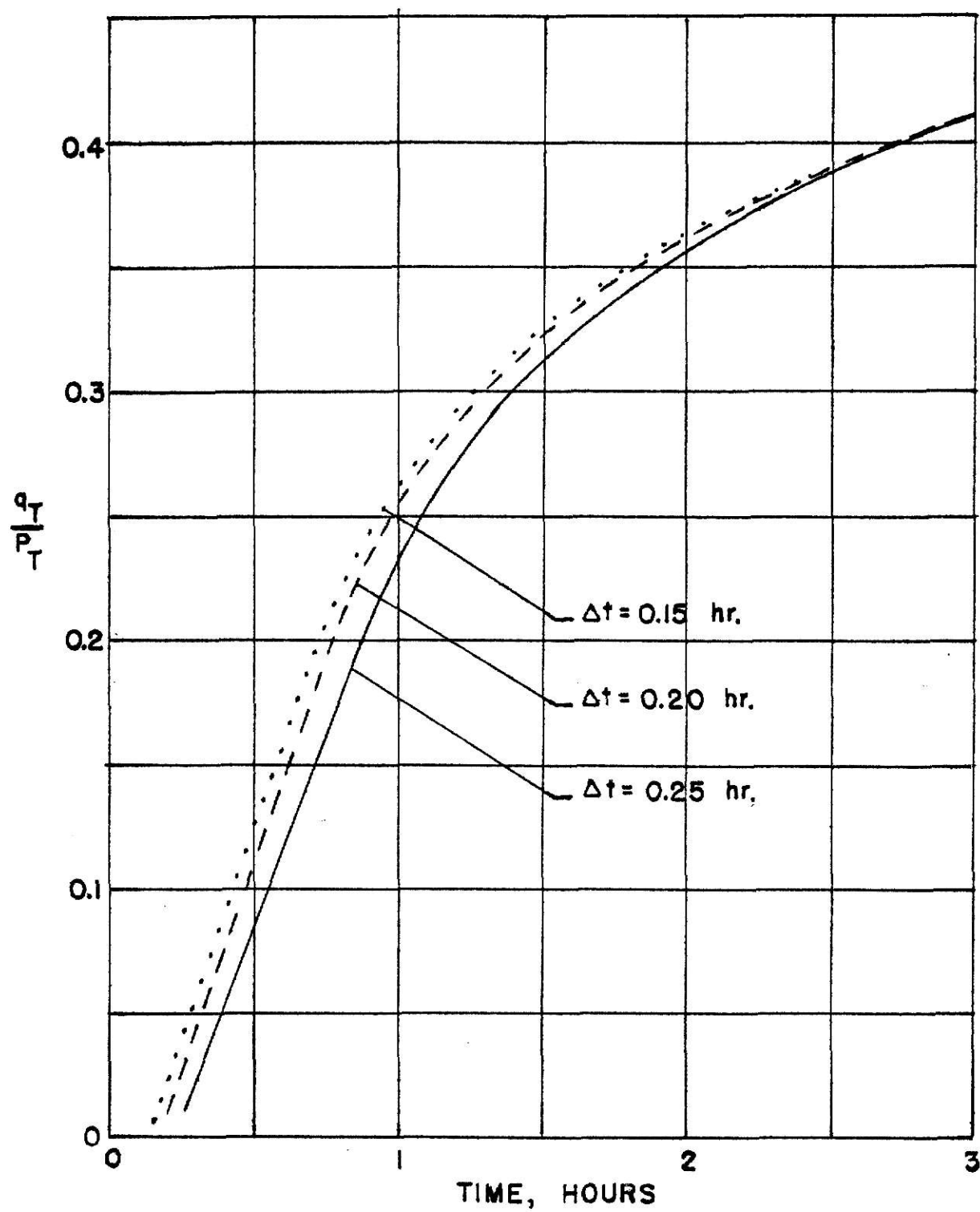


Figure 17. The Effects of Time Period Size on Model Prediction.

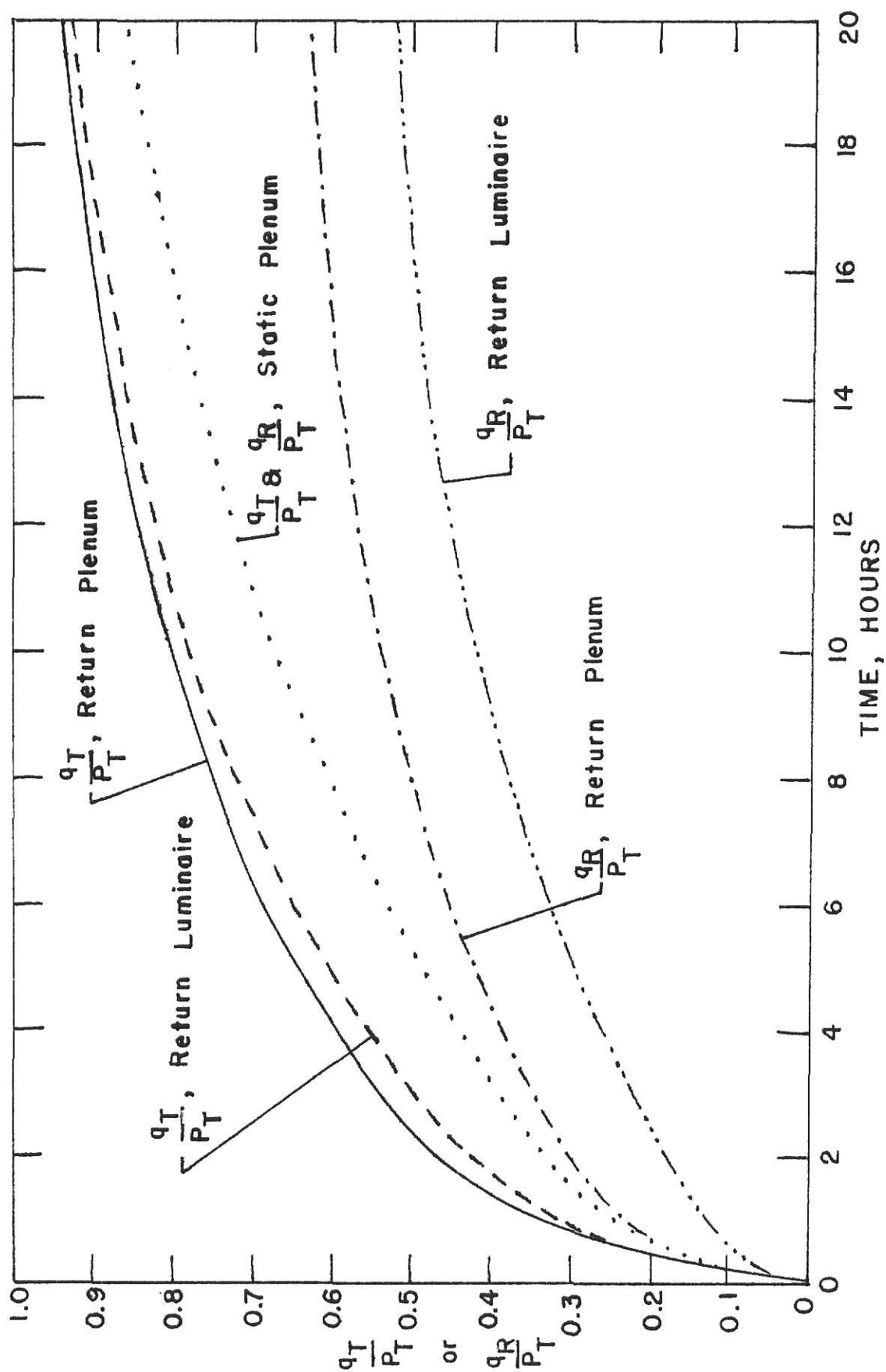


Figure 18. Model Prediction of q_R/P_T and q_T/P_T for Three Standard Cases.

ing load depends upon the total energy from lighting that is picked up, both before and after the air leaves the room, which is called q_T . The total energy used by the luminaires during the same time period is called P_T . The terms q_R/P_T and q_T/P_T are thus the fraction of the lighting energy used that enters the room space and the fraction that must be absorbed by the air conditioning system, respectively. In Cases B and C, where the return air is vented through the plenum, q_T/P_T is larger than in Case A, where a static plenum is modeled. But q_R/P_T is larger for Case A than the other two. Case C has the lowest q_R/P_T value because much of the energy is convected to return air in the luminaire before it can enter the room space.

The effects of different convection coefficients and ventilation flow rates were investigated for each of these three cases. The room, plenum, and luminaire convection coefficients were multiplied by 0.5 and 1.5, for each space separately. The ventilation flow rate was changed to 50 and 200 cfm in other computer runs. Table 2 is a listing of the percent variations from standard values that were produced when each of these parameters were changed. The size of these deviations may be pictured by comparing them against the $\pm 5\%$ variation curves in Fig. 19.

The largest variations in all three cases were caused by changing the room convection coefficients. Case A was most affected, as q_T comes only from room surfaces in this type of model. Case C was least affected because much of q_T is picked up as return air passes through the luminaire. Changes in the ple-

TABLE 2
% CHANGES IN q_T/P_T FOR TRIAL RUNS

Room Condition Variations	Time hrs.	% Changes from Standard Values		
		Case A	Case B	Case C
Ventilation flow rate is 0.5 times standard	5	+2%	-11%	-13%
	10	+1%	-7%	-8%
	20	0	-3%	-3%
Ventilation flow rate is 2.0 times standard	5	-2%	+14%	+18%
	10	-1%	+8%	+9%
	20	0	+3%	+3%
Room convection factors are 0.5 times standard	5	-27%	-15%	-12%
	10	-22%	-12%	-11%
	20	-15%	-6%	-6%
Room convection factors are 1.5 times standard	5	+18%	+9%	+8%
	10	+13%	+6%	+6%
	20	+7%	+2%	+3%
Plenum convection factors are 0.5 times standard	5	+4%	0	+7%
	10	+2%	-1%	+3%
	20	+1%	-1%	+1%
Plenum convection factors are 1.5 times standard	5	-1%	-1%	-3%
	10	0	0	-1%
	20	0	0	0
Lamp convection factors are 0.5 times standard	5	0	0	0
	10	0	0	0
	20	0	0	0
Lamp convection factors are 1.5 times standard	5	+1%	0	0
	10	+1%	0	0
	20	+1%	0	0

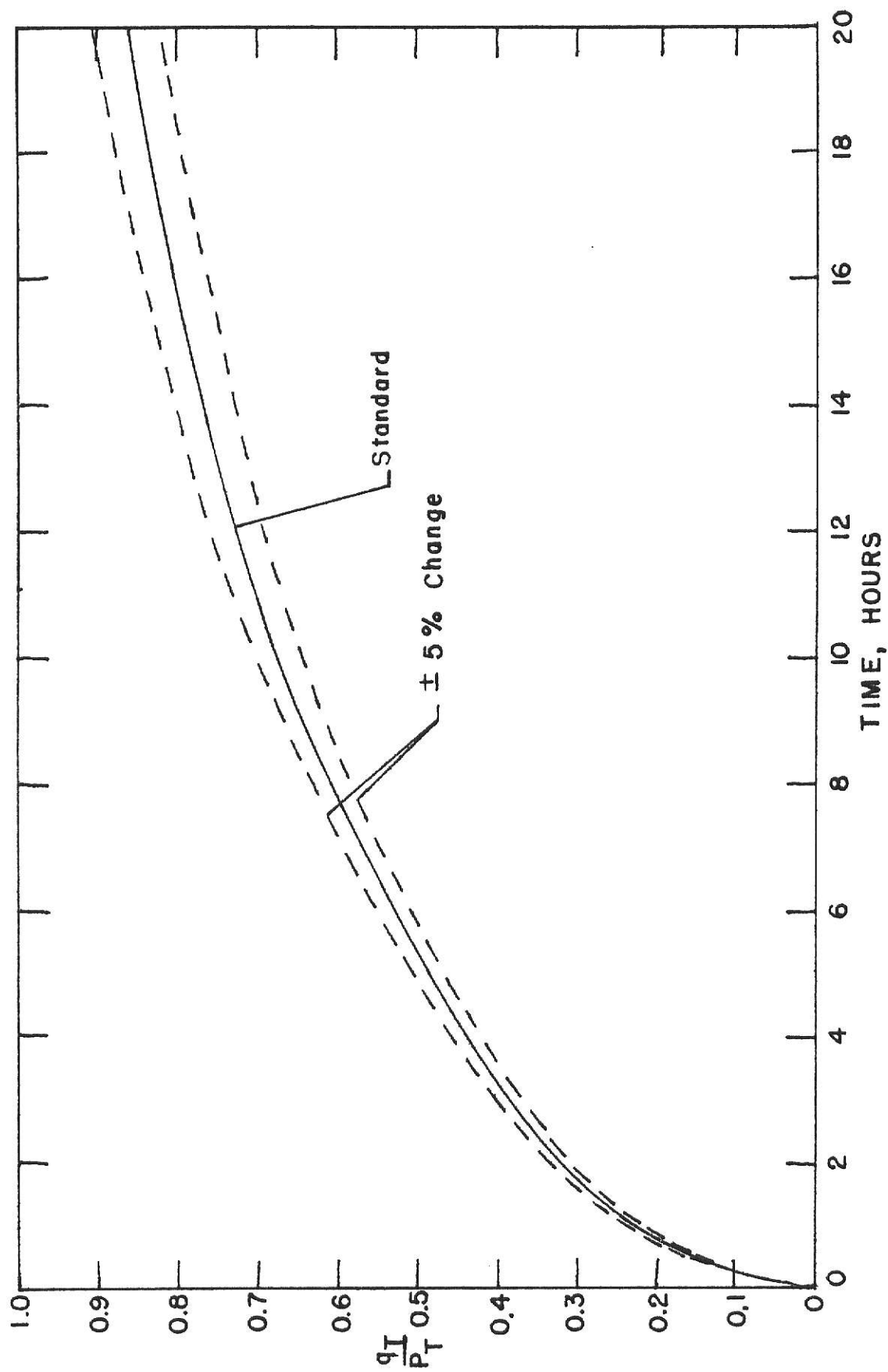


Figure 19. An Example of $\pm 5\%$ Change in q_T/P_T Compared to the Standard Curve - Case A
- Static Plenum.

num or luminaire convection coefficients resulted in only small differences. This might seem suprising, especially in Cases B and C, where the convective transfer to return air is a large part of q_T . Examining the mean temperature method for calculating convective transfer to air masses shows why the heat transfer is somewhat independent of convection coefficients when the air is flowing through a plenum or luminaire. As the air flows through such a volume, it is heated and its temperature rises towards an equilibrium temperature, following an exponential decay path. If the thermal capacitance of the air mass is small compared to the amount of surface area it contacts, then the temperature of the air mass will quickly approach the equilibrium value, whether the convection coefficients are large or not. Because the amount of heat transferred to the air mass is proportional to the final temperature of the air mass, and this temperature is close to the equilibrium temperature independent of the convection coefficients, then the heat transfer will be nearly the same. Changes in the flow rate have a much greater effect because this also changes the heat capacitance of the air mass. The analysis in Appendix C may help some in understanding this process.

The predicted variations in q_T/P_T due to changes in the ventilation flow rate were much larger when the plenum of luminaires were vented. In the static plenum condition, the only changes resulting from higher ventilation rates are slight increases in room convection coefficients. But in Cases B and C, the ventilation rate is related to the mean plenum temperature.

As the plenum temperature is decreased by higher ventilation rates, more energy is convected to the plenum air and removed, increasing q_T .

The changes in luminaire characteristics after 20 hours as the return flow rate through vented luminaires varies agrees well with previously published steady state experimental results. The predicted fluorescent tube temperatures shown in Fig. 20 are very close to the average values obtained by Nevins, Straub, and Ball (54) and also follow the data of Nottage and Park (55). The model predictions of the exhaust air temperature in Fig. 20 and the input to lighting predictions in Fig. 21 were used to calculate the amount of heat convected to return air, also shown in Fig. 21, along with the range of values Nevins, Straub, and Ball (54) found by testing. Although the model prediction for heat to return matches the predicted range closely, it should be noted that Nevins, Straub, and Ball (54) measured the temperature difference between the air entering the luminaire and the air leaving their box calorimeter. Even though the calorimeter was well insulated, a small amount of energy passed through the walls of the calorimeter. Also, the calorimeter temperature at low flow rates was higher than at high flow rates, which gives a larger temperature difference through the calorimeter walls at low flow rates. Both of these factors indicate that the model predictions of heat transfer to return air is even better than shown in Fig. 21.

The transient changes in q_T/P_T that result when furnishings are considered in the model are shown in Figs. 22, 23, and 24.

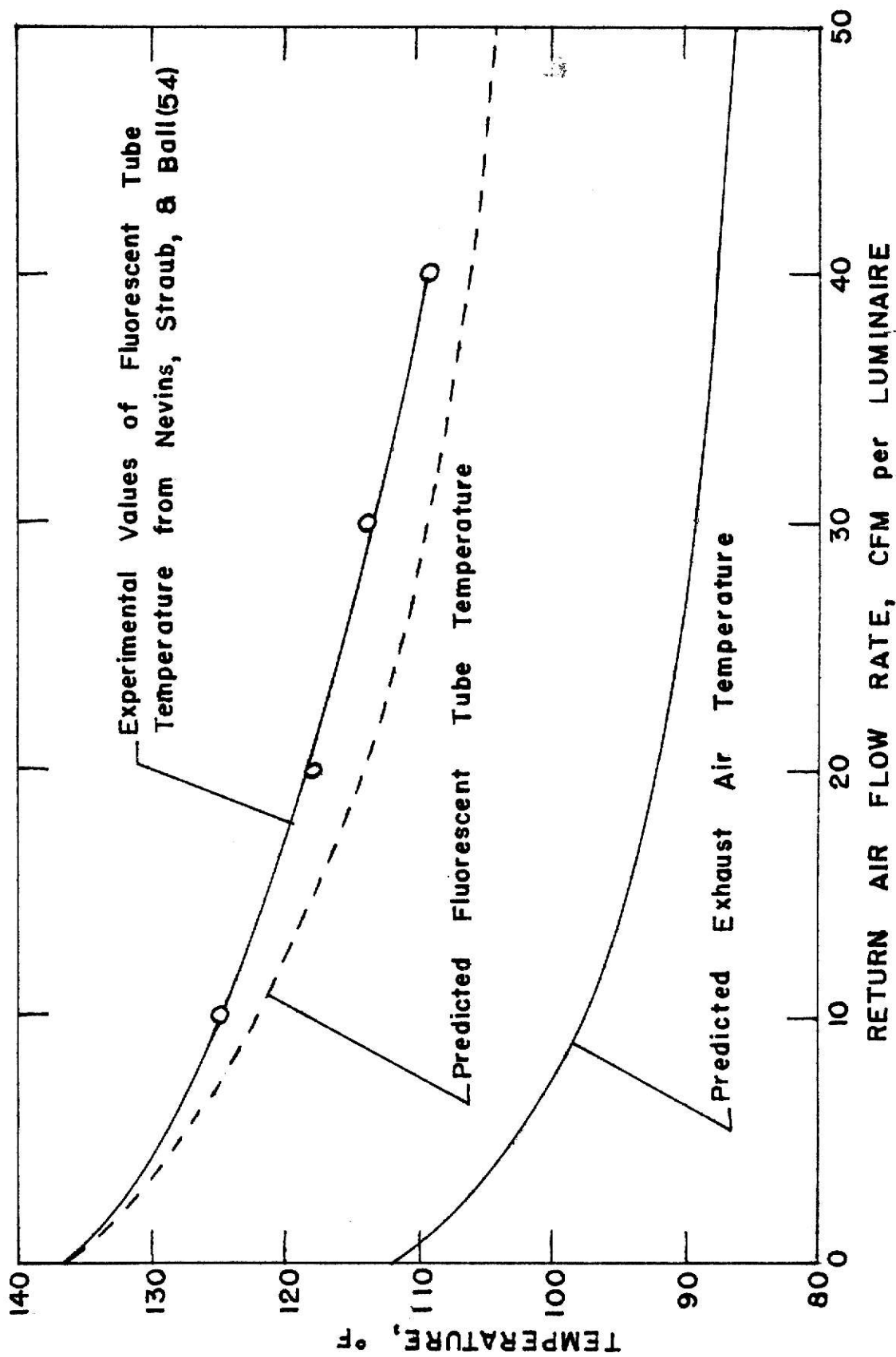


Figure 20. The Effects of Return Air Flow Rate on Luminaire Temperatures.

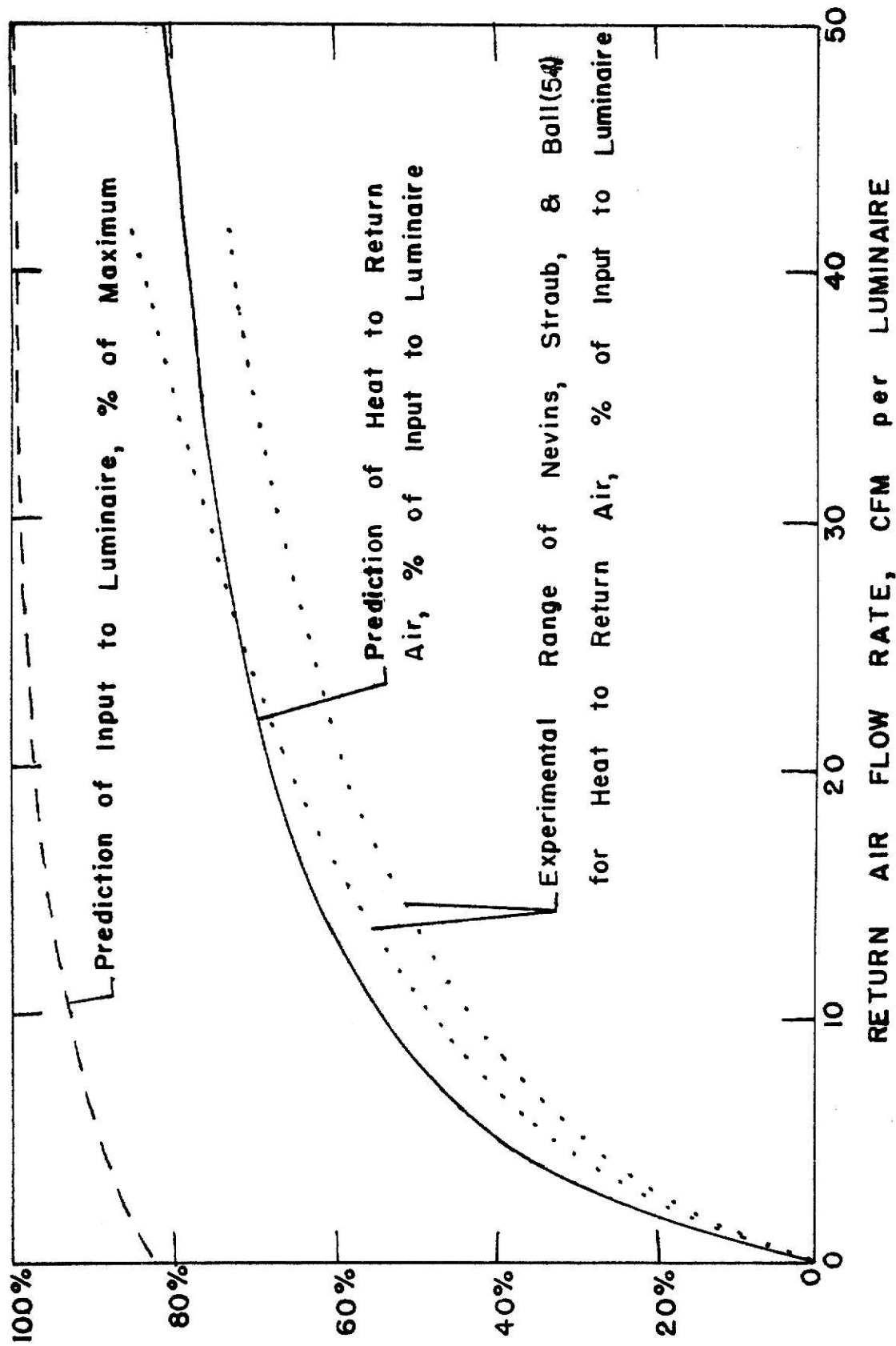


Figure 21. The Effects of Return Air Flow Rate on Energy Transfer in the Luminaire.

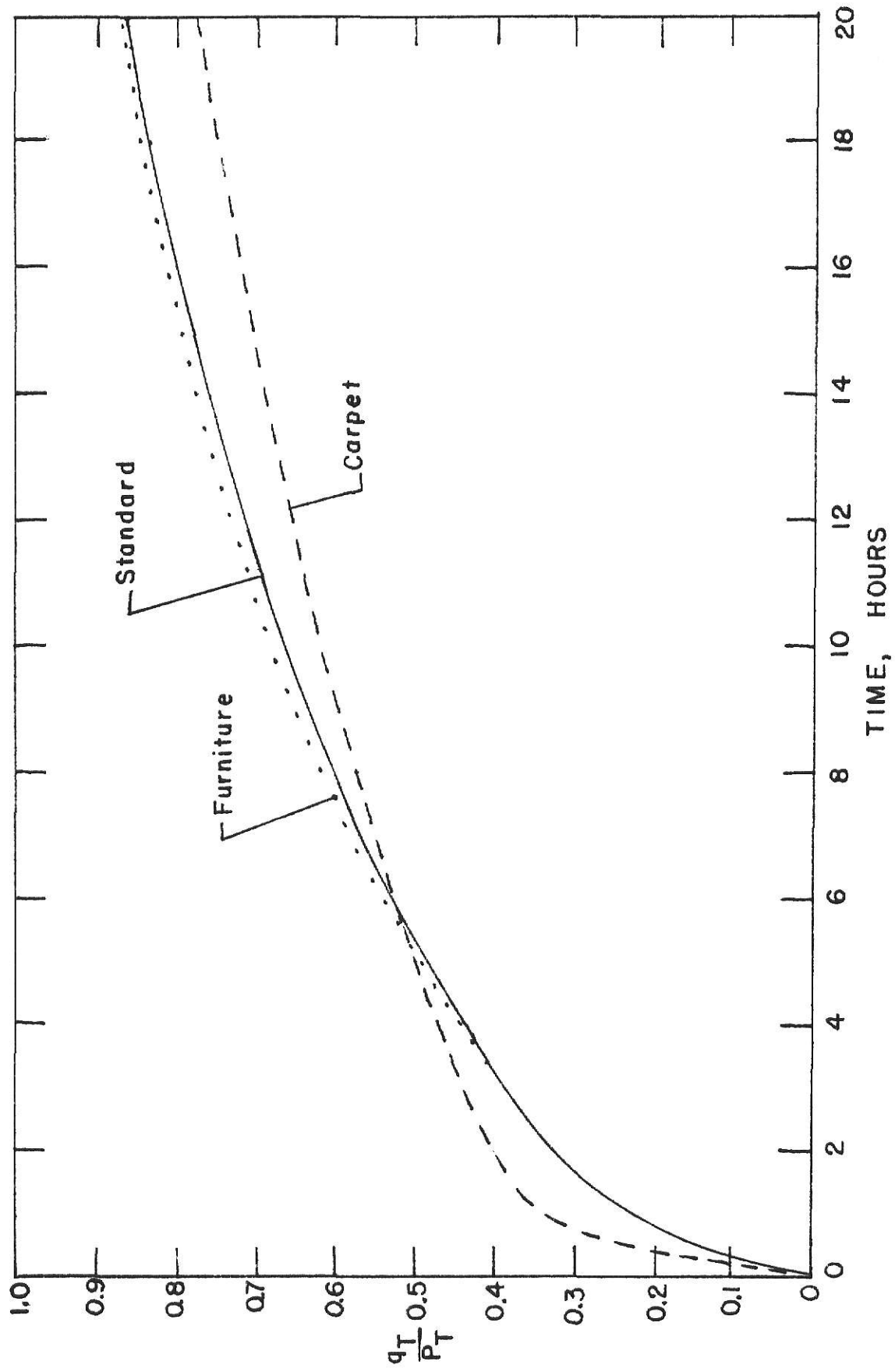


Figure 22. The Effects of Carpet and Furniture on Model Predictions - Case A - Static Plenum.

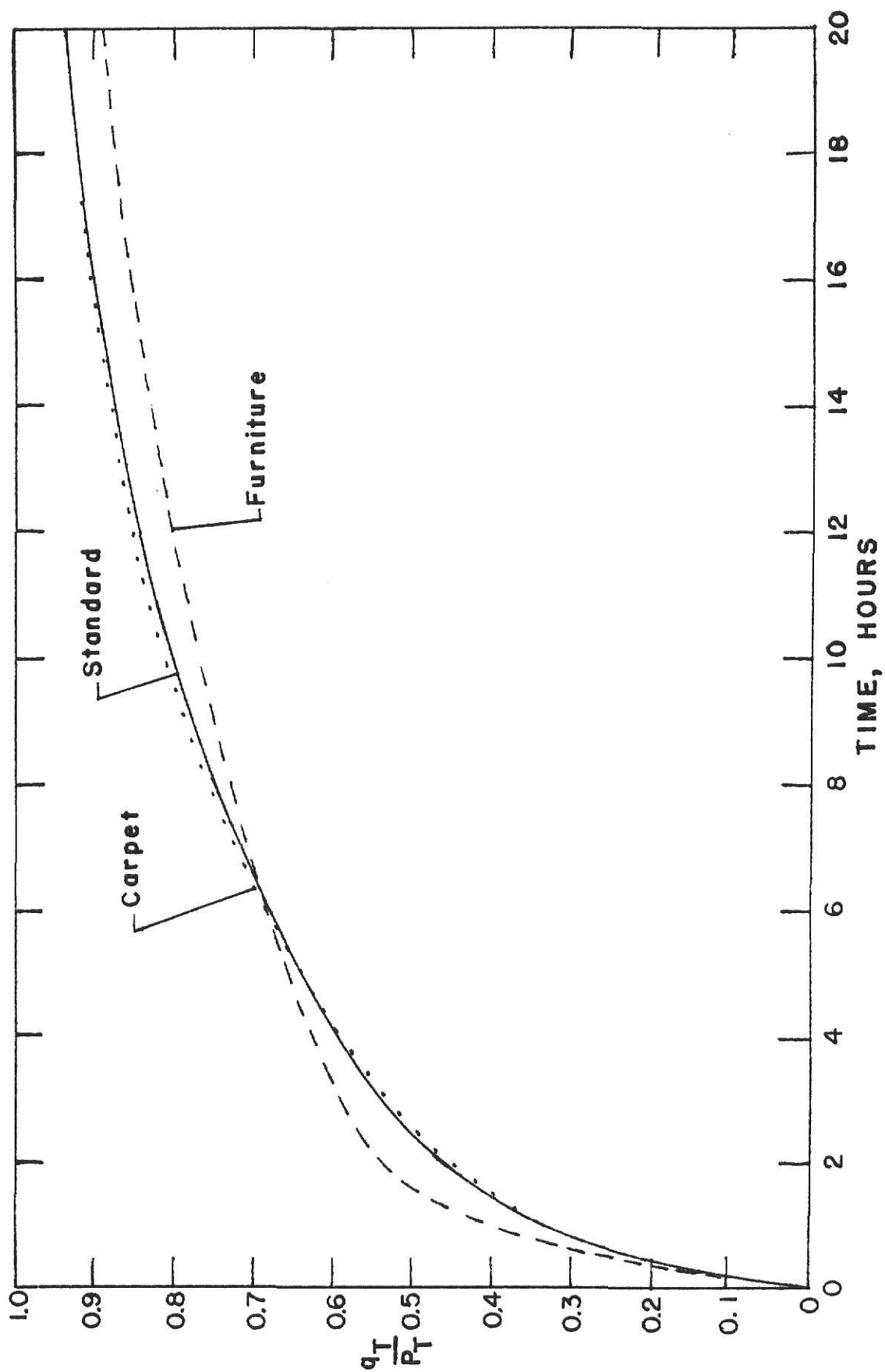


Figure 23. The Effects of Carpet and Furniture of Model Predictions - Case B - Return Plenum.

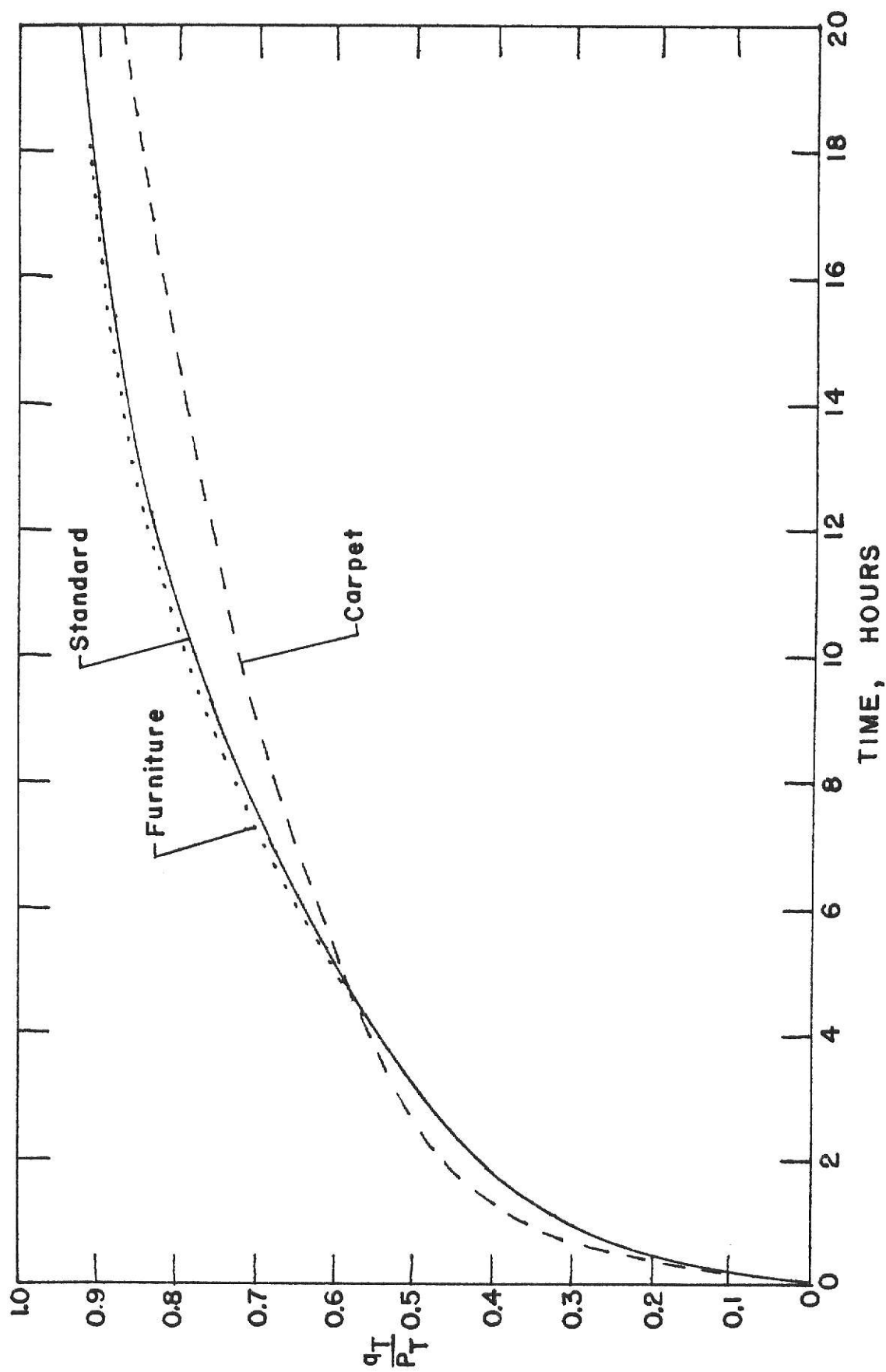


Figure 24. The Effects of Carpet and Furniture on Model Predictions - Case C - Return Luminaires and Return Plenum.

The general trends are similar for all three ventilation models. The addition of a carpet layer to the model causes q_T/P_T to be higher during the first five hours, then become less than the standard curves for the rest of the test. The early increase in q_T/P_T is caused by the carpet layer heating up and convecting heat to the room quicker than the bare slab. But after five hours, energy starts reaching the room by conduction from the plenum below. However, this energy flow is reduced when carpeting is present because of the insulating qualities of carpeting. Case B is not affected as strongly as the other two because its plenum temperature is slightly lower. Much smaller changes resulted when furniture was modeled, but because Kelles (34) determined that furniture might have some appreciable effect, furniture considerations have been left in the model.

COMPARISON OF MODEL PREDICTIONS WITH MITALAS' TRANSIENT TESTS

Because the model predictions seemed reasonable, the next step was to compare the transient experimental results of Mitalas (48) against model predictions under similar conditions. The experimental calorimeter room tests were done by letting the room come to steady state, then turning the lighting on until the room again reached steady state, when the lighting was switched off. Heat transfer measurements were taken from a few hours before the lights were turned on until two days after the lights were turned off. The actual results were not presented in the paper. Instead, Mitalas (48) approximated the results with Eq. (2) which is repeated here:

$$q_T(t) = a_1 P_T(t-\Delta t) + a_2 P_T(t-2\Delta t) + b_1 q_T(t-\Delta t) \quad (2)$$

The different values of a_1 , a_2 , and b_1 and the corresponding test conditions are listed in Table 3, taken from Mitalas (48). The main problem with comparing the model predictions against the values obtained from Eq. (2) is that this equation is a truncated series, which results in overestimated values for $q_T(t)$ during the first few hours after the lighting is switched. This deviation is shown in Fig. 25, which is redrawn from a graph in Mitalas (48).

Mitalas (48) also stated that he produced values for a_1 accurate to ± 0.03 and for b_1 accurate to ± 0.005 . For the same experiment, Test 18, the values for q_T/P_T were calculated using a_1 and b_1 with their error limits and the adjusted equations are bounded by the dashed curves in Fig. 26. The differences shown in this graph should be considered when comparing the predicted values to other graphs of these fitted equations. While the magnitude of q_T/P_T differs a lot, the slopes of the three curves are nearly the same. Thus, the model predictions may be accurate, even if they are not extremely close to the fitted equations, as long as the curves have the same slope after several hours.

All but two of the room configurations listed in Table 3 were simulated using the computer model. Tests 9, 10, and 11 and Tests 20, 21, and 22 were not modeled because the supply air entered the room through side slots on the luminaire and the actual configuration is not described by Mitalas (48). The furniture was guessed to be a wood table and chair and two empty fil-

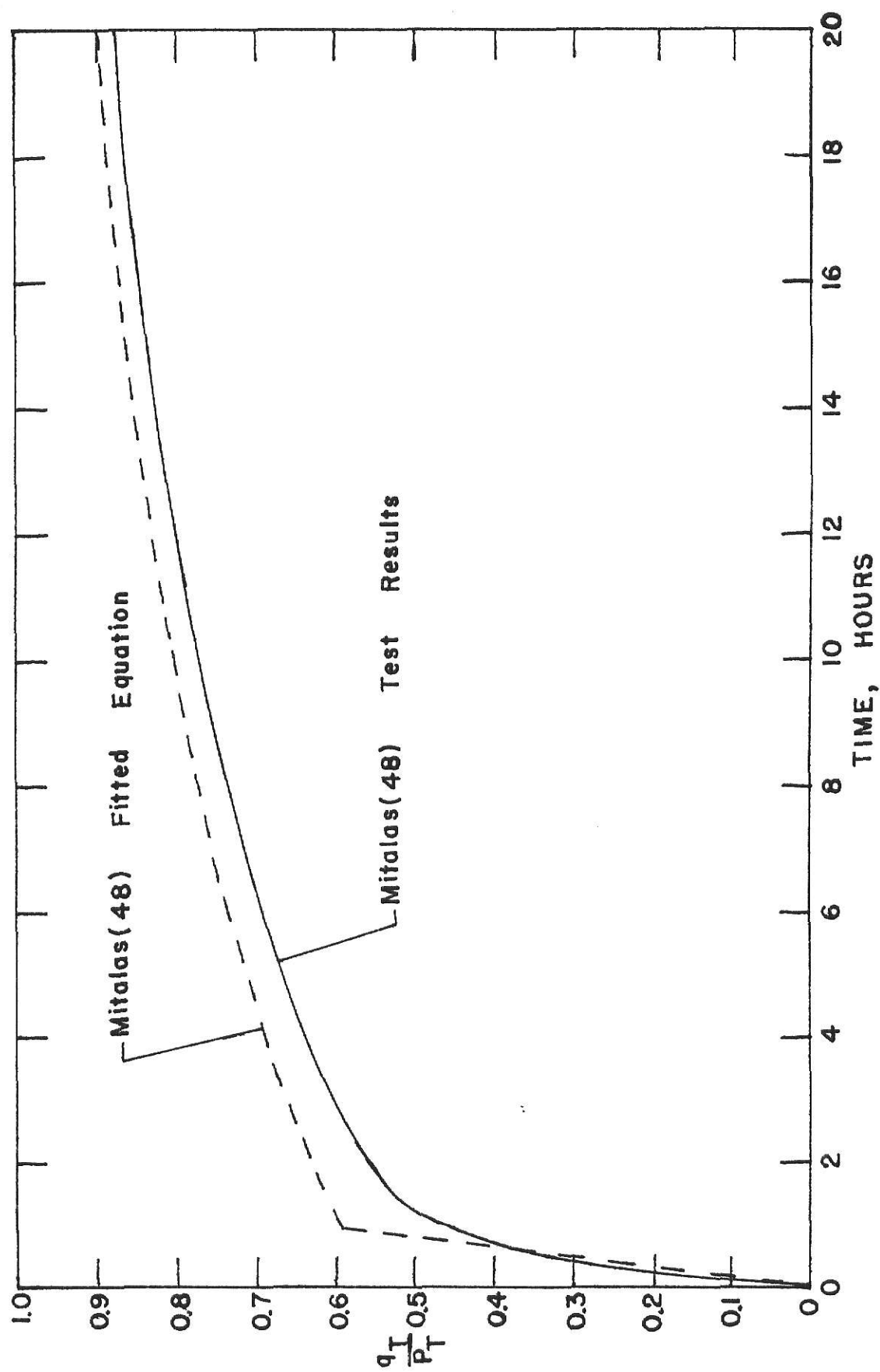


Figure 25. Comparison of q_m/P_T from Mitalas (48) Fitted Equation and Mitalas (48) Experimental Results - Test 18.

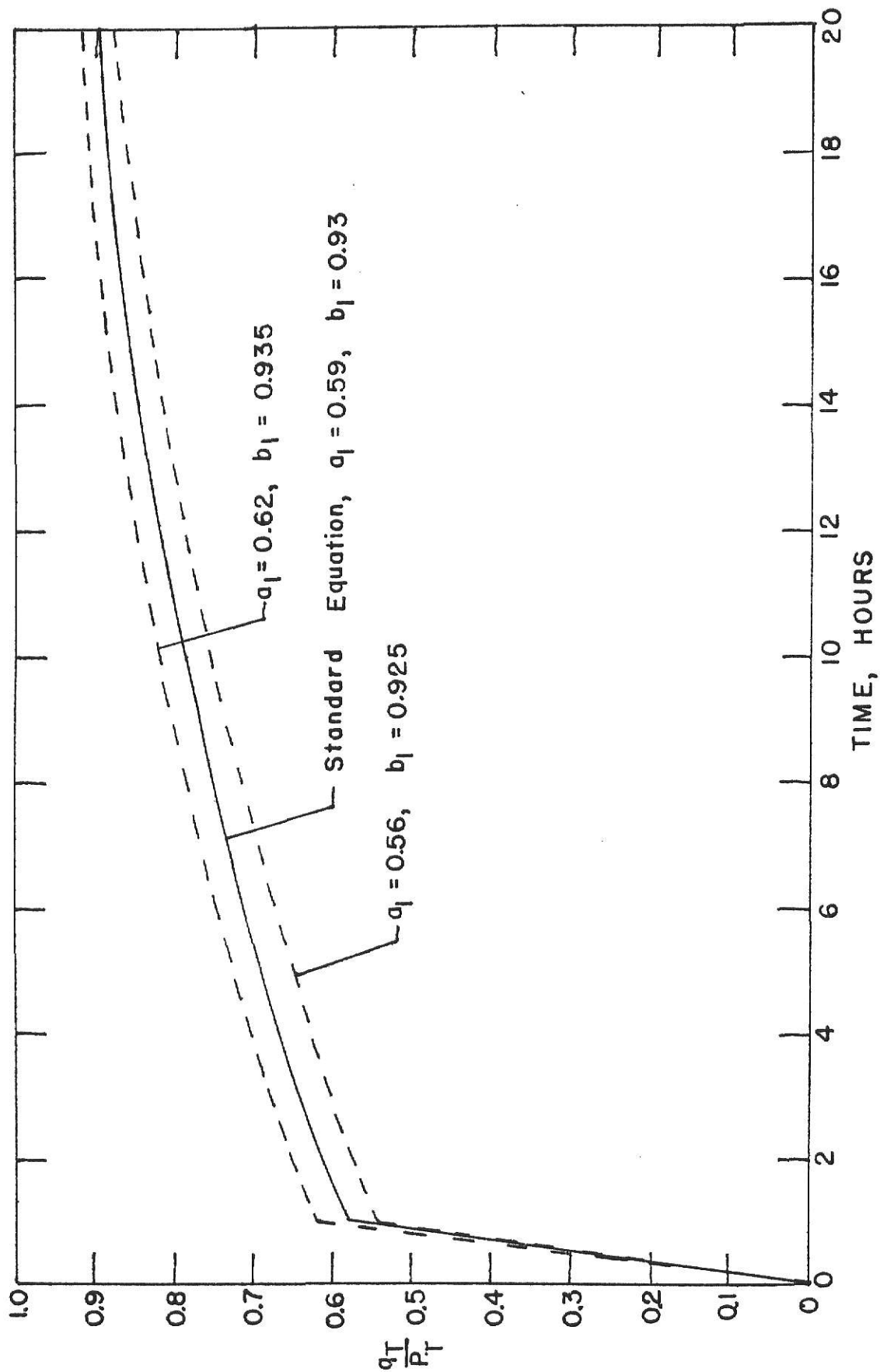


Figure 26. Limits of Accuracy of Mitalas (48) Fitted Equation - Test 18.

ing cabinets. The carpet properties were guessed to be those listed for carpet and rubber pad in Chapter 22 of the ASHRAE Handbook 1977 Fundamentals (1). The results of the computer predictions of q_T/P_T for each of the other tests, along with the corresponding equations Mitalas (48) fitted to the experimental results, are graphed in Figs. 27-36. The model predictions are well below the fitted equations during the first few hours, but as the time increases, the differences between the pairs of curves lessens, which follows the trend of the experimental results as shown in Fig. 25. There are two model predictions that deviate from this pattern, Tests 7 and 8 in Fig. 31 and Tests 14-17 in Fig. 33. The reason is that the initial estimate of the carpet conductivity was too small. This was adjusted from 0.81 to 1.60 Btu/hr-ft²-°R and the results are shown in Figs. 37, 38, and 39. The curves in Figs. 37 and 39 are still not approaching the fitted equations of Mitalas (48) as they should, although increasing the carpet conductivity more would bring the predictions closer to these fitted equations. But the prediction shown in Fig. 38 is slightly high, and any further increase in carpet conductivity will increase this error, though not as much as the model prediction for Tests 14-17 will be improved. Fig. 35 is a comparison of model prediction and the fitted equation of Mitalas (48) when a 2-inch slab of styrofoam was placed under the carpet. The properties of the styrofoam can be estimated more accurately than the carpet properties from the given information, yet the styrofoam has little mass and can be modeled as very thick carpet. In this case, the model prediction is

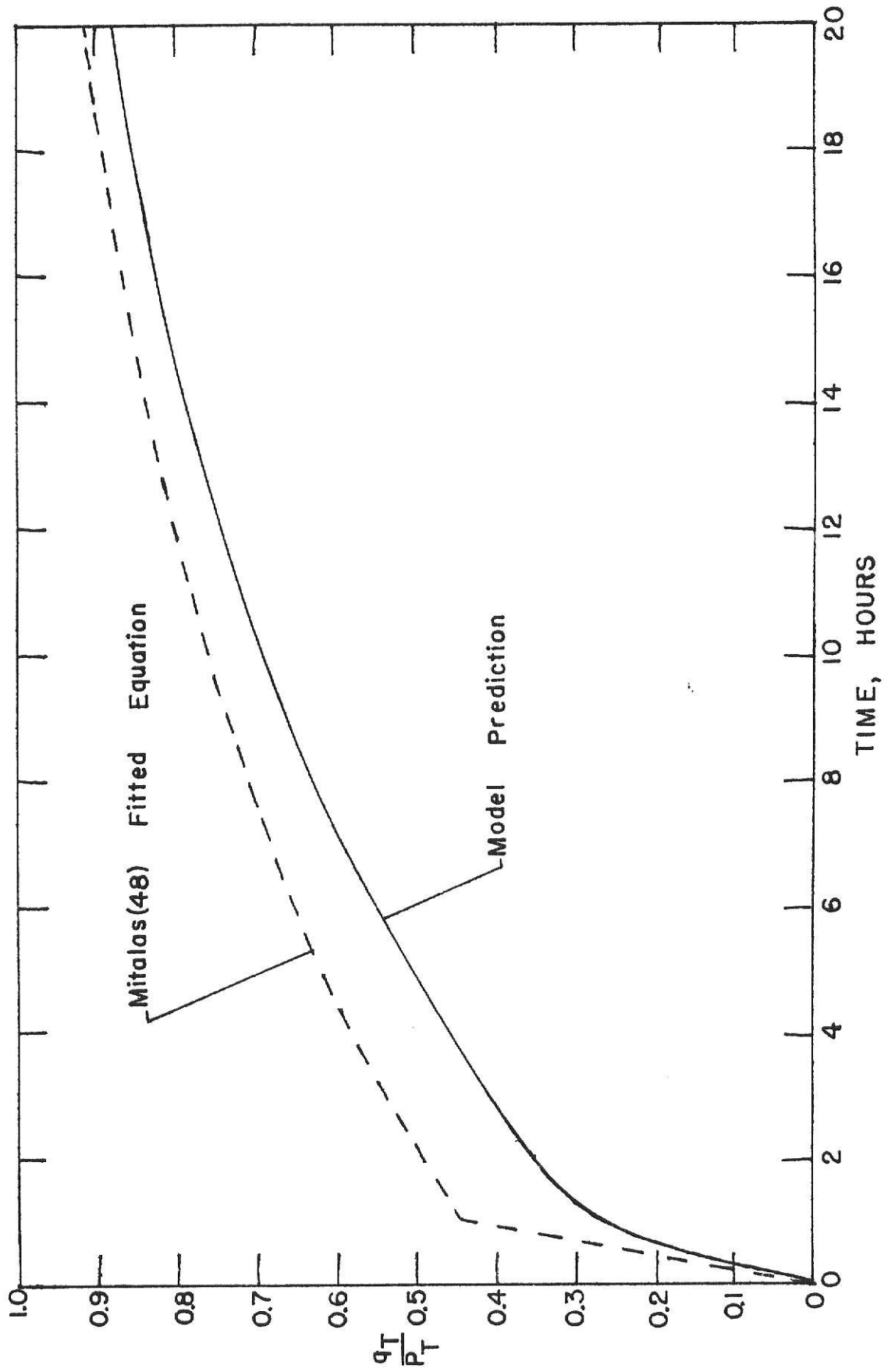


Figure 27. Comparison of q_T/P_T from Model Prediction and Mitalas (48) Fitted Equation - Test 1.

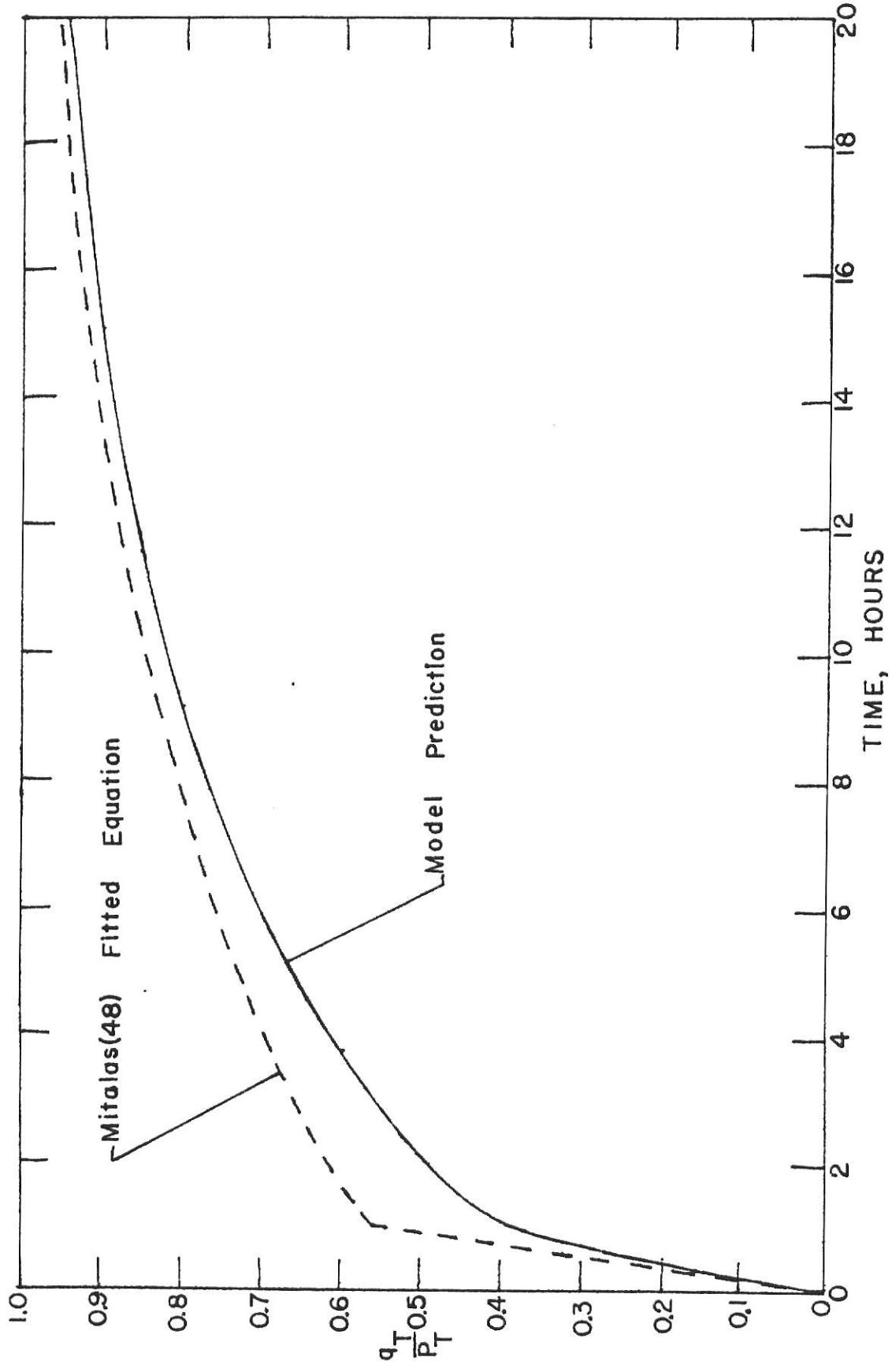


Figure 28. Comparison of q_T/P_T from Model Prediction and Mitalas (48) Fitted Equation - Tests 2 and 3.

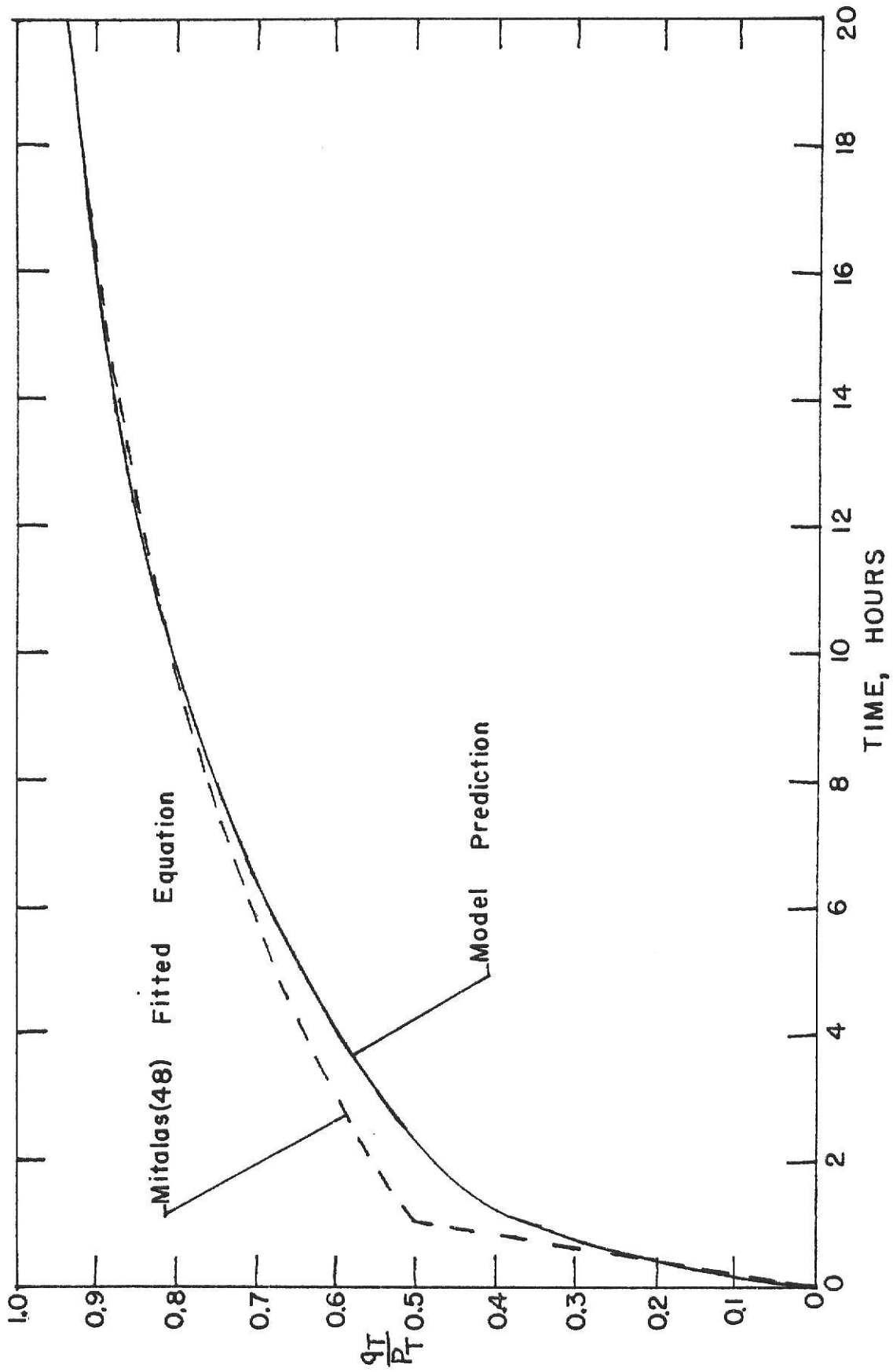


Figure 29. Comparison of q_T/P_T from Model Prediction and Mitalas (48) Fitted Equation - Test 4.

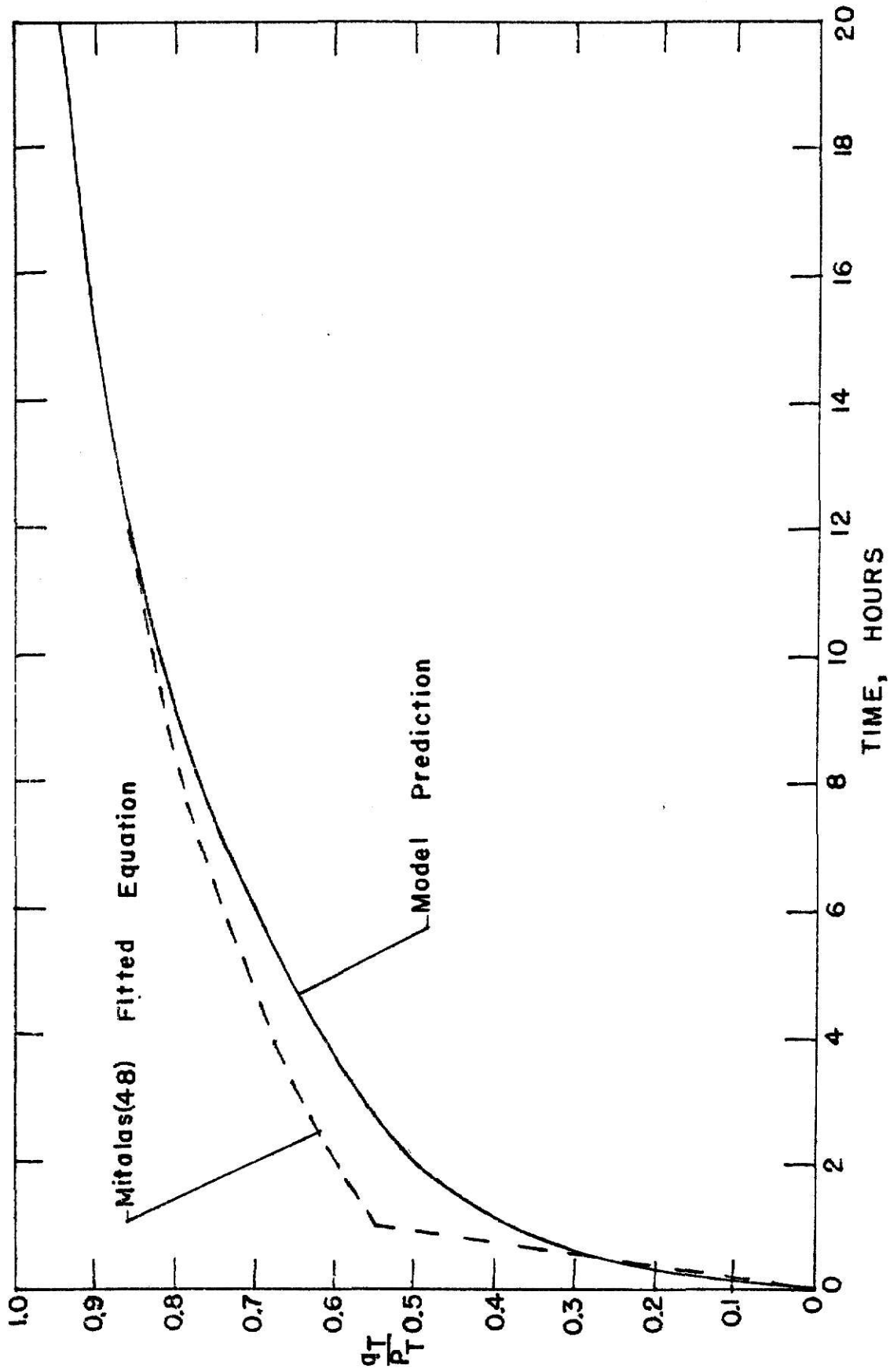


Figure 30. Comparison of q_T/P_T from Model Prediction and Mitalas (48) Fitted Equation - Tests 5 and 6.

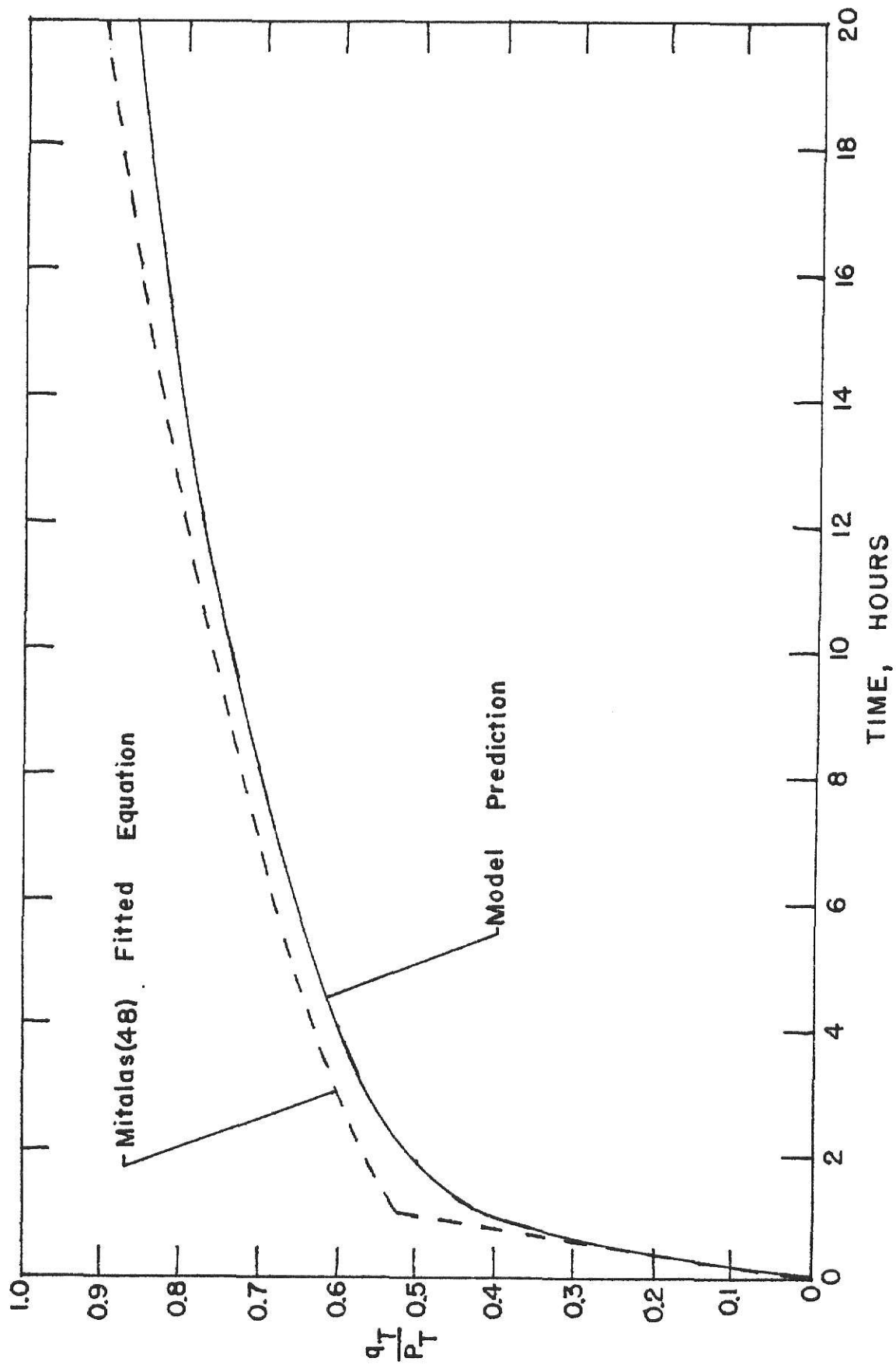


Figure 31. Comparison of q_T/P_T from Model Prediction and Mitalas (48) Fitted Equation - Tests 7 and 8.

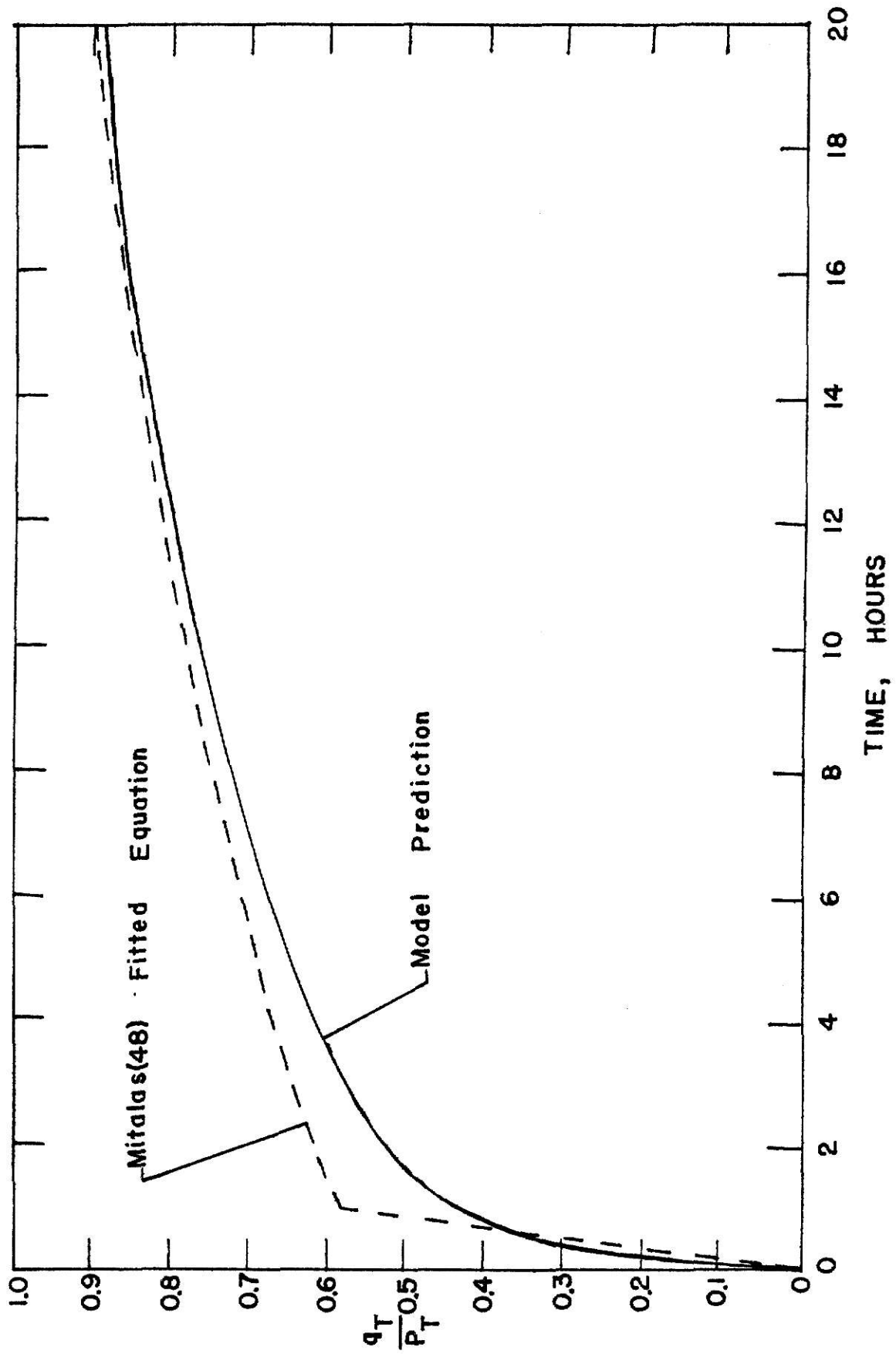


Figure 32. Comparison of q_T/P_T from Model Prediction and Mitalas (48) Fitted Equation - Tests 12 and 13.

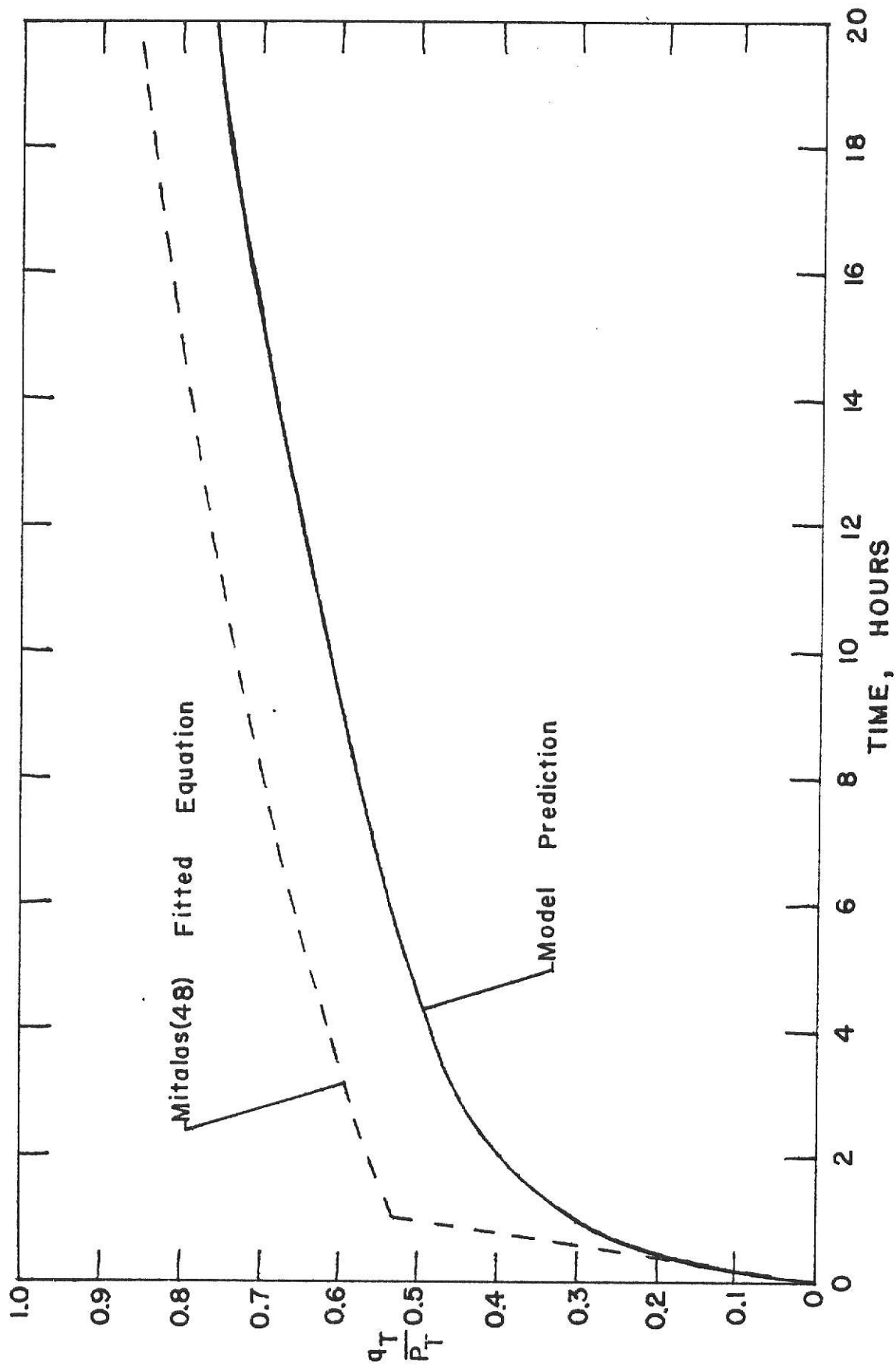


Figure 33. Comparison of q_T / P_T from Model Prediction and Mitalas (48) Fitted Equation - Tests 14, 15, 16, and 17.

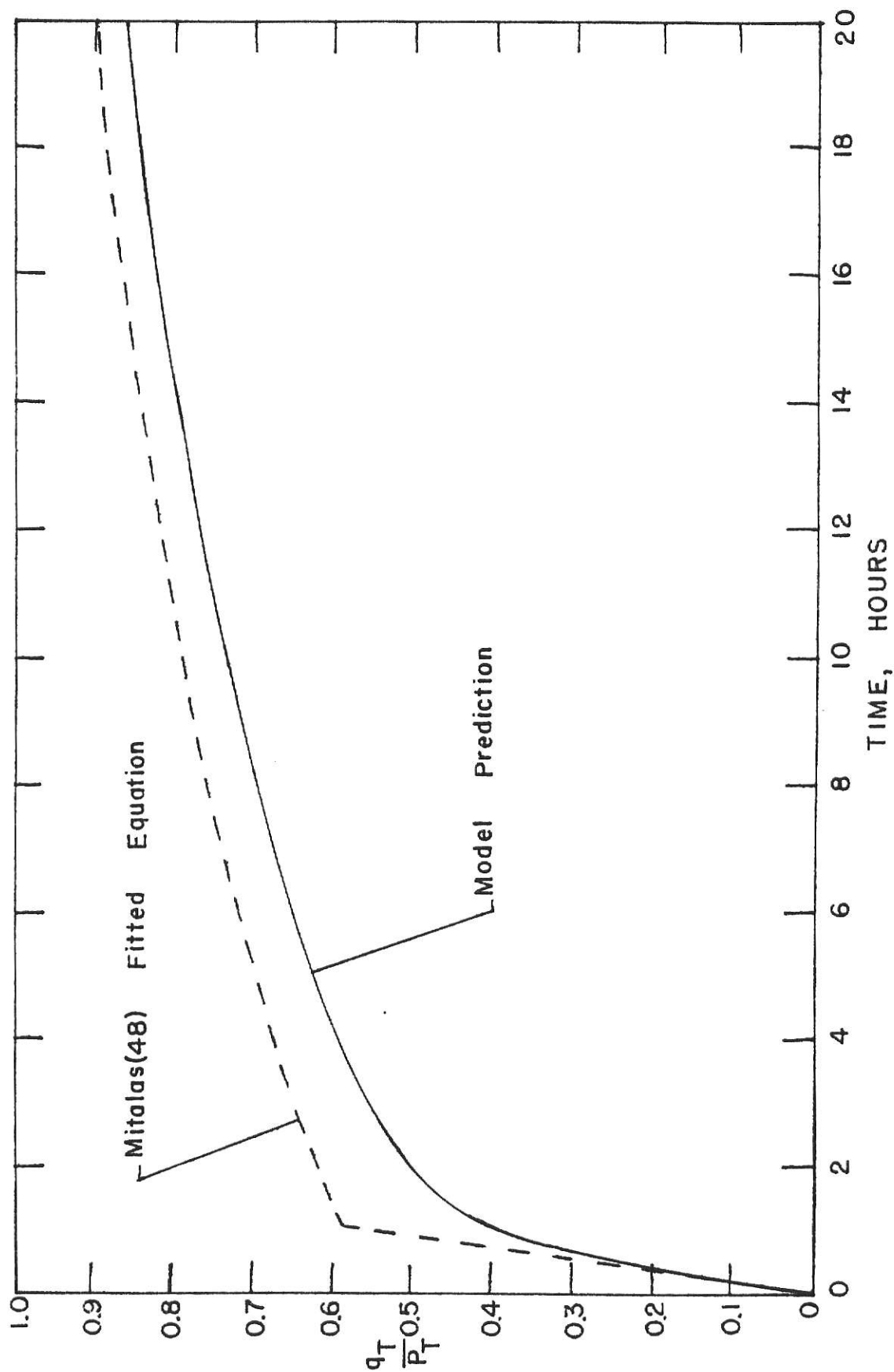


Figure 34. Comparison of q_T/P_T from Model Prediction and Mitalas (48) Fitted Equation - Test 18.

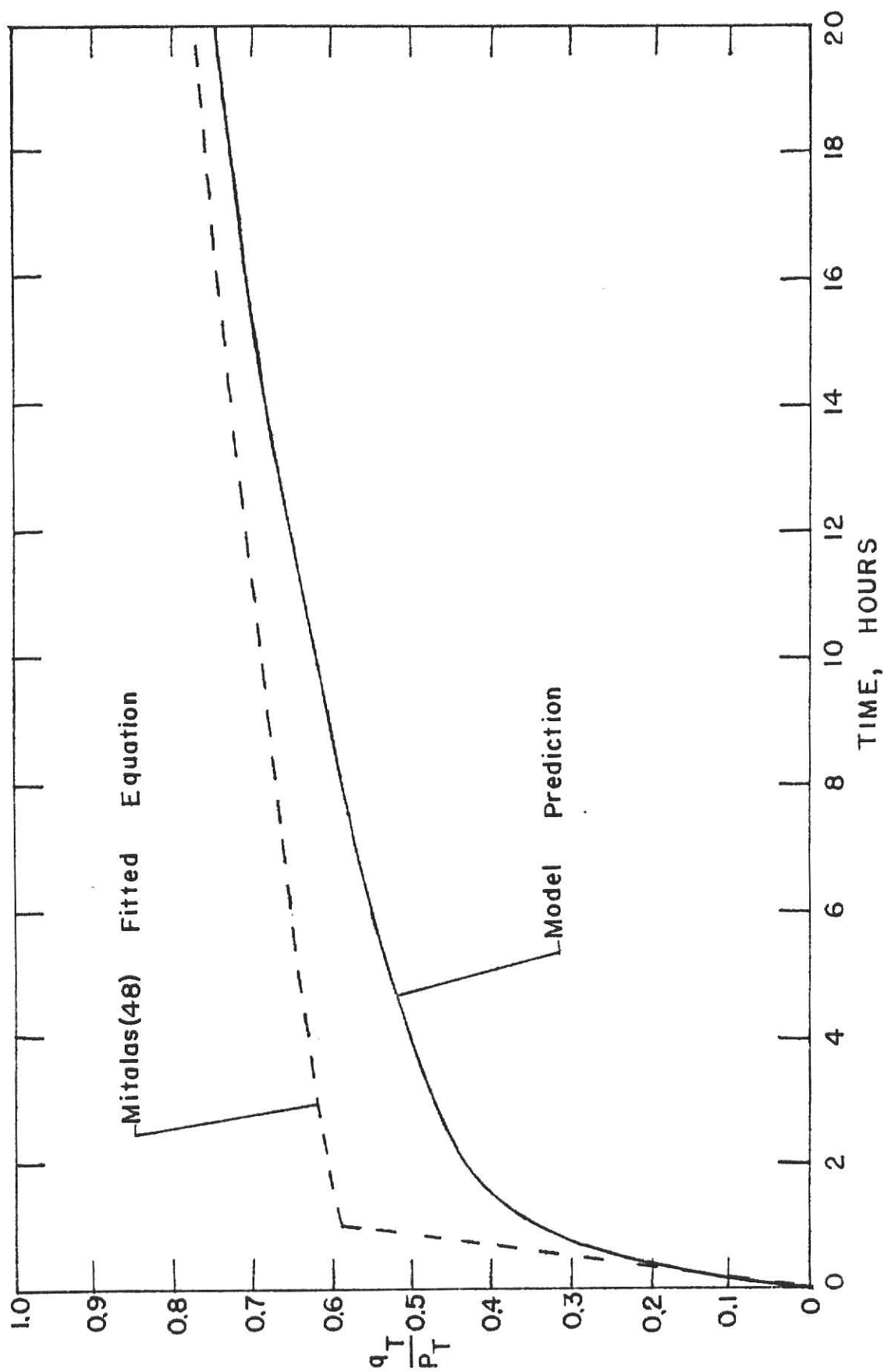


Figure 35. Comparison of q_T/P_T from Model Prediction and Mitalas (48) Fitted Equation - Tests 23, 24, and 25.

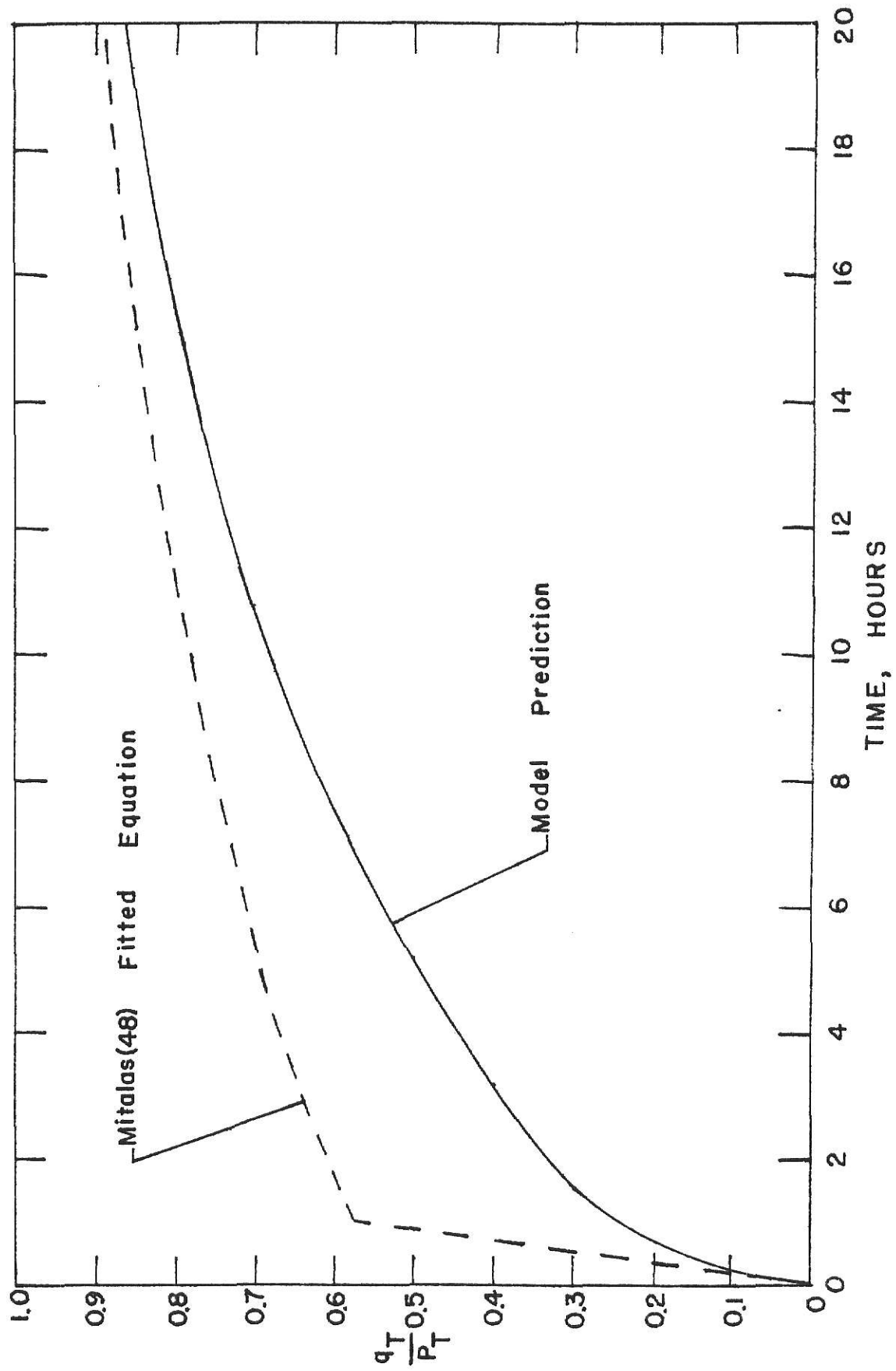


Figure 36. Comparison of q_T/P_T from Model Prediction and Mitalas (48) Fitted Equation - Test 26.

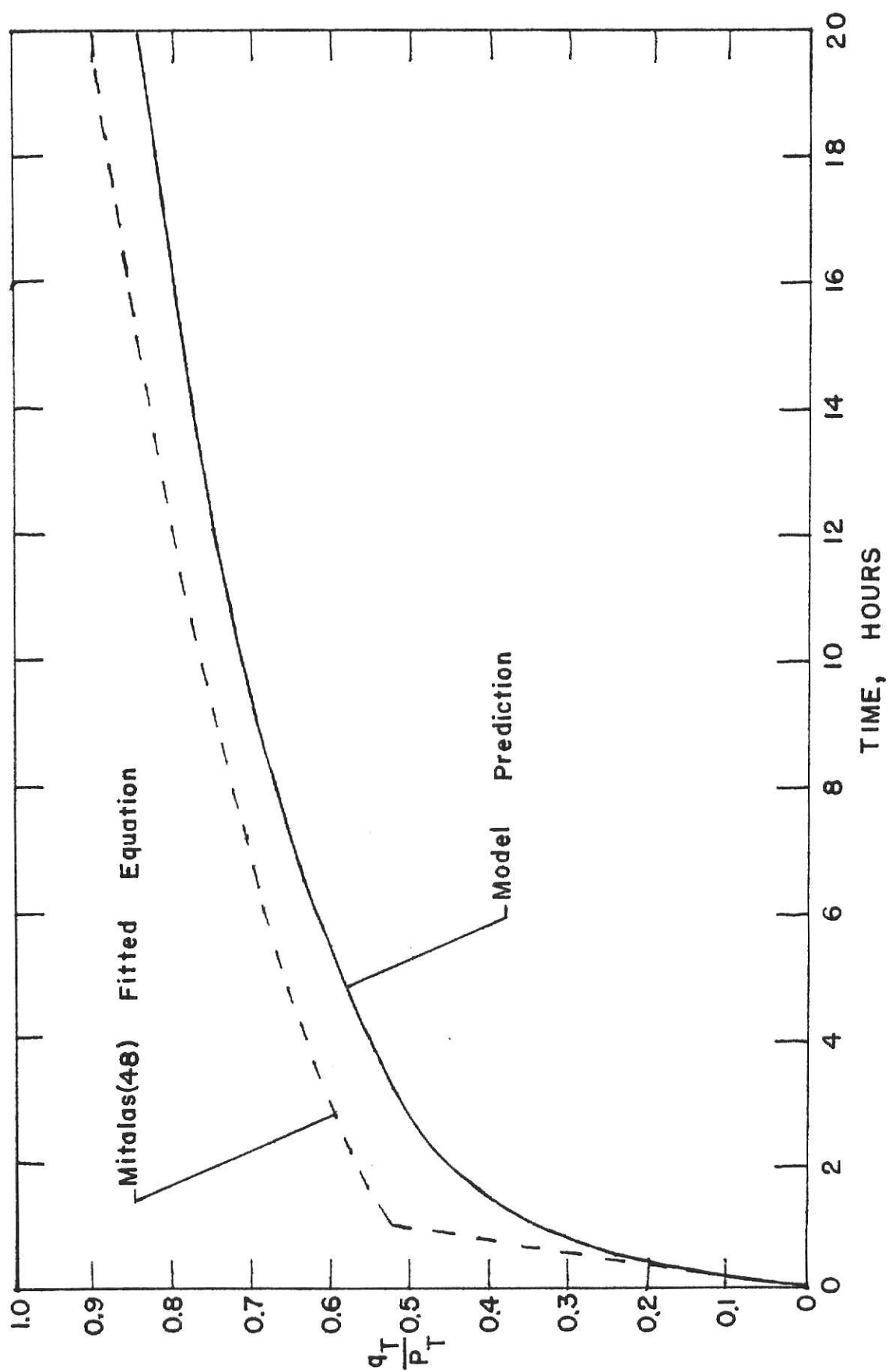


Figure 37. Comparison of q_T/P_T from Model Prediction with Adjusted Carpet Properties and Mitalas (48) Fitted Equation - Tests 7 and 8.

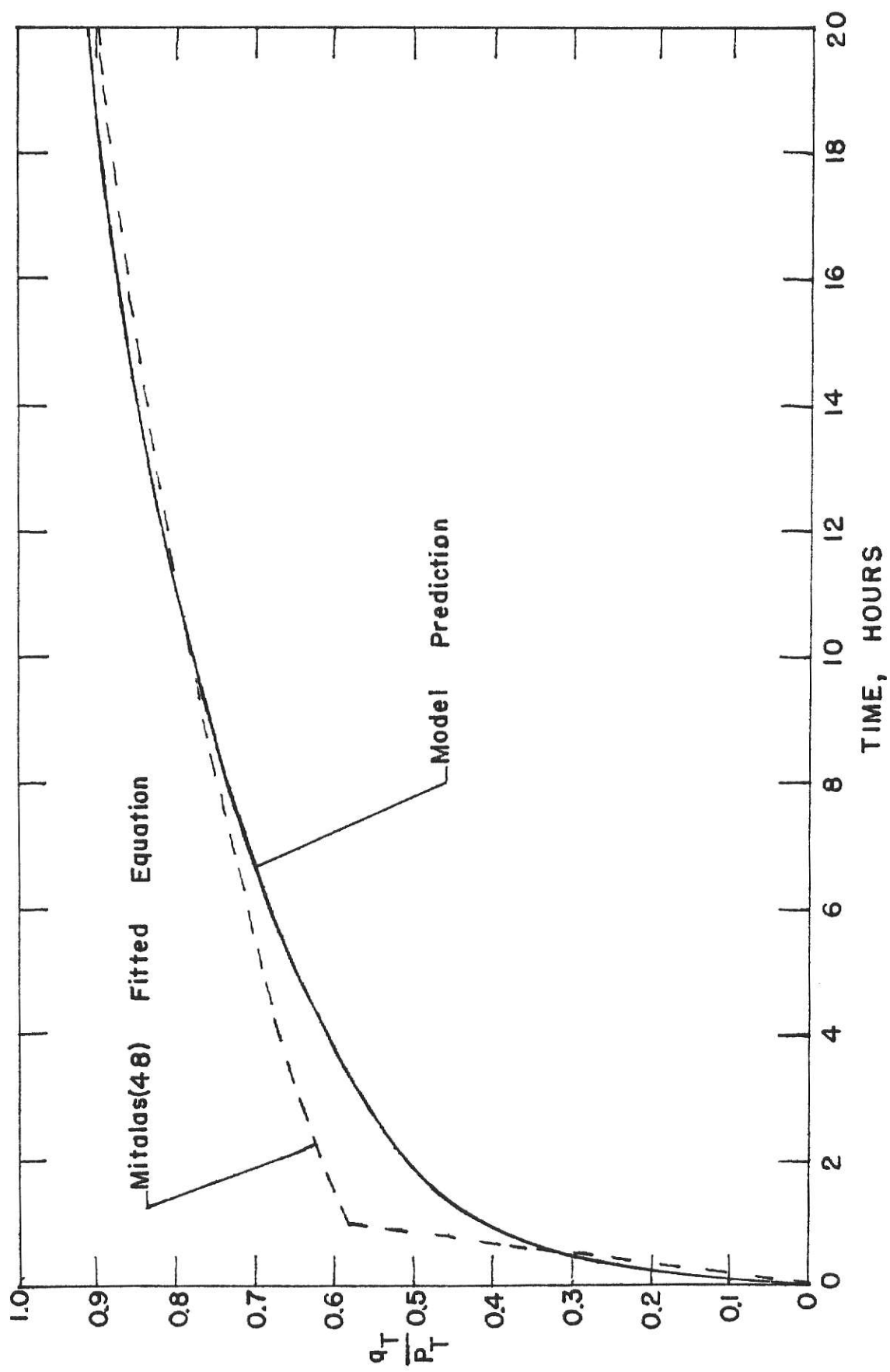


Figure 38. Comparison of q_T/P_T from Model Prediction with Adjusted Carpet Properties and Mitalas (48) Fitted Equation - Tests 12 and 13.

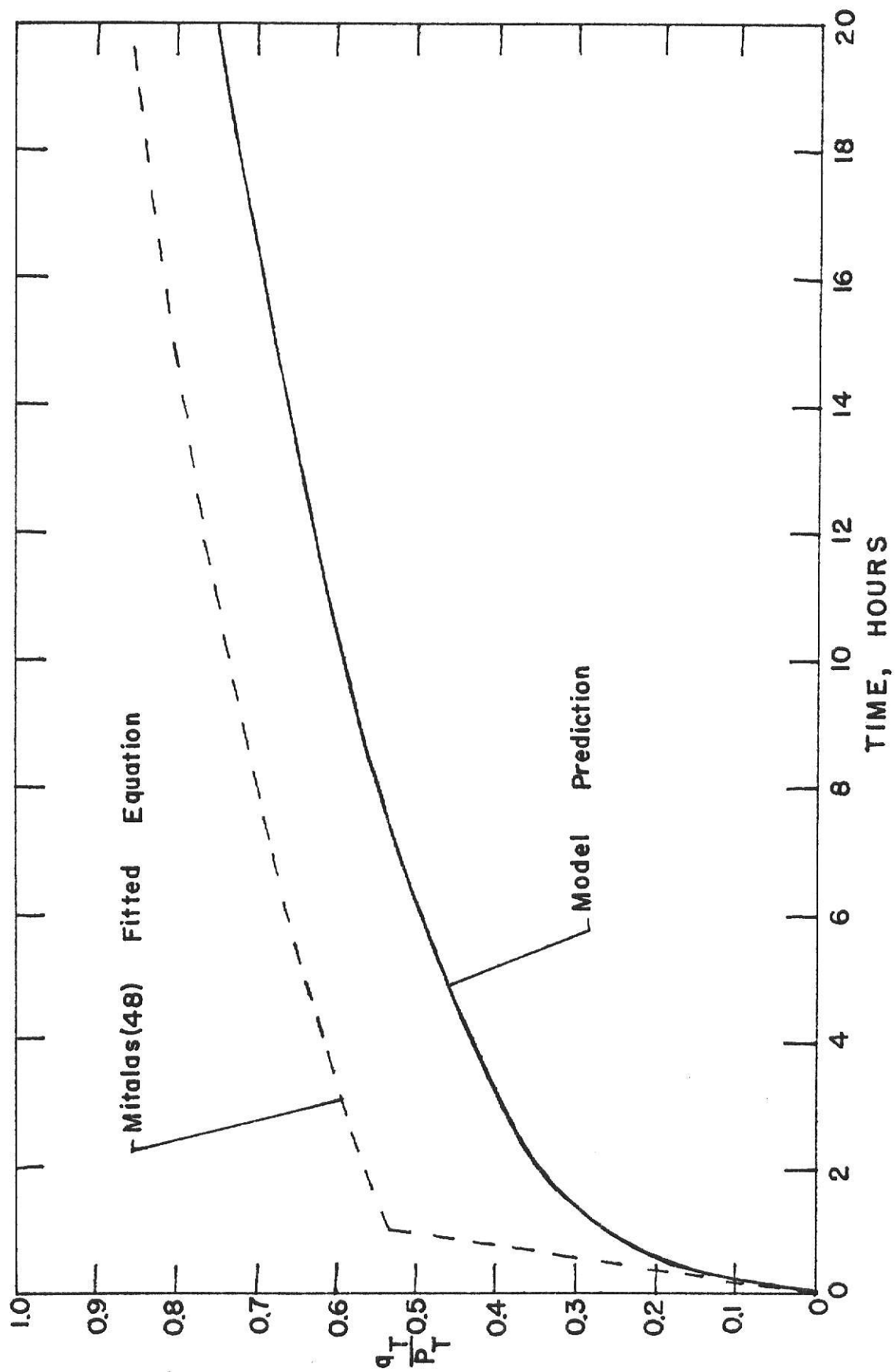


Figure 39. Comparison of q_T/P_T from Model Prediction with Adjusted Carpet Properties and Mitalas (48) Fitted Equation - Tests 14, 15, 16, and 17.

much closer to the fitted equation, even though other conditions are similar to those of Figs. 33 and 39.

During this testing, it was noticed that the carpet heat capacitance is often too small for the carpet temperature to be stable when using the general time period. To use the smaller time period, $\Delta t'$, for carpet calculations would have required the computer program to be extensively rewritten. Therefore, a mean temperature method, similar to that described in Appendix C, was incorporated into the carpet calculations. Although this method solved the stability problem, it is not entirely satisfactory because radiation equations, which are functions of temperatures to the fourth power, had to be linearized.

The accuracy of the model was also checked when the lights are shut off. Figs. 40 and 41 are comparisons of model predictions and the fitted equations of Mitalas (48) for Tests 4 and 26, respectively, when the lighting is turned off after 20 hours. The behavior of both model predictions is very similar to the corresponding curves in Figs. 29 and 36, indicating that the model works as well under lights off conditions and also that the model is generally working as it should.

Next, the model prediction was checked over a simulated 5-day period. The lights were on for the first 9 hours, then off for 15 hours for each of 5 consecutive 24-hour periods. This was meant to model a response similar to what occurs in an office building used during a 5-day work week. The 5-day model predictions and the corresponding fitted equations of Mitalas (48) for the room conditions used in Test 4 are shown in Fig.

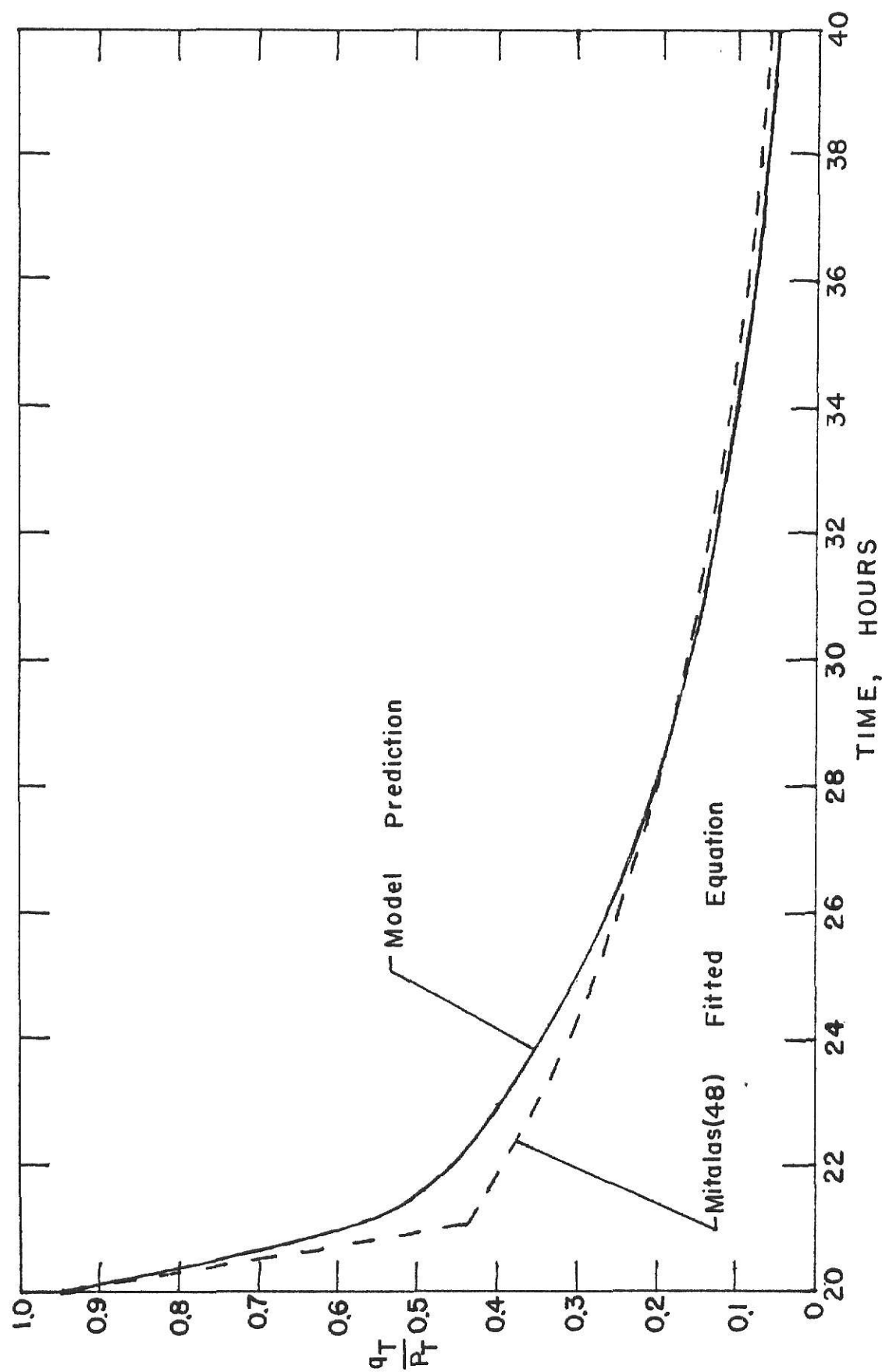


Figure 40. Comparison of q_T/P_T when the Lights are Switched Off at Time = 20 Hours from Model Prediction and Mitalas (48) Fitted Equation - Test 4.

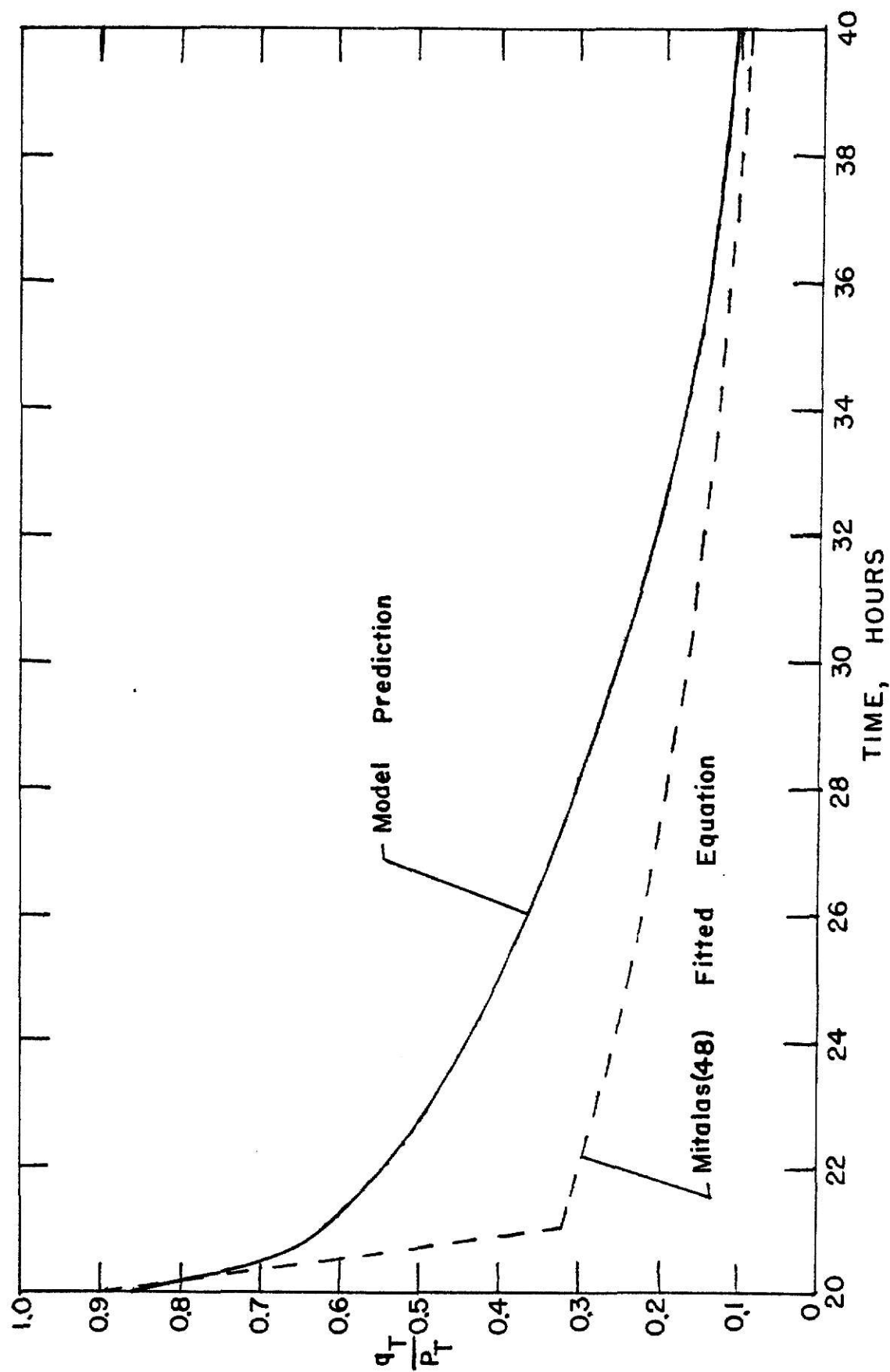


Figure 41. Comparison of $\frac{q_T}{P_T}$ when the Lights are Switched Off at Time = 20 Hours from Model Prediction and Mitalas (48) Fitted Equation - Test 26.

42. Only the results from the first and third days are shown because the results from the second day are only slightly below the third day results and after that, there was hardly any change in either the model predictions or the fitted equations. The model predictions followed the behavior of the fitted equations closely and there were no problems with oscillation or stability from day to day.

The majority of the comparisons of q_T/P_T between the model predictions and the fitted equations of Mitalas (48) shown in Figs. 26-42 are in good agreement. But for the tests using a static plenum and static luminaires, Figs. 26, 32, 34, 35, 38, and 40, the model predictions do not become asymptotic with the fitted equations until much later in the tests. Mitalas (48) used an equation which cannot follow a steep curve for a few hours and a flatter arc afterwards. To calculate the time constants a_1 , a_2 , and b_1 , Mitalas (48) did not use the cooling loads measured during the first few hours, thus he may have missed some exponential terms that decay after a few hours. When the luminaires or plenum are vented, much of the heat going into q_T comes from luminaire surfaces, which reach equilibrium very quickly. If the plenum or luminaires are not vented, all heat going into q_T must be convected from heavy room surfaces, which have much longer thermal time periods. This is the reason for the large differences between the fitted equations of Mitalas (48) and the model predictions during the early hours of the tests when there are no vented luminaires or a vented plenum.

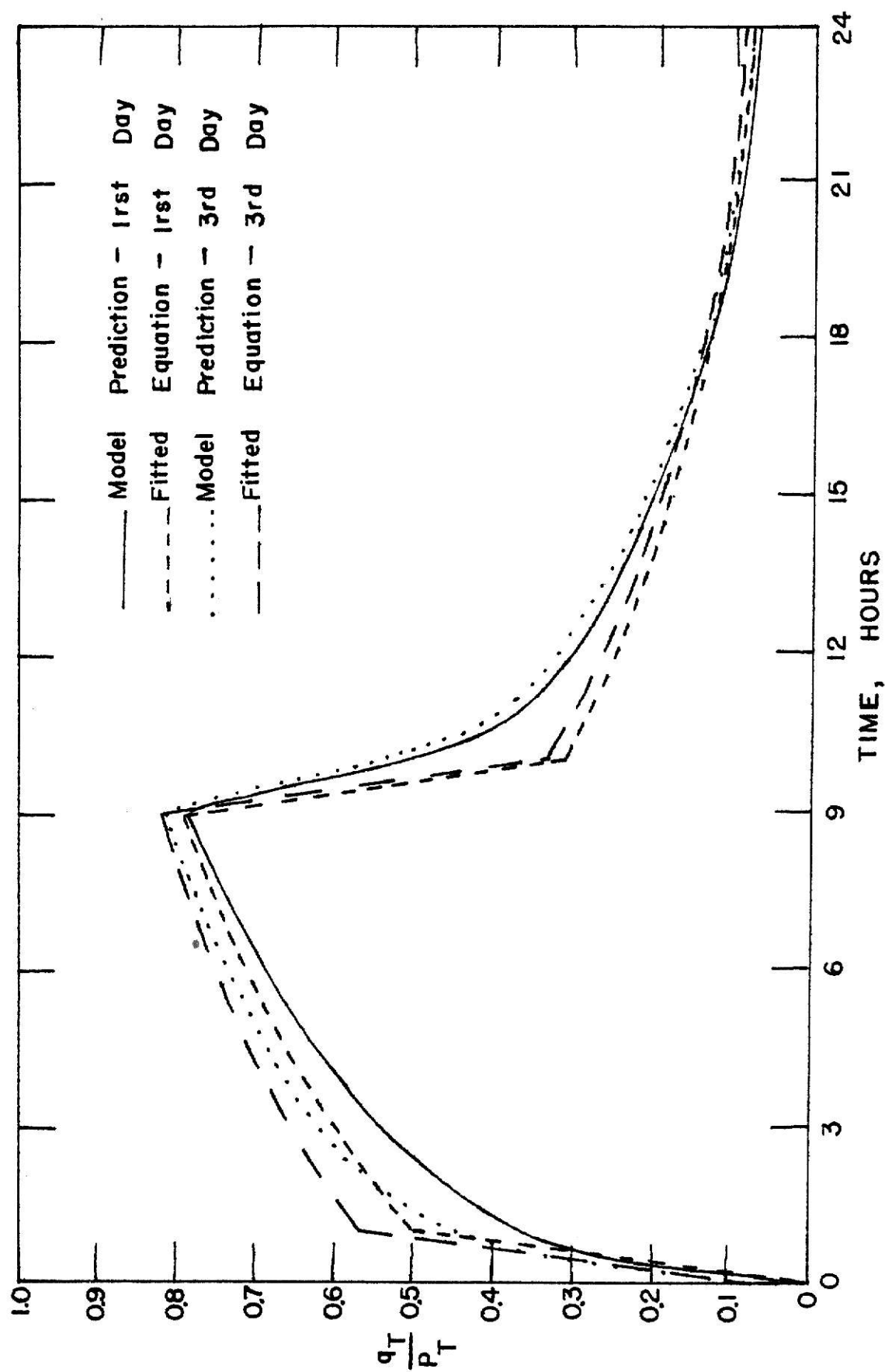


Figure 42. Comparison of q_T/P_T from Model Prediction and Mitalas (48) Fitted Equation for the 1st and 3rd Days of a 5-Day Continuous Run - Test 4.

A factor that has not been considered and could increase q_R slightly is the supply air coming out through a ceiling diffuser or high wall grill. This air is blown across the ceiling and causes extra heat transfer to the room by increasing both the convection coefficients and the temperature differentials near the ceiling. Obtaining an equation for the average increase in these factors is much too complicated an undertaking for this thesis. But, an approximation for this specific situation was made to show how this problem could be handled. Chapter 30 of the ASHRAE Handbook 1977 Fundamentals (1) concerns space air diffusion. Fig. 43, which was drawn from Fig. 2 of that chapter, shows the air flow conditions in a room with a ceiling diffuser. ASHRAE (1) describes the primary air as being delivered directly from the supply duct and having velocities greater than 150fpm. ASHRAE describes the total air as:

...principally influenced by the primary air and is of relatively high velocity, but less than 150fpm, with an air temperature generally within 1°F of room temperature. The total air is also influenced by the environment and tends to drop during cooling and rise during heating. The total air is not subject to precise analytical treatment.

From Fig. 43, it appears that most of the total air is near the ceiling. If it is assumed that all of the supply air flows within a foot of the ceiling before diffusing into the room, a mean air velocity can be calculated. This velocity can be used in Eq. (18), along with a length parameter equal to the mean room radius, to calculate an increase in the ceiling convection coefficients. If it is also assumed that the experiments this sketch was derived from were done at steady state, then the

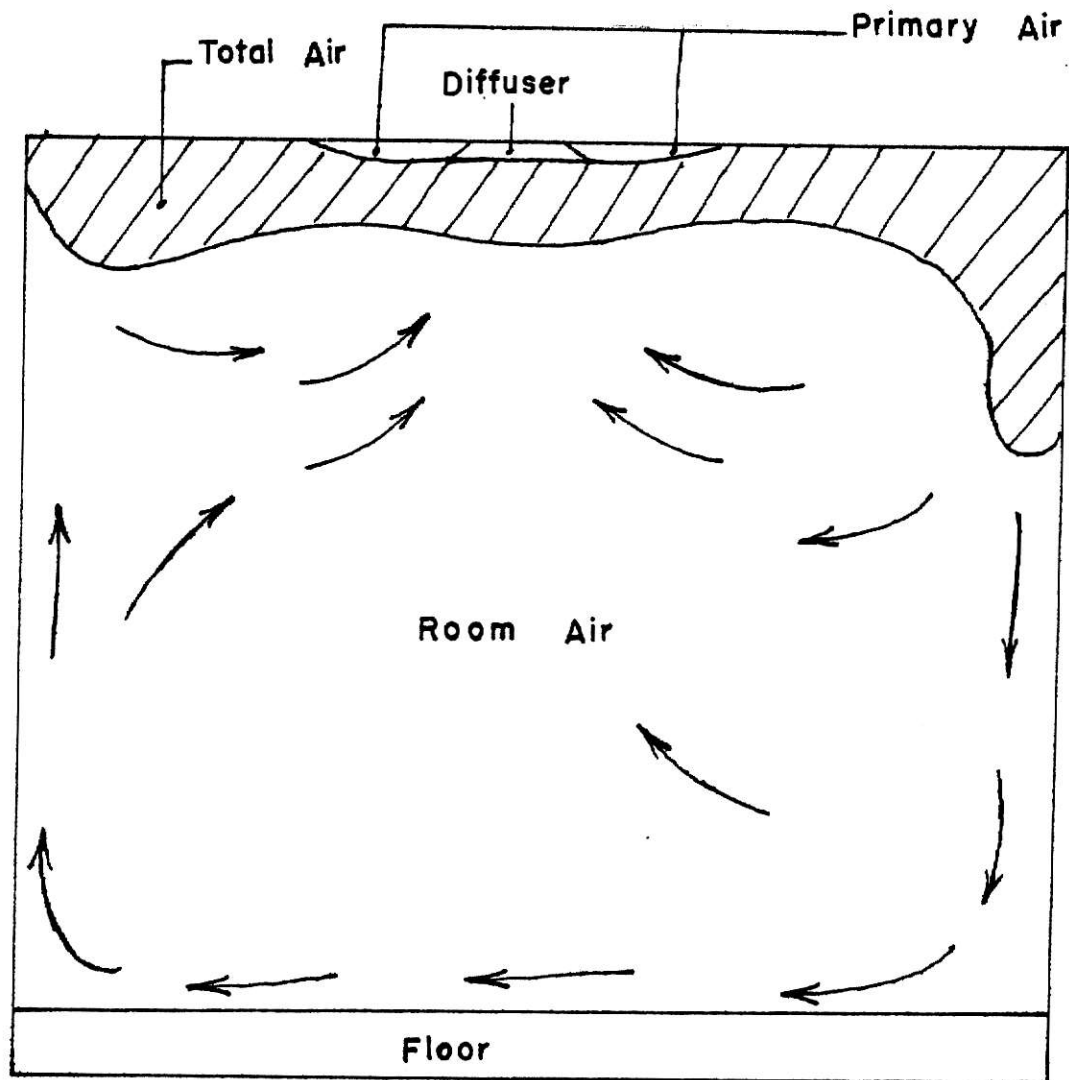


Figure 43. Air Motion Characteristics of Rooms with Circular Ceiling Diffusers.

supply air would have accounted for the full energy load. But, when a transient condition is considered, the supply air temperature differs from the room air temperature in proportion to q_R/P_T . Thus the temperature of the total air could also be assumed to be a fraction of 1°F different from the room air temperature as given by:

$$T_a = T_R - 1^\circ\text{F}(q_R/P_T) \quad (40)$$

The primary air has a temperature that differs from the room air temperature much more than the total air temperature, but the primary air covers only a small portion of the ceiling. Still, the presence of this primary air may mean the ceiling is convecting to air at an average temperature more than 1°F different from the room air temperature. The velocity effects of the primary air have already been accounted for. To see what effects considering such a model produces, Cases A and B, which were discussed earlier in this chapter, were run with an increased ceiling convection coefficient and with 1°F and 2°F maximum temperature differences for Eq. (40). The results are shown in Figs. 44 and 45. From these results, it appears that the supply air has an important effect on the transfer of lighting energy. It also appears that the effect is greater in the case of a static plenum. Further investigation of this problem and methods of modeling it will be made later in this research project.

There are some other characteristics of the calorimeter room of Mitalas (48) that were not considered or could only be guessed at, and errors may have resulted because of this. The walls of the experimental test room were lined with aluminum

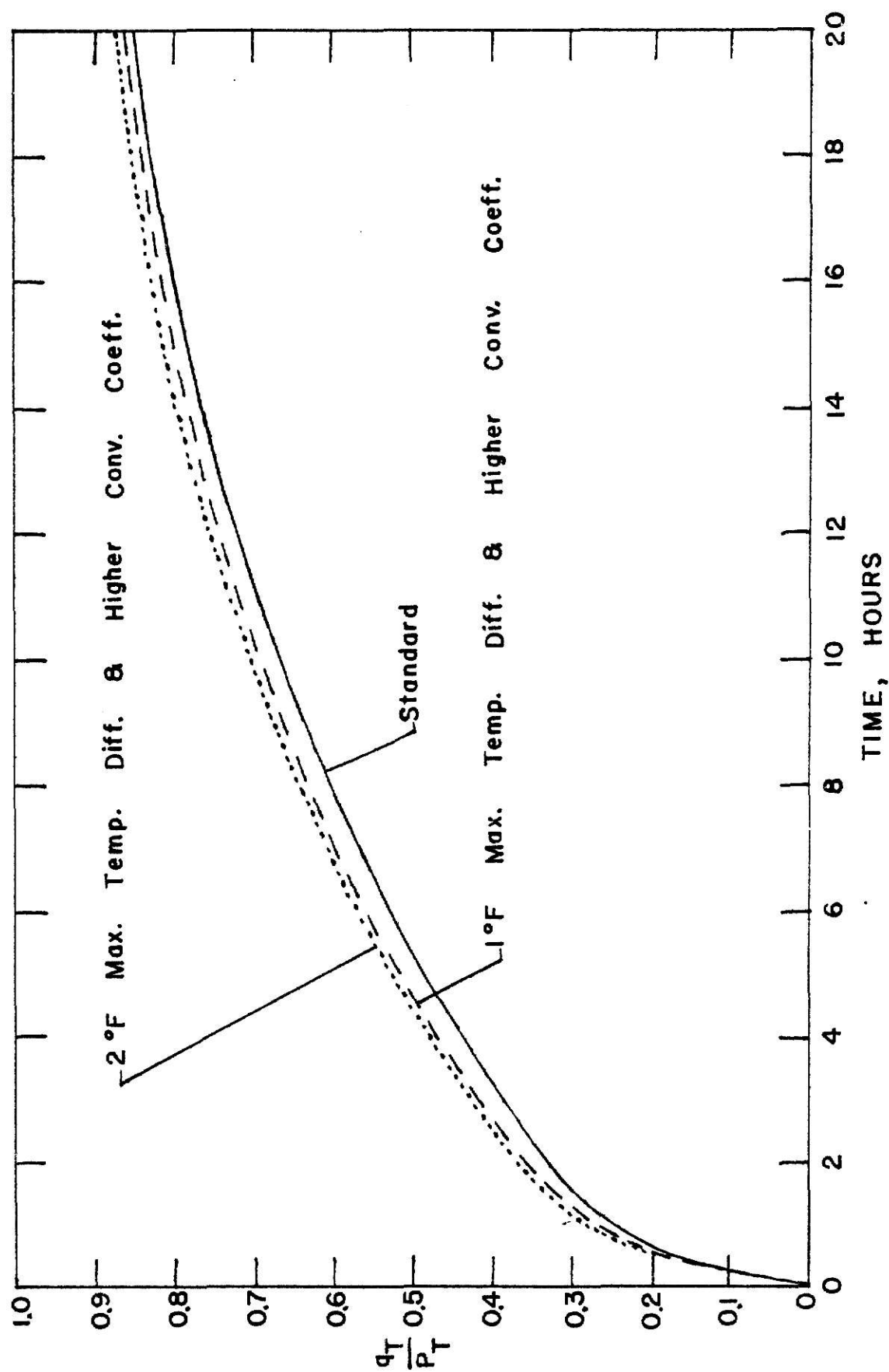


Figure 44. The Changes in Model Prediction when Higher Temperature Differences and Convection Coefficients are Used Near the Ceiling - Case A - Static Plenum.

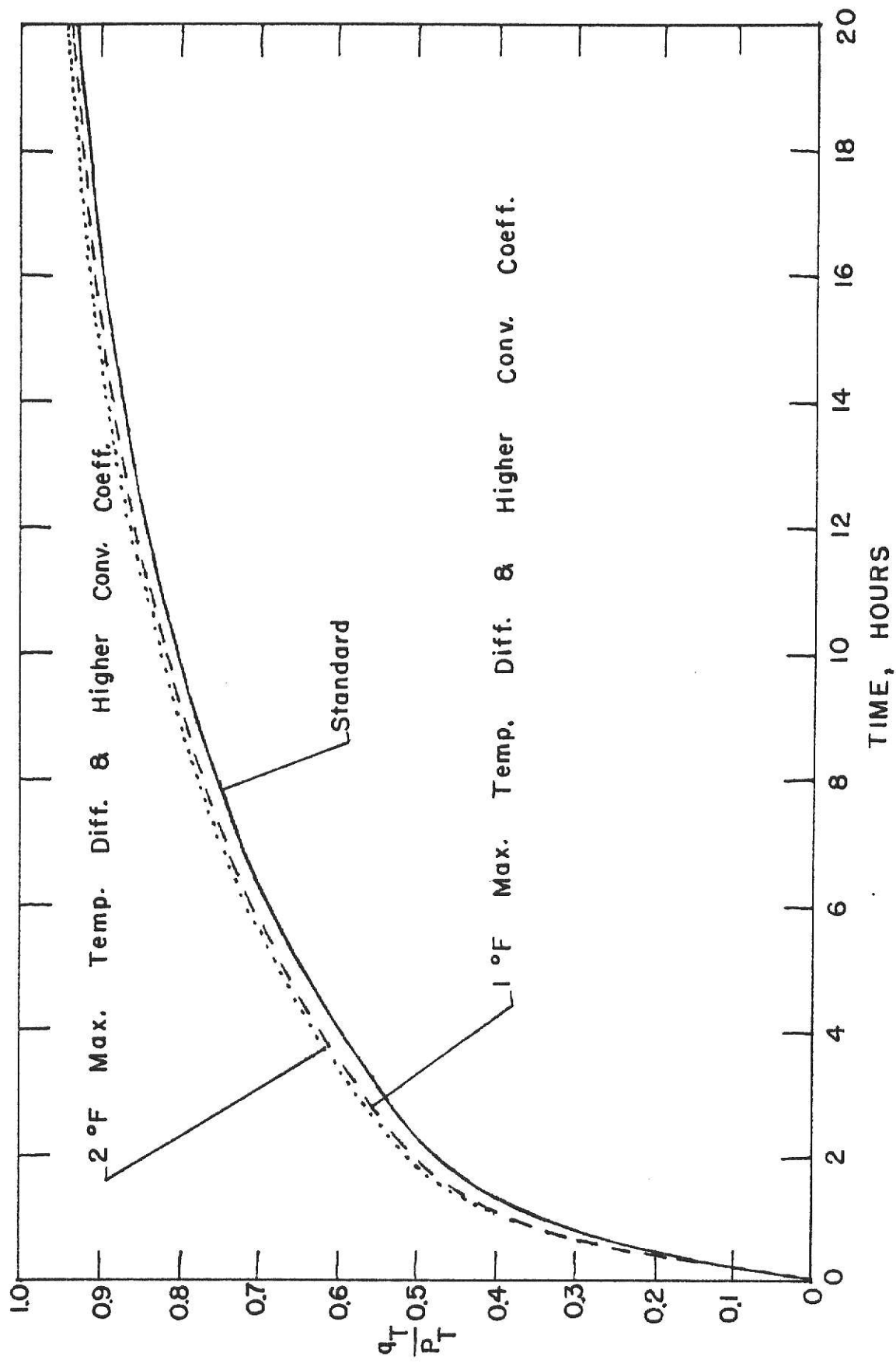


Figure 45. The Changes in Model Prediction when Higher Temperature Differences and Convection Coefficients are Used Near the Ceiling - Case B - Return Plenum.

foil to reflect most of the light to the floor, but aluminum still absorbs about 20% of the incident short wavelength radiation. If the foil and the wall were not in good thermal contact, most of the energy absorbed by the foil would have been quickly convected to the room air. The ducting is another room component which is not described well by Mitalas (48), and its surface area and U-value had to be estimated. Comparing Figs. 28 and 29 and Figs. 27 and 36 shows that the fitted equations used by Mitalas (48) differ during the early hours of the tests much more than the model predictions, even though the main difference is the presence of insulation around the ducting. The surface reflectances, especially those of the luminaire surfaces, were also unknown, as was the transmittance of the lens. Because the range of values for these properties is not large, the surface characteristics were probably well estimated. But the amount of light to the room will change with even small differences in these properties.

CHAPTER V

CONCLUSION

From the comparisons shown on the previous pages between the model predictions and both steady state and transient experimental results, the author feels that a mathematical model of the type outlined in this thesis can be used with good accuracy to describe the actual energy transfer from lighting. The surface temperatures predicted using the model follow the trends obtained from past testing. The results obtained using the mathematical model also compared well with tests when the flow rates through the luminaire were changed. And most importantly, the model predictions were close to the fitted equations of Mitalas (48) for transient analysis, in many cases being within the accuracy of the equations themselves and almost always approaching the slopes of the fitted equations.

This model is also flexible enough to be used to predict the energy transfer from lighting for almost any circumstance. The basic program allows for a large variety of conditions to be modeled by changing the input only. Some changes, such as light performance, supply air temperature, and convection coefficients can be accounted for by inserting the proper equations. Any other problems with specific room conditions can be handled by using the methods outlined in this thesis.

For these two reasons, accuracy and flexibility, this model fulfills the requirements ASHRAE specified and should be used to

analyze the heat transfer from lighting for ASHRAE Research
Project RP-231.

CHAPTER VI

RECOMENDATIONS

There are three tasks remaining in order to complete
RP-231:

1. Establishment of a plan for parametric study of many types of lighting and fixtures.
2. Carry out this plan using the mathematical model.
3. Development of easily used tabular or graphical condensations of the results.

This author would like to offer several suggestions that may help the work on these later parts of the project:

1. The number of variations in the different parameters is quite large, but limiting the need for accuracy may reduce the complexity of the problem.
2. A very thorough investigation should be made into the energy consumption and transmission characteristics of each type of luminaire studied, especially in relation to temperature.
3. Some changes or additions to the program may need to be made to account for air leakage through suspended ceilings, side slots on luminaires for supply air, and other specific conditions.
4. The effects of different supply air arrangements will have to be modeled to some degree.
5. The heat transfer during the first few hours after the

lights are switched should be closely modeled, both for better air conditioner sizing and better heater sizing.

6. The performance of lights as they age might cause higher loads and thus should be investigated.
7. The final results should remain useful for many years and thus must allow for some increase in efficiency and new developments.

REFERENCES

1. ASHRAE Handbook & Product Directory-1977 Fundamentals Vol., The American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., New York, 1977.
2. ASHRAE Handbook & Product Directory-1976 Systems Vol., The American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., New York, 1976.
3. Ballman, T. L., Bradley, R. D., and Hoelscher, E. C., "Calorimetry of Fluorescent Luminaires," Illuminating Engineering, Vol. 59, No. 12, Dec., 1964, pp. 779-785.
4. Ballman, T. L., Bradley, R. D., Schmidt, R. D., and Winston, J. R., "Economic Effects of Lighting and Air Conditioning on Building Costs," IEEE-Trans. on Applications and Industry, Vol. 83, No. 3, July, 1964, pp. 206-208.
5. Bevans, J. T. and Dunkle, R. V., "Radiant Interchange within an Enclosure," ASME Trans.-Jour. of Heat Transfer, Vol. 82, No. 1, Feb., 1960, pp. 1-19.
6. Bevans, J. T. and Edwards, D. K., "Radiation Exchange in an Enclosure with Directional Wall Properties," ASME Trans.-Jour. of Heat Transfer, Vol. 87, No. 3, Aug., 1965, pp. 388-396.
7. Bonvallet, G. G., "Methods of Determining Energy Distribution Characteristics of Fluorescent Luminaires," Illuminating Engineering, Vol. 61, No. 2, Feb., 1963, pp. 69-74.
8. Boyer, L. L., Jr., "The Role of the Supply-Air Plenum in Interior Zone Recessed Illumination Systems," Illuminating Engineering, Vol. 61, No. 8, Aug., 1966, pp. 528-533.
9. Boyer, L. L., Jr., "Thermal Environment in Perimeter Office Spaces Cooled with Ventilating Ceilings," ASHRAE Trans., Vol. 73, Part 1, 1967, pp. III3.1-III3.11.
10. Boyer, L. L., Jr., "Radiant Panel Effects for Floor-Ceiling Assemblies Incorporating Static, Supply, and Return Plenums," ASHRAE Trans., Vol. 74, Part 2, 1968, pp. 31-42.
11. Carroll, J. R., Jr., "Natural Convection in Panes Heating," Heating and Ventilating Jour., Vol. 48, No. 1, Jan., 1948, pp. 70-76.

12. Clark, J. A. and Korybalski, M. E., "Radiation Heat Transfer in an Enclosure Having Surfaces which are Adiabatic or of Known Temperature," Proceedings of the First National Heat and Mass Transfer Conference, Indian Institute of Technology, Madras, 1971, pp. III-1-III-20.
13. Clark, J. A. and Korybalski, M. E., "Algebraic Methods for the Calculation of Radiation Exchange in an Enclosure," Wärme-und Stoffübertragung Thermo-Fluid Dynamics, Vol. 7, No. 1, 1974, pp. 31-44.
14. Cooper, K. W. and Tree, D. R., "A Re-evaluation of the Average Convection Coefficient for Flow Past a Wall," ASHRAE Trans., Vol. 79, Part 1, 1973, pp. 48-51.
15. Cuplinskas, E. L., "Thermal Response Calculations by the Finite Difference Method," ASHRAE Trans., Vol. 83, Part 2, 1977, pp. 118-131.
16. Dally, G. R. and Ballman, T. L., "Higher Lighting Levels and Interior Zone Air-Distribution Systems: Action-Reaction," Illuminating Engineering, Vol. 60, No. 1, Jan., 1965, pp. 43-48.
17. Darley, W. G., "Cooler Footcandles for Air Conditioning," ASHVE Trans., Vol. 44, 1938, pp. 213-244.
18. Darley, W. G., "Three in One-Light, Sound, and Air Conditioning," Magazine of Light, Vol. 9, No. 2, 1940, pp. 27-29.
19. Dusenberre, G. M., Heat Transfer by Finite Differences, International Textbook Company, Scranton, 1961.
20. "Effects of Ceiling Plenum Returns and Recessed Lighting Fixtures," Carrier Technical Newsletter, May, 1974.
21. Fischer, W. D. and Hering, R. G., "Radiant Heat Transfer between Nongray Directional Surfaces," Jour. of Spacecraft and Rockets, Vol. 9, No. 12, Dec., 1972, pp. 915-920.
22. Fisher, W. S., "Electric Space Conditioning," Electrical Construction and Maintenance, Vol. 64, No. 7, July, 1965, pp. 82-85.
23. Fisher, W. S., Roehr, L. A., and Smith, J. M., "Some Factors Affecting Heat Transfer In Luminaires," Illuminating Engineering, Vol. 60, No. 1, Jan., 1965, pp. 51-66.
24. Gross, S., "An Improved Finite Difference Method for Heat-Transfer Calculations," Jour. of Spacecraft and Rockets, Vol. 14, No. 4, April, 1967, p. 538.

25. Hanna, G. B., "Computer Analysis to Estimate the Temperature Response of an Enclosure by the Finite Difference Method," Building Services Engineer, Vol. 45, No. 8, Aug., 1977, pp. 59-69.
26. Henderson, T. A., "Heat Transfer from Ceiling Plenums," Air Conditioning, Heating, and Ventilation, Nov., 1966, pp. 72-76.
27. Hering, R. G. and Fischer, W. D., "Radiant Heat Transfer between Nongray Surfaces," Progress in Astronautics and Aeronautics, Vol. 29, 1972, pp. 269-288.
28. Hottel, H. C. and Sarofim, A. F., Radiative Transfer, McGraw-Hill Book Company, 1967.
29. Houchens, A. F. and Hering, R. G., "Directional Property Effects on Radiant Heat Transfer and Equilibrium Temperatures," Progress in Astronautics and Aeronautics, Vol. 29, 1972, pp. 243-268.
30. "IES Approved Guide for the Photometric and Thermal Testing of Air Cooled Heat Transfer Luminaires," Journal of the IES, Oct., 1978, pp. 57-62.
31. Ishimoto, T. and Bevans, J. T., "Methods of Evaluating Script F for Radiant Exchange within an Enclosure," AIAA Jour., Vol. 1, No. 6, June, 1963, pp. 1428-1429.
32. Jackson, T. W. and Howard, H. Y., "Combined Forced and Free Convective Equations to Represent Combined Heat-Transfer Coefficients for a Horizontal Cylinder," ASME Trans.-Jour. of Heat Transfer, Vol. 93, No. 3, May, 1971, pp. 247-248.
33. Kaufman, J. E. and Christensen, J. F., IES Lighting Handbook, Fifth Edition, The Illuminating Engineering Society, New York, 1978.
34. Kelles, G. J., "The Influence of Furniture on Cooling Loads: Proposal of a Simple Method," Building Services Engineer, Vol. 40, No. 7, July, 1972, pp. 79-82.
35. Kimura, K., Isrino, H., Hidetaka, K., and Junji, K., "Study on the Comparative Verification of Various Methods of Air Conditioning Load Calculation by Field Measurements," Second Symposium on the Use of Computers for Environmental Engineering Related to Buildings, June 13-15, 1974, pp. 1-15.
36. Kimura, K., Koyo, M., Kenji, I., Yoshiadi, T., Teiji, H., and Yoshinobu, K., "Experimental Study on the Heat Removal from Lighting through a Troffer with Intake of Air Conditioning," Trans. SHASE-Japan, Vol. 5, 1967, pp. 1-15.

37. Kimura, K. and Stephenson, D. G., "Theoretical Study of Cooling Load Caused by Lights," ASHRAE Trans., Vol. 74, Part 2, 1968, pp. 189-197.
38. Lauvray, T. L., "Experimental Heat Transmission Coefficients for Operating Air Duct Systems," ASHRAE Jour., June, 1978, pp. 69-73.
39. "Lighting and Air Conditioning," Illuminating Engineering, Vol. 61, No. 3, March, 1966, pp. 123-147.
40. Lighting and Air Conditioning...Part 1-The Lighting System's Relation to the Cooling Load," Illuminating Engineering, Vol. 51, No. 9, Sept., 1956, pp. 647-652.
41. "Lighting and Air Conditioning Design Factors," Heating, Piping, and Air Conditioning, Sept., 1941, pp. 605-608.
42. Lim, K. and Dean, E., "Energy Conservation Opportunities in Luminaire Design and Instalation," Energy and Building, Vol. 1, No. 1, May, 1977, pp. 65-67.
43. Luckiesh, M., "We Have Wanted Cooler Footcandles... Now We Have Them," Magazine of Light, Vol. 7, No. 8, 1938, pp. 23-24, 38.
44. Mathews, C. P., "Ceiling Temperature Equations," Building Systems Design, April, 1971, pp. 37-38.
45. McMenamin, D. L., "Combined Roof and Recessed Light Loads with Ceiling Insulation and Ventilation," Proceedings of the Conference on Improving Efficiency and Performance of HVAC Equipment and Systems for Commercial and Industrial Buildings, Vol. 1, April 12-14, 1976, pp. 2-11.
46. Miller, P. L. and Nevins, R. G., "An Analysis of the Performance of Room Air Distribution Systems," ASHRAE Trans., Vol. 78, Part 2, 1972, pp. 191-198.
47. Min, T. C., Schutrum, L. F., Parmelee, G. V., and Vouris, J. D., "Natural Convection and Radiation in a Panel Heated Room," ASHRAE Trans., Vol. 62, 1956, pp. 337-358.
48. Mitalas, G. P., "Cooling Load Caused by Lights," CSME, Trans., Vol. 2, No. 3, 1973-1974, pp. 169-174.
49. Mitalas, G. P., "Calculating Cooling Load Caused by Lights," ASHRAE Jour., June, 1973, pp. 37-40.
50. Mitalas, G. P., "Comments on the Z-Transfer in Buildings," ASHRAE Trans., Vol. 84, part 1, 1978, pp. 667-674.
51. Mitalas, G. P. and Kimura, K., "A Calorimeter to Determine

- Cooling Loads Caused by Lights," ASHRAE Trans., Vol. 77, Part 2, 1971, pp. 65-72.
52. Micoglu, A. and Chen, T. S., "Mixed Convection Across a Horizontal Cylinder with Uniform Surface Heat Flux," ASME Trans.-Jour. of Heat Transfer, Vol. 99, No. 4, Nov., 1977, pp. 679-681.
 53. Mueller, T. and Benson, B. S., Jr., "Testing and Performance of Heat-Removal Troffers," Illuminating Engineering, Vol. 57, No. 12, Dec., 1962, pp. 793-802.
 54. Nevins, R. G., Straub, H. E., and Ball, H. D., "Thermal Analysis of Heat Removal Troffers," ASHRAE Trans., Vol. 77, Part 2, 1971, pp. 58-64.
 55. Nottage, H. B. and Park, K. S., "Performance of a Vented Luminaire," ASHRAE Trans., Vol. 75, Part 2, 1969, pp. 50-70.
 56. O'Brien, P. F. and Krumland, L. R., "An Investigation of the Luminous and Thermal Radiative Transfer Characteristics of Luminaires," Illuminating Engineering, Vol. 56, No. 4, April, 1970, pp. 218-230.
 57. Oosthuizen, P. H. and Madan, S., "Combined Convective Heat Transfer from Horizontal Cylinders in Air," ASME Trans.-Jour. of Heat Transfer, Vol. 92, No. 1, Feb., 1970, pp. 194-196.
 58. Parmelee, G. V. and Huebscher, R. G., "Forced Convection Heat Transfer from Flat Surfaces," ASHVE Trans., Vol. 53, 1947, pp. 245-284.
 59. Quin, M. L. and Bradley, R. D., "Convective Transfer of Lighting Heat Loads," Illuminating Engineering, Vol. 57, No. 1, Jan., 1962, pp. 45-56.
 60. "Removing Heat from High-Intensity Lighting," Electrical World, April 15, 1933, pp. 480-481.
 61. Roberts, G. G. M., "The Problem of Heat Gain from High Levels of Illumination," IHVE, Jour., Jan., 1964, pp. 349-370.
 62. Schornhorst, J. R. and Viskanta, R., "An Experimental Examination of the Validity of the Commonly Used Methods of Radiant Heat Transfer Analysis," ASME Trans.-Jour. of Heat Transfer, Vol. 90, No. 4, Nov., 1968, pp. 429-436.
 63. Siegel, R. and Howell, J. R., Thermal Radiation Heat Transfer, McGraw-Hill Book Company, 1972.

64. Somerville, R. C., "Lighting in Relation to Heating and Ventilating," Electrical Review (Supplement), Sept. 21, 1973, pp. 23-26.
65. Sowell, E. F. and O'Brien, P. F., "The Transport of Lighting Energy," ASHRAE Trans., Vol. 79, Part 2, 1973, pp. 172-182.
66. Sparrow, E. M. and Cess, R. D., Radiation Heat Transfer, Brooks/Cole Publishing Company, Belmont, 1966.
67. Stephenson, D. G. and Mitalas, G. P., "Cooling Load Calculations by Thermal Response Factor," ASHRAE Trans., Vol. 73, Part 1, 1967, pp. III1.1-III1.7.
68. Stephenson, D. G. and Mitalas, G. P., "Room Thermal Response Factors," ASHRAE Trans., Vol. 73, Part 1, 1967, pp. III2.1-III2.10.
69. Sturrock, W., "Effects of Artificial Lighting on Air Conditioning," ASHVE Trans., Vol. 44, 1938, pp. 213-244.
70. Sylvester, G. C., "Performance of a Heat Removal Troffer with a Specific Building Design," ASHRAE Trans., Vol. 70, 1964, pp. 328-334.
71. Toor, J. S. and Viskanta, R., "A Critical Examination of the Validity of Simplified Models for Radiant Heat Transfer Analysis," International Jour. of Heat and Mass Transfer, Vol. 15, 1972, pp. 1553-1567.
72. Toor, J. S. and Viskanta, R., "Experiment and Analysis of Directional Effects on Radiant Heat Transfer," ASME Trans.-Jour. of Heat Transfer, Vol. 74, No. 4, Nov., 1972, pp. 459-466.
73. Wilkes, G. B. and Peterson, C. M. F., "Radiation and Convection from Surfaces in Various Positions," ASHVE Trans., Vol. 44, 1938, pp. 513-522.

APPENDIX A

LIGHTING FIXTURE SURVEY

One of the requirements of Research Project RP-231 is to conduct a survey of U.S. and Canadian lighting fixture manufacturers to obtain heat release data and to determine the frequency of installation of various types of lighting. A letter was written to explain the project and ask for a response in three areas:

1. Any information on heat release tests run on company products.
2. Any other information or references concerning heat release from lighting.
3. An opinion as to the frequency of installation of different types of lights and lighting fixtures.

This letter was sent to 130 U.S. lighting fixture manufacturers and responses were received from 16 companies. Ten of the replies were short letters explaining that the company had no access to heat release data or testing. Some of these ten letters did contain information about the frequency of installation of different lighting fixtures. Several also expressed interest in the outcome of the project. Six of the replies contained substantial information, including company heat release test data, articles concerning heat release tests, and pamphlets describing the line of luminaires the companies produced. All of the heat release testing was done using box calorimeter tests

and the results were in graphical forms following the IES recommendations. One of the companies also conducted transient "cold start" tests using a box calorimeter. These replies also contained estimates of present and near future lighting installation frequencies. In a few of the letters, it is mentioned that the U.S. Department of Commerce would be the best source of information on lighting shipments. From the reports in the Series: MA-36L, Table A1 was made.

TABLE A1
SHIPMENTS OF ELECTRIC LIGHTING FIXTURES

Quantities in 1,000 units
For the years 1973-1977

<u>COMMERCIAL & INSTITUTIONAL</u>	<u>1977</u>	<u>1976</u>	<u>1975</u>	<u>1974</u>	<u>1973</u>
INCANDESCENT					
Surface or Pendant	2,945	2,260	2,141	1,846	1,844
Recessed	9,554	6,515	6,245	7,838	7,568
MERCURY & OTHER HID	338	253	230	185	133
FLUORESCENT					
Recessed air handling	1,320	1,181	1,411	1,965	1,647
Recessed nonair handling	7,291	6,687	6,958	8,594	7,025
Striplights	8,641	8,040	7,779	8,295	7,371
Plastic wraparound	3,865	3,321	2,826	3,233	2,781
Surface and Pendant	1,117	1,257	973	1,241	1,083
<u>INDUSTRIAL</u>					
INCANDESCENT	1,271	1,267	1,409	1,150	966
FLUORESCENT	4,958	4,662	4,202	5,241	4,187
MERCURY & OTHER HID	797	620	631	827	641

Letters identical to those sent to the U.S. manufacturers were sent to 53 Canadian companies. Only three replies were received and none of these had any heat release data. In one reply, Statistics Canada is mentioned as a source of information concerning lighting sales, but when this was followed up, all of the sales figures were in dollar amounts instead of number of units. In another reply, the Canadian Standards Association and in particular the CSA Technical Committee "Tecpole" dealing with the performance of lighting equipment is said to be a good source. A member of the Technical Committee, Mr. G. E. Mulvey, of Mulvey Engineering Limited, wrote a letter with some general information as to practice in Canada. In particular, Mr. Mulvey included the paragraph:

Our general objective is to provide lighting in office occupancies of less than two watts per square foot and we have some projects where the energy input to the lighting system is close to $1\frac{1}{4}$ watts per square foot, while still maintaining a good productive visual environment. To be specific regarding sources and their components, those of us who are sensitive to energy conservation objectives are avoiding the use of incandescent sources whenever possible and relying on mercury, metal halide, or high pressure sodium for point sources, and fluorescent lamps for general office and school lighting. The standard warm white or cool white 4-foot fluorescent lamp is the preferred lamp and is usually operated from a standard ballast (rather than solid state, although these will be here soon, we hope) with voltages of 120, 240, 277, or 347 volts.

Another letter was sent to 87 architecture and engineering firms in the U.S. to obtain further information that might help decide what type of data and format would be most helpful for lighting load calculations. In this letter, the project is described and three requests are made of the firm contacted:

1. Information regarding the method used to account for lighting in heating and cooling load calculations.

2. How the present method of lighting load calculations could be improved.
3. An estimate of the frequency of installation of various types of lighting fixtures and lights.

Eighteen companies sent replies, none more than a page or two long. Only two of the companies replying state that they account for energy from lighting in any way in their heating load calculations. The others assumed that the maximum heating load could occur when the lighting is off. The methods used to calculate the cooling load caused by lights ranged from assuming all of the lighting energy goes to the room load, to following the procedures recommended by ASHRAE. Many of the companies also said they used manufacturers data to determine what percent of the lighting energy is picked up in the room and what percent in the plenum.

About 60% of the companies indicated that the methods they were using to calculate the load from lighting seemed adequate. The replies expressing dissatisfaction with present methods stated that the available analysis covers only a few types of lighting and that the fraction of the load picked up in the plenum needs more complete analysis. One reply requested that any new procedure not be more complicated than the present methods.

The estimates of the use of different light fixtures and lights followed the same general trends as shown in Table A1. In commercial and office installations, the breakdown is: fluorescent 80 to 90% with 90% recessed, the balance being high intensity discharge and a small portion incandescent. In indus-

trial installations, the breakdown is: 60 to 80% suspended fluorescent with the rest being high intensity discharge and HID usage increasing rapidly.

APPENDIX B

TOTAL RADIATION EXCHANGE FACTOR COMPUTATION

The equations used in this thesis to calculate the total exchange factors for radiation transfer were taken mostly from a short paper by Ishimoto and Bevans (31). They outline a simple matrix inversion procedure for finding script-F factors, which are related to total exchange factors by:

$$\overline{S_j S_k} = \mathcal{F}_{jk} A_j \quad (\text{B-1})$$

Ishimoto and Bevans (31) contains a few serious errors, which have been corrected in the equations that follow. The script-F expression for the net radiation transfer between two surfaces per unit time is:

$$q_{jk} = A_j \mathcal{F}_{jk} (\sigma T_j^4 - \sigma T_k^4) \quad (\text{B-2})$$

If the view factor, $\mathcal{F}_{jk}$, is the fraction of the energy emitted per unit area of surface j that goes directly to surface k and the radiosity, J_j , is the total radiation leaving unit area of surface j , both emitted and reflected, then the radiosity equation for an n surface enclosure is:

$$J_j A_j = A_j \epsilon_j \sigma T_j^4 + \rho_j \sum_{k=1}^n \mathcal{F}_{jk} J_k A_k \quad (\text{B-3})$$

If this equation is applied to all surfaces in an enclosure and expressed in matrix form, the result is:

$$\begin{bmatrix} (1-\rho_1 \mathcal{F}_{11}) \dots (-\rho_1 \mathcal{F}_{1j}) \dots (-\rho_1 \mathcal{F}_{1n}) \\ \vdots \\ (-\rho_j \mathcal{F}_{j1}) \dots (1-\rho_j \mathcal{F}_{jj}) \dots (-\rho_j \mathcal{F}_{jn}) \\ \vdots \\ (-\rho_n \mathcal{F}_{n1}) \dots (-\rho_n \mathcal{F}_{jn}) \dots (1-\rho_n \mathcal{F}_{nn}) \end{bmatrix} \begin{bmatrix} J_1 A_1 \\ \vdots \\ J_j A_j \\ \vdots \\ J_n A_n \end{bmatrix} = \begin{bmatrix} \epsilon_1 A_1 \sigma T_1^4 \\ \vdots \\ \epsilon_j A_j \sigma T_j^4 \\ \vdots \\ \epsilon_n A_n \sigma T_n^4 \end{bmatrix} \quad (\text{B-4})$$

If the value of σT_j^4 is set equal to 1.0 and all of the other temperatures are set equal to zero, then all energy in the enclosure must originate at surface j and the equations are now in the form:

$$\begin{bmatrix} (1-\rho_1 F_{11}) \dots (-\rho_1 F_{j1}) \dots (-\rho_1 F_{n1}) \\ \vdots \\ (-\rho_j F_{1j}) \dots (1-\rho_j F_{jj}) \dots (-\rho_j F_{nj}) \\ \vdots \\ (-\rho_n F_{1n}) \dots (-\rho_n F_{jn}) \dots (1-\rho_n F_{nn}) \end{bmatrix} \begin{bmatrix} J_1 A_1 \\ \vdots \\ J_j A_j \\ \vdots \\ J_n A_n \end{bmatrix} = \begin{bmatrix} 0 \\ \vdots \\ \epsilon_j A_j \\ \vdots \\ 0 \end{bmatrix} \quad (B-5)$$

Using Gauss-Jordan reduction, the radiosity values can be calculated from the above equations. Because all energy comes from surface j, the radiosity at surface k is given by:

$$q_{jk} = \alpha_k J_k A_k / \rho_k \quad (B-6)$$

Combining Eq. (B-2) where σT_j^4 equals 1.0 and T_k^4 equals zero with Eq. (B-6) and the assumptions that α_k equals ϵ_k and ρ_k equals one minus ϵ_k gives:

$$\mathcal{F}_{jk} = \epsilon_k J_k A_k / A_j (1 - \epsilon_k) \quad (B-7)$$

Because the energy emitted from surface j must be subtracted from its radiosity, Eqs. (B-6) and (B-7) become Eqs. (B-8) and (B-9) respectively for transfer from j to j.

$$q_{jj} = \alpha_j A_j (J_j - \epsilon_j) / \rho_j \quad (B-8)$$

$$\mathcal{F}_{jj} = (J_j - \epsilon_j) \epsilon_j A_j / A_j (1 - \epsilon_j) \quad (B-9)$$

Using Eq. (B-1), the total exchange factors can be expressed as:

$$\overline{S_j S_k} = \epsilon_k J_k A_k / (1 - \epsilon_k) \quad (B-10)$$

$$\overline{S_j S_j} = (J_j - \epsilon_j) A_j \epsilon_j / (1 - \epsilon_j) \quad (B-11)$$

This process, from Eq. (B-4) to Eq. (B-11) must be done n times, each with a different σT_j^4 set equal to 1.0, to calculate

the complete $\overline{S_j S_k}$ factor matrix. But this matrix depends only upon areas, emittances, and view factors, thus it has to be calculated only once for each section of the building under study. The temperature values do not affect the calculation of these factors because the product $\epsilon_j A_j \sigma T_j^4$ used in Eq. (B-4) is dimensionless.

The view factors, F_{jk} , used in this procedure were hand calculated. Most of the room surfaces are rectangles, either perpendicular or parallel to each other, and because of this, the calculation of the view factors is very manageable.

```

C-----THIS PROGRAM WILL PRODUCE HOTTEL AND SARCFIM'S TOTAL EXCHANGE
C-----FACTORS, SS(I,J), USING A GAUSS-JORDAN REDUCTION OF N SETS OF N
C-----SIMULTANEOUS RADIATION EQUATIONS. THE INPUTS NEEDED ARE:
C-----N - THE NUMBER OF SURFACES, A(I) - AREA OF SURFACE I,
C-----E(I) - EMISSIVITY OF SURFACE I, AND F(I,J) - THE VIEW FACTOR
C-----FROM SURFACE I TO SURFACE J.
      DIMENSION E(15),F(15,15),SS(15,15),Z(15,15),C(15,15),A(15),Q(15),X
      1(15)
      READ(5,1) N
      READ(5,2) (A(I),I=1,N)
      READ(5,3) (E(I),I=1,N)
      DO 20 I=1,N
20  READ(5,3) (F(I,J),J=1,N)
      1 FORMAT(I2)
      2 FORMAT(11F7.2)
      3 FORMAT(13F5.3)
      4 FORMAT(13F6.3)
      DO 100 I=1,N
      DO 110 J=1,N
110  B(I,J) = (E(I) - 1.) * F(J,I)
100  B(I,I) = E(I,I) + 1.
      DO 230 K=1,N
      CIV = B(K,K)
      S = 1./CIV
      Q(K) = S
      DO 240 J=K,N
240  B(K,J) = B(K,J) * S
      DO 250 M=1,N
      IF(M=K) 260,250,260
260  BMK = -B(M,K)
      C(M,K) = BMK
      DO 270 J=K,N
      IF((BMK .LT. .000001) .AND. (BMK .GT. -.000001)) GO TO 275
      Z = BMK * B(K,J)
      GO TO 270
275  Z = 0.
270  B(M,J) = B(M,J) + Z
250  CONTINUE
230  CONTINUE
      DO 300 I=1,N
      DO 310 J=1,N
310  X(J) = 0.
      X(I) = E(I) * A(I)
      DO 330 K=1,N
      X(K) = X(K) * Q(K)
      DO 350 M=1,N
      IF(M=K) 360,350,360
360  CONTINUE
      IF((C(M,K) .LT. .000001) .AND. (C(M,K) .GT. -.000001)) GO TO 375
      Z = C(M,K) * X(K)
      GO TO 370
375  Z = 0.
370  X(M) = X(M) + Z
350  CONTINUE
330  CONTINUE
      DO 380 J=1,N
380  SS(I,J) = X(J) * E(J)/(1. - E(J))
      SS(I,I) = (X(I) - E(I) * A(I)) * E(I)/(1. - E(I))
300  CONTINUE
      DO 900 I=1,N
900  WRITE(6,4) (SS(I,J),J=1,N)
      END

```


APPENDIX C

MEAN TEMPERATURE CALCULATIONS FOR AIR MASSES

The heat capacitances of plenum, luminaire, and duct air masses are usually so small that the Euler finite difference model is unstable when using a time interval that will suffice for other nodes. Therefore, a method was needed to either approximate the model or devise another model that is stable. The first method used was to iterate the sections of the program concerning heat transfer to these air masses several times while all other properties remain constant. Although this system was not used because it was awkward and used too much computing time, it led to another method. Because only the air temperature is changing during these iterations, the convection equations can be combined and integrated, producing a simpler and more accurate model.

If an air mass is surrounded by n surfaces at various temperatures, and the surface temperatures can be assumed to be relatively constant during a time interval, $\Delta t'$, then the temperature relation for well circulated air is:

$$dT_a = \sum_{j=1}^n dq_{ja}/C_a \quad (C-1)$$

Because energy enters the air by convection only, the equation for the differential energy coming from surface j is:

$$dq_{ja} = (T_j - T_a)h_j A_j dt \quad (C-2)$$

If these two equations are combined and rearranged, the result is:

$$dT_a = (S2 - T_a S1)dt/C_a \quad (C-3)$$

where: $S1 = \sum_{j=1}^n h_j A_j$ and $S2 = \sum_{j=1}^n h_j A_j T_j$.

Eq. (C-3) can be integrated over the time interval to give:

$$T_a^+ = \{S2 + (T_a S1 - S2)e^{-(S1\Delta t'/C_a)}\}/S1 \quad (C-4)$$

The change in the air temperature can be obtained with Eq. (C-4), but the amount of energy convected at each surface cannot be found. As the air temperature increases according to Eq. (C-3), it passes through some mean value. If this mean air temperature is used for the entire time period, $\Delta t'$, the total energy transferred to the air is the same as could be obtained using Eq. (C-4). And, because the convection transfer equations are linear, the heat transfer at each surface is also correct if this mean temperature is used for the entire time period. Eqs. (C-1), (C-2), and (C-3) can be rewritten using a mean air temperature for the whole time period and this gives:

$$T_a^+ - T_a = \sum_{j=1}^n q_{ja}/C_a \quad (C-5)$$

$$q_{ja} = (T_j - \bar{T}_a)h_j A_j \Delta t' \quad (C-6)$$

$$T_a^+ - T_a = (S2 - \bar{T}_a S1)\Delta t'/C_a \quad (C-7)$$

Solving for the mean air temperature, $\bar{T}_a$, gives:

$$\bar{T}_a = (S2/S1) - (C_a/S1\Delta t')(T_a^+ - T_a) \quad (C-8)$$

Substituting T_a^+ from Eq. (C-4) gives Eqs. (20), (21), and (22), which are restated here.

$$S3 = S2/S1 \quad (20)$$

$$Z = S1\Delta t'/C_a \quad (21)$$

$$\bar{T}_a = S3 + (T_a - S3)(1 - e^{-Z})/Z \quad (22)$$

If the plenum or luminaire is vented, a similar analysis is used. The air mass that flows through the space during $\Delta t'$ is

pictured as a series of slugs of air that each stay in the space for a proportional part of $\Delta t'$. This results in Eq. (27), which is identical to Eq. (22), except that the initial air temperature is the air temperature as it enters the space. Also, because the mass of the air passing through the space depends on $\Delta t'$, C_a becomes proportional to $\Delta t'$ and the new equation for Z is:

$$Z = S1/(0.018\text{Btu/ft}^3 \times \text{air flow rate}) \quad (\text{C-9})$$

For air flow through ducting, there is only one surface node, and comparison of S1 and S2 will show that S3 reduces to the inside surface temperature, T_i , as shown in Eq. (27).

APPENDIX D

THE LIGHTING ENERGY TRANSFER COMPUTER PROGRAM

Symbols Used in the Program

I-Input variable; E-Equation given by user describes variable.

Symbol	Meaning or Use	I	E	Units
ABSORB	Fraction of short wave energy absorbed by windows= τ_g/ϵ_g	I		-----
ADR	Inside area of return ducting	I		ft ²
ADS	Inside area of supply ducting	I		ft ²
AL(I)	Area of lamp surface I	I		ft ²
AP(I)	Area of Plenum surface I	I		ft ²
AR(I)	Area of room surface I	I		ft ²
CAPL(I)	Thermal capacitance of lamp surface I	I		Btu/°R
CAPP(I)	Thermal capacitance of a layer of plenum surface I			Btu/°R
CAPR(I)	Thermal capacitance of a layer of room surface I			Btu/°R
CAPRUG	Thermal capacitance of the carpet	I		Btu/°R
CLAMP	Thermal capacitance of lamp air			Btu/°R
CNP(I)	Thermal conductivity of plenum surface I	I		Btu/hr-ft-°R
CNR(I)	Thermal conductivity of room surface I	I		Btu/hr-ft-°R
CNRUG	Thermal conductance of the carpet	I		Btu/hr-°R
COND(I,J)	Thermal conductance from surface I to J in the lamp	I		Btu/hr-°R
CPLNM	Thermal capacitance of plenum air			Btu/°R
CPP(I)	Specific heat capacity of plenum	I		Btu/lbm

Symbol	Meaning or Use	I	E	Units
	surface I			
CPR(I)	Specific heat capacity of room surface I	I		Btu/lbm
DELT	Lightweight surface time period	I		hrs
DELTT	Counter for summing DELT			hrs
DENP(I)	Density of plenum surface I	I		lbm/ft ³
DENR(I)	Density of room surface I	I		lbm/ft ³
DT	General time period			hrs
DT1	Base value for DT	I		hrs
DUP	General dummy			-----
ELS	Short wavelength emittance of luminaire tube or bulb	I		-----
ERS	Short wavelength emittance of luminaire lens= $\tau_s + \alpha_s$	I		-----
F	Dummy for FLOWR or FLOWS			ft ³ /hr
FLOWR	Return air flow rate	I		ft ³ /hr
FLOWS	Supply air flow rate	I		ft ³ /hr
HDR	Convection coefficient inside return ducting			Btu/ft ² -hr-°R
HDS	Convection coefficient inside supply ducting			Btu/ft ² -hr-°R
HL(I)	Convection coefficient of lamp surface I		E	Btu/ft ² -hr-°R
HOS	Convection coefficient outside of window	I		Btu/ft ² -hr-°R
HP(I)	Convection coefficient of plenum surface I		E	Btu/ft ² -hr-°R
HR(I)	Convection coefficient of room surface I		E	Btu/ft ² -hr-°R
HRSON	Time lights are left on	I		hrs
I,J,K,L,M,N	General counters			-----

Symbol	Meaning or Use	I	E	Units
NDAY	Days since the run began			days
NDAYS	Days the run lasts	I		days
NLP(I)	Number of layers in plenum surface I			-----
NLR(I)	Number of layers in room surface I			-----
NL1-NL5	Lamp surface section numbers	I		-----
NOL	Luminaire option number	I		-----
NOP	Plenum option number	I		-----
NP1-NP4	Plenum surface section numbers	I		-----
NR1-NR5	Room surface section numbers	I		-----
NUMBER	Counter to write every 4th DT			-----
PB	Power to the lamp ballast		E	Btu/hr
PINPUT	Energy to the luminaire per DT			Btu
PLS	Short wavelength power from a light producing surface in lamp		E	Btu/hr
POWER	Power to a lamp, excluding ballast		E	Btu/hr
PRATE	Rated power of a luminaire	I		Btu/hr
PUD	General dummy			-----
QDR	Energy from return air to ducting			Btu
QDS	Energy from supply air to ducting			Btu
QLAMP	Energy to lamp air			Btu
QL(I)	Energy to lamp surface I		E	Btu
QPLNM	Energy to plenum air			Btu
QP(I)	Energy to plenum surface I		E	Btu
QROOM	Energy to the room air before it leaves the room space			Btu
QR(I)	Energy to room surface I		E	Btu
QTOUT	Total energy to the air while in			Btu

Symbol	Meaning or Use	I E Units
	the building section	
QX,QX1,QX2	Energy dummies	Btu
Q(I)	Energy to layer I of a surface	Btu
RAD	Short wavelength radiation through a luminaire lens	Btu
RADT	Sum of RAD for all lamps during DT	Btu
SIG	Stephan-Boltzman constant	Btu/ft ² -hr-°R ⁴
SSLS(I,J)	$\overline{S_i S_j}$ for short wavelengths in lamp I	-----
SSLT(I,J)	$\overline{S_i S_j}$ for long wavelengths in lamp I	-----
SSPT(I,J)	$\overline{S_i S_j}$ for long wavelengths in plenumI	-----
SSRS(I,J)	$\overline{S_i S_j}$ for short wavelengths in room I	-----
SSRT(I,J)	$\overline{S_i S_j}$ for long wavelengths in room I	-----
S1	Sum of total convection coefficients of all surfaces around an air volume	Btu/hr-°R
S2	Sum of(temperatures X total convection coefficients)of all surfaces around an air volume	Btu/hr
S3	S2/S1	°R
T,TT	Temperature dummies	°R
TBAR	Mean effective temperature dummy	°R
TDR	Return ducting inside surface temperature	°R
TDS	Supply ducting inside surface temperature	°R
TIME	Time after lights are turned on	hrs
TIN	Temperature of air entering room	°R
TLAMP	Temperature of lamp air	°R
TLP(I,J)	Temperature of layer J of plenum surface I	°R

Symbol	Meaning or Use	I	E	Units
TLR(I,J)	Temperature of layer J of room surface I			$^{\circ}\text{R}$
TL(I)	Temperature of lamp surface I		E	$^{\circ}\text{R}$
TOS	Temperature outside of window	I		$^{\circ}\text{R}$
TOUT	Temperature of air leaving the building section			$^{\circ}\text{R}$
TPLNM	Temperature of the plenum air			$^{\circ}\text{R}$
TP(I)	Temperature of plenum surface I		E	$^{\circ}\text{R}$
TRANS	Luminaire lens transmittance	I		-----
TROOM	Temperature of the room air	I		$^{\circ}\text{R}$
TRUG	Carpet temperature			$^{\circ}\text{R}$
TLUPLY	Supply air temperature as it enters the building section			$^{\circ}\text{R}$
T4	Dummy for 4th power temperatures			$^{\circ}\text{R}$
UDR	Return ducting thermal transmittance(includes film coefficients)	I		$\text{Btu}/\text{ft}^2\text{-hr-}^{\circ}\text{R}$
UDS	Supply ducting thermal transmittance(includes film coefficients)	I		$\text{Btu}/\text{ft}^2\text{-hr-}^{\circ}\text{R}$
VLAMP	Volume of a luminaire	I		ft^3
VPLNM	Plenum volume	I		ft^3
WP(I)	Thickness of plenum surface I	I		ft
WR(I)	Thickness of room surface I	I		ft
XLAMPS	The number of luminaires	I		-----
X(I)	Energy to outer boundary layer of surface I		E	Btu
ZDR	Return air weighting factor			-----
ZDS	Supply air weighting factor			-----
ZA,ZC,ZD, ZH,ZK,ZN, ZW	Dummies for area, specific heat, density, convection coefficient, conductivity, number of layers, and thickness respectively			Units match original properties

Numbering of Surfaces

The surfaces in the room, in the plenum, and in the luminaire each have a number of properties. To keep track of which property value goes with which surface, the surfaces are given integer numbers. These numbers are grouped into sets of surfaces that undergo similar operations in the program. And the last number in each of these sets is input into the program to designate which operations are applied to each set of surfaces. The divisions are listed below:

Room Surfaces:

1 - Ceiling

2 - Floor

3 through NR1 - Walls

NR1+1 through NR2 - Heavy furnishings

NR2+1 through NR3 - lightweight furnishings

NR3+1 through NR4 - Windows

NR4+1 through NR5 - Luminaire surfaces visible in the room

NR5 - The luminaire lens

Plenum Surfaces:

1 - Ceiling

2 - Floor

3 through NP1 - Walls

NP2 - Supply ducting

NP3 - Return ducting

NP3+1 through NP4 - Luminaire surfaces in the plenum

Luminaire Surfaces:

- 1 through NL1 - Light producing surfaces
- NL1+1 through NL3 - Surfaces receiving light
- NL4 - The luminaire lens
- NL4+1 through NL5 - Surfaces not receiving light
- NL5 - The ballast

If one set of surfaces has no members, the final number of that set is given the same number as the final number of the previous set. For example, if NR1 equals 5 and there are no heavy furnishings, then NR2 also equals 5.

Lamp and Plenum Options

Different types of ventilation arrangements are defined by the value of the Lamp Option Number (NOL) and the value of the Plenum Option Number (NOP).

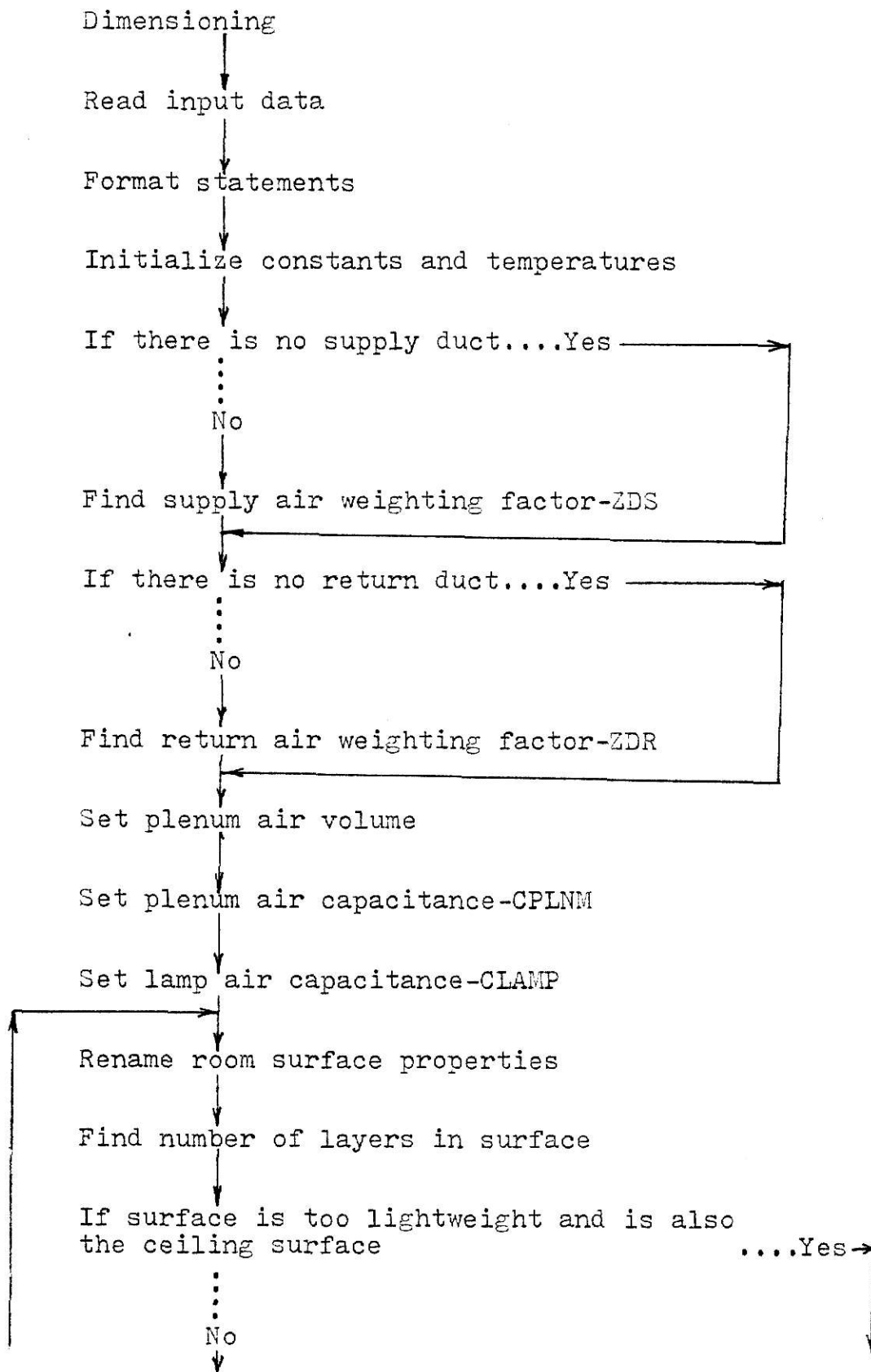
Lamp Option Numbers - NOL:

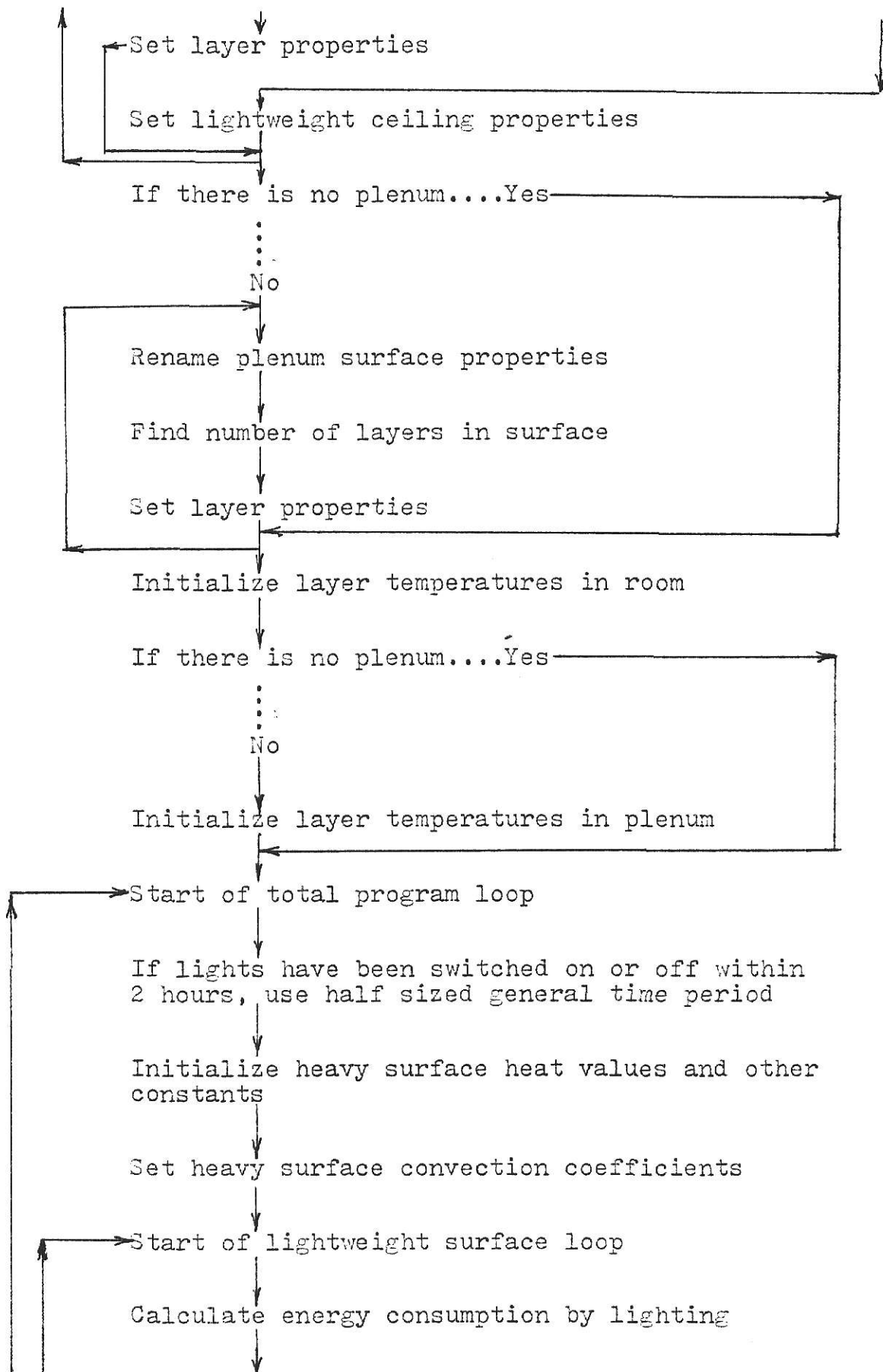
- 1 - Static lamp or luminaire
- 2 - Return through the lamp or luminaire
- 3 - Supply through the lamp or luminaire

Plenum Option Numbers - NOP:

- 1 - No plenum
- 2 - Static plenum
- 3 - Return through the plenum
- 4 - Supply through the plenum

Computer Program Flow Chart





```

graph TD
    A[Set lightweight surface convection coefficients] --> B[Initialize lightweight surface heat values]
    B --> C[Initialize lamp air temperature]
    C --> D[Find mean lamp air temperature]
    D --> E[Calculate convection to lamp air]
    E --> F[If supply luminaire, heat added to QROOM]
    F --> G[Calculate convection from lamp surfaces to room air]
    G --> H[Calculate short wavelength radiation to lamp surfaces-  
If the surface is the lamp lens, also store value at RAD]
    H --> I[Calculate long wavelength radiation exchange in the luminaire]
    I --> J[Calculate long wavelength radiation exchange from room lamp surfaces]
    J --> K{If there is no plenum....Yes}
    K -- Yes --> L[Calculate long wavelength radiation exchange from room lamp surfaces]
    K -- No --> M[Calculate long wavelength radiation exchange from lightweight plenum surfaces]
    L --> N[Calculate conduction exchange in the luminaire]
    M --> N
    N --> O[Calculate the fraction of RAD that is transmitted]
    O --> P[Calculate short wavelength radiation to room luminaire surfaces]
  
```

Set lightweight surface convection coefficients

Initialize lightweight surface heat values

Initialize lamp air temperature

Find mean lamp air temperature

Calculate convection to lamp air

If supply luminaire, heat added to QROOM

Calculate convection from lamp surfaces to room air

Calculate short wavelength radiation to lamp surfaces-
If the surface is the lamp lens, also store value at RAD

Calculate long wavelength radiation exchange in the luminaire

Calculate long wavelength radiation exchange from room lamp surfaces

If there is no plenum....Yes

...

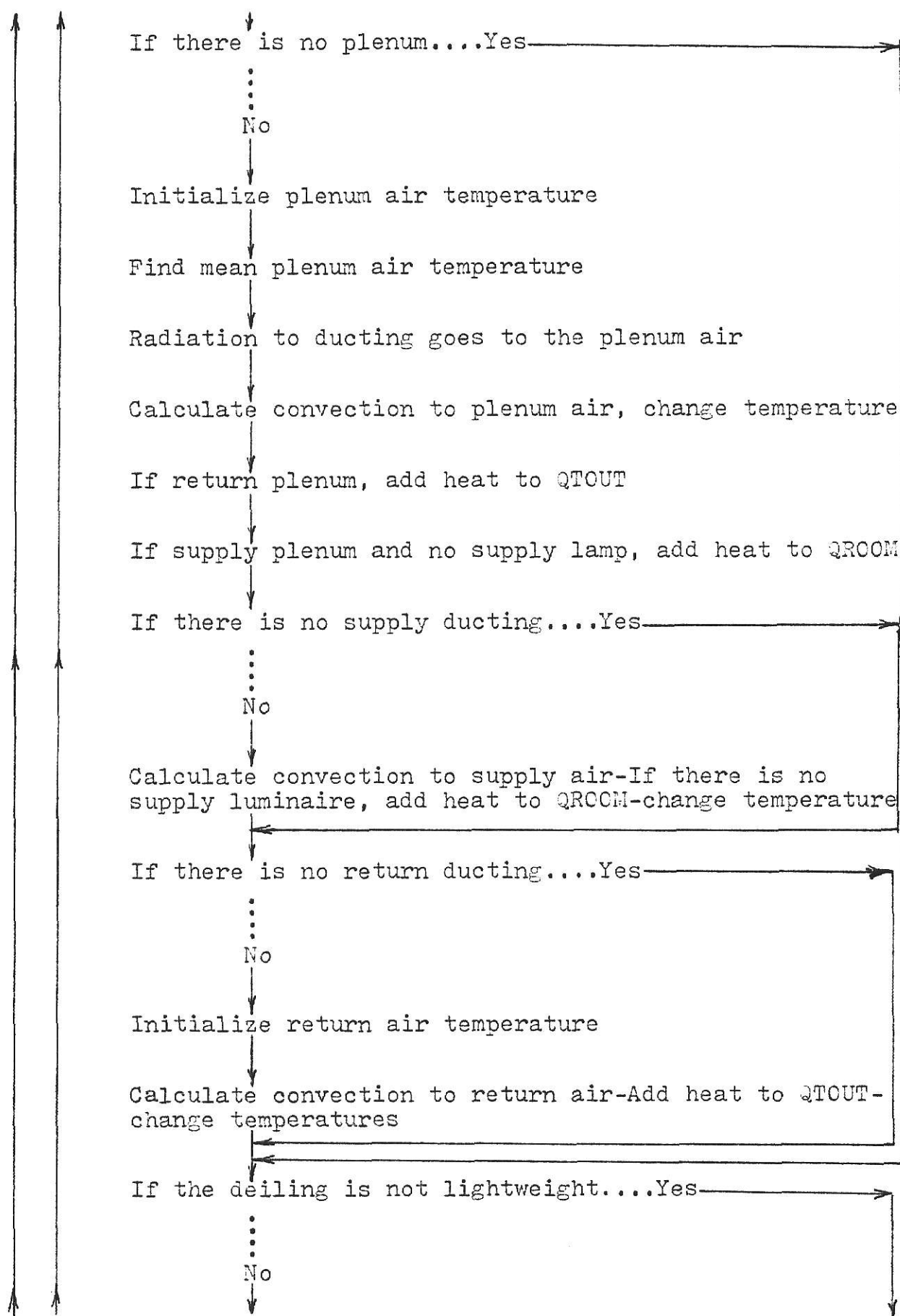
No

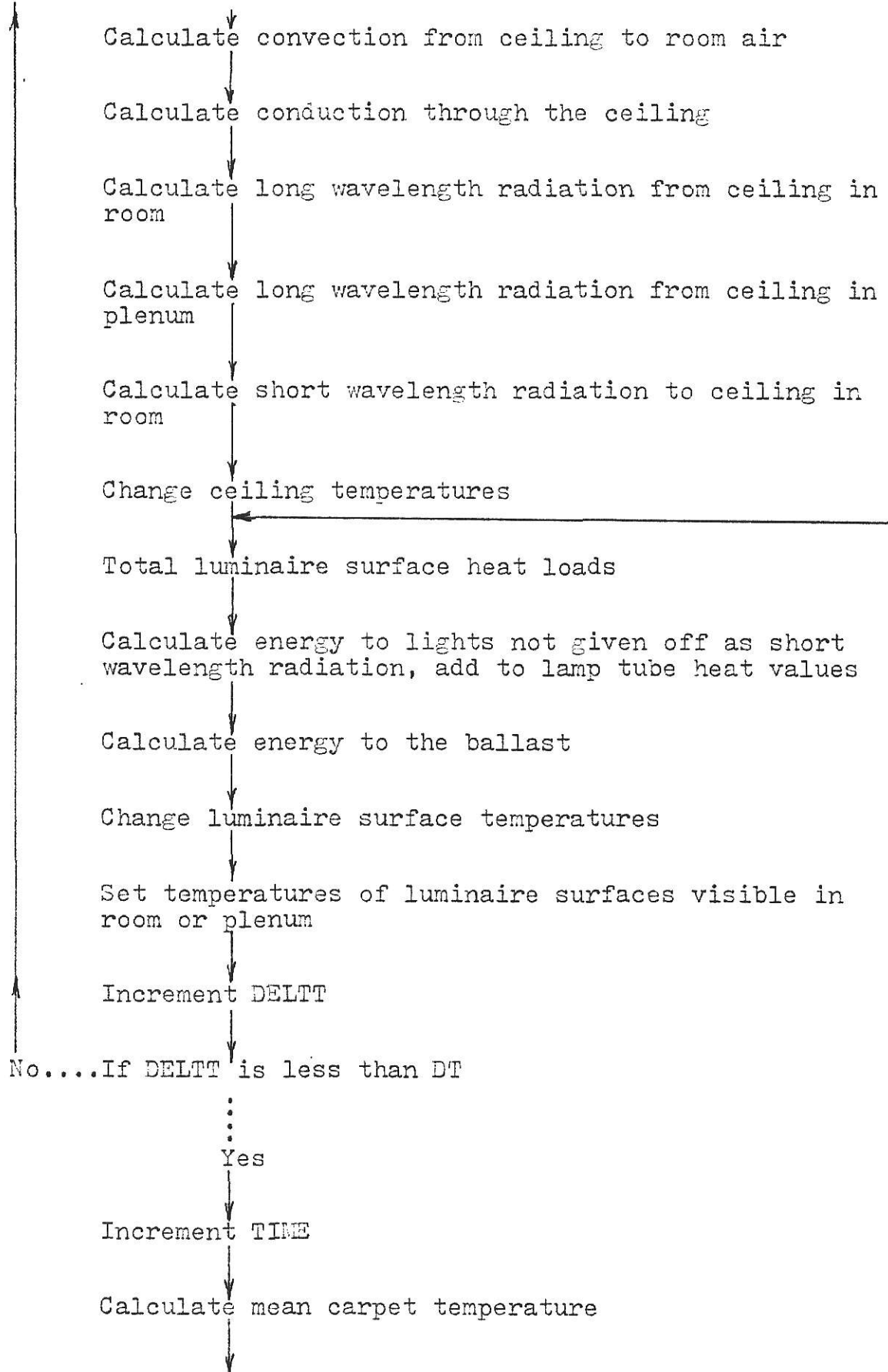
Calculate long wavelength radiation exchange from lightweight plenum surfaces

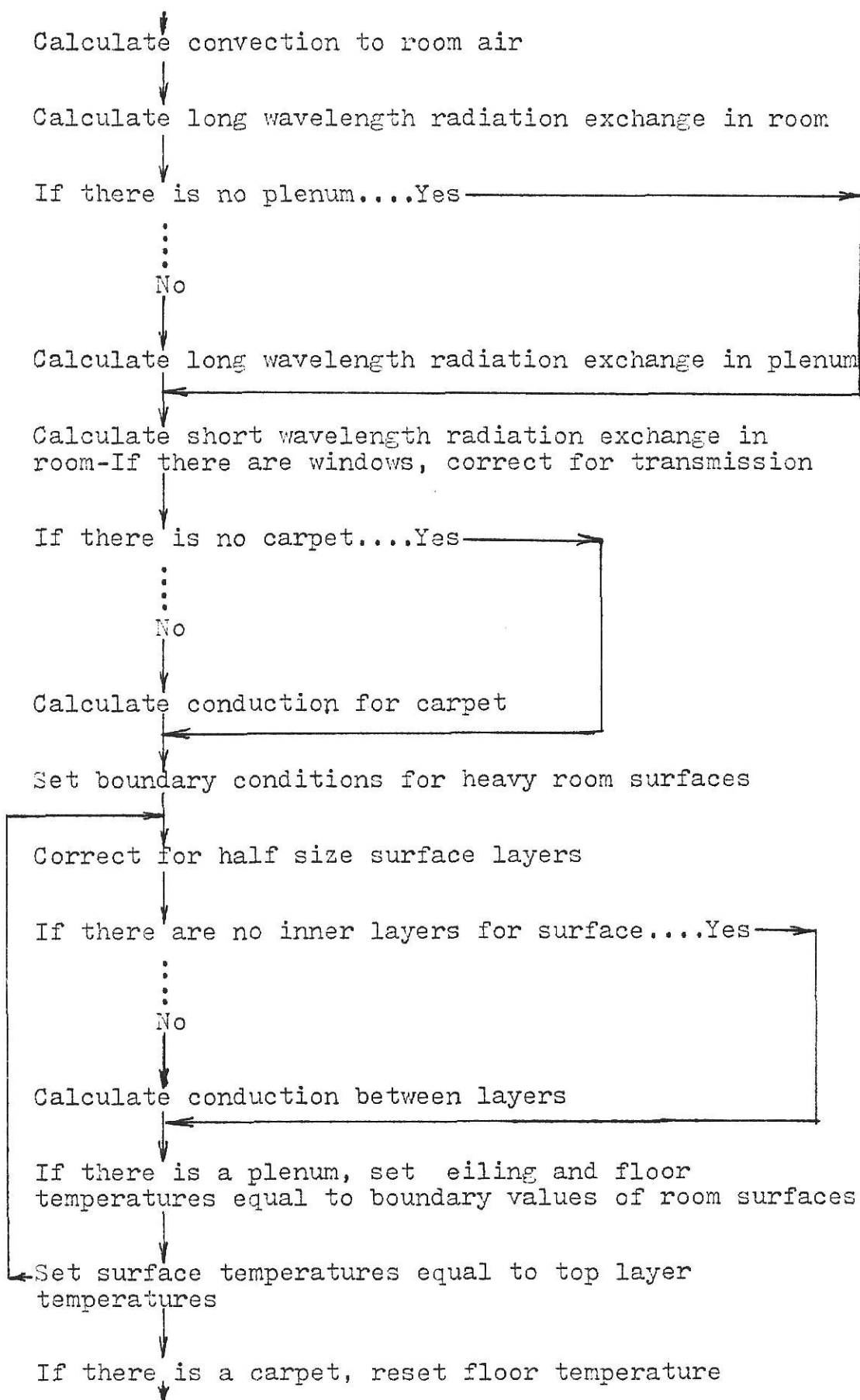
Calculate conduction exchange in the luminaire

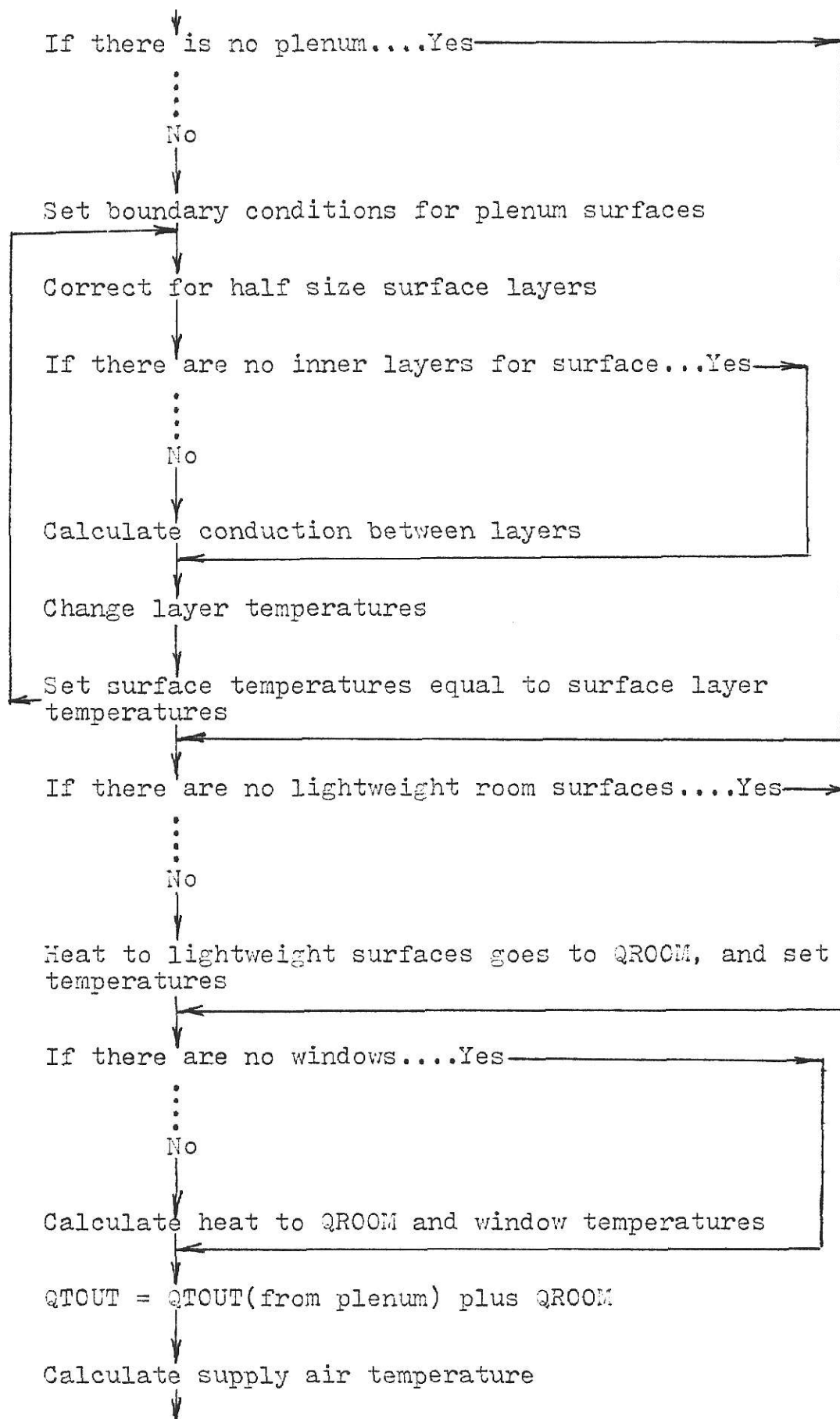
Calculate the fraction of RAD that is transmitted

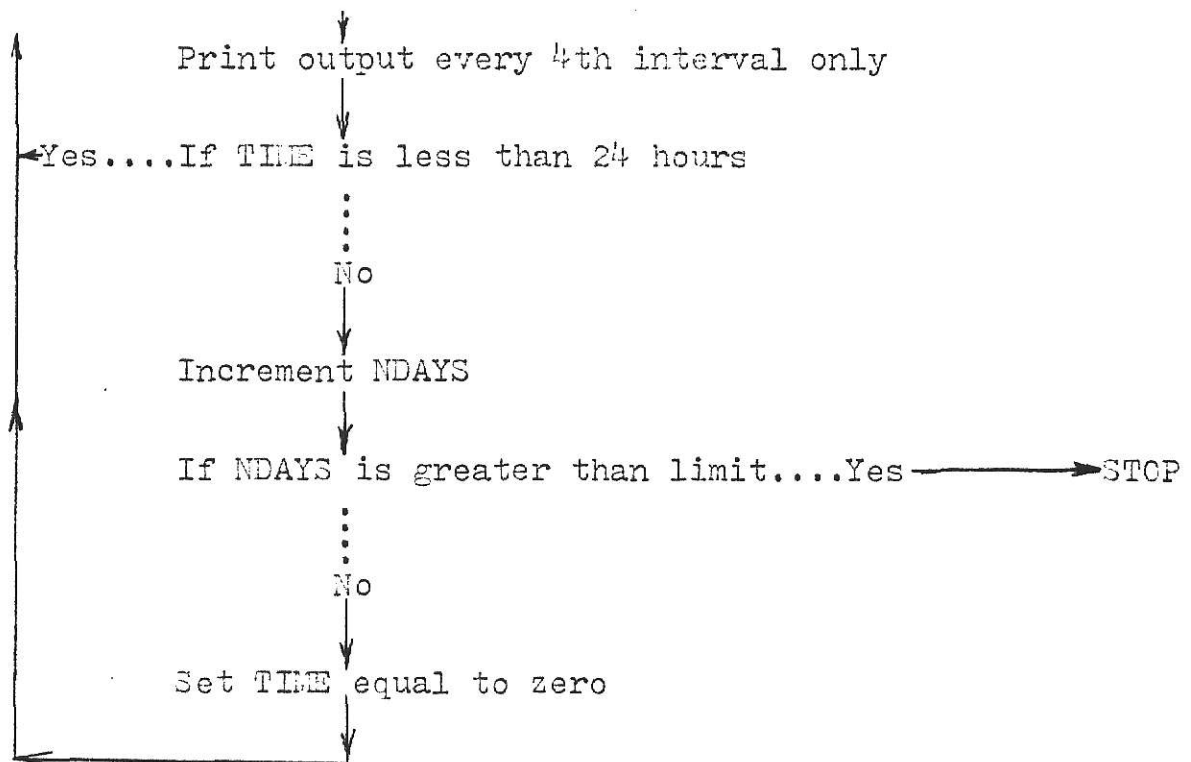
Calculate short wavelength radiation to room luminaire surfaces











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C-----THIS PROGRAM CALCULATES THE TRANSFER OF LIGHTING ENERGY.
DIMENSION AR(15),AP(15),AL(15),ALF(15),ALF(15),CCAG(15,15),CAPL(15
1),CNR(15),CNP(15),WR(15),WP(15),DEAR(15),DENP(15),CPR(15),CPP(15),
1SSRT(15,15),SSRS(15,15),SSPT(15,15),SSLT(15,15),SSLS(15,15),C(15),
1X(15),QR(15),QP(15),QL(15),TR(15),TP(15),TL(15),TLR(15,15),TLP(15,
115),HR(15),HP(15),HL(15),CAPR(15),CAPP(15)
READ(5,20) NQP,NCL,NQAYS
READ(5,20) NR1,NR2,NR3,NR4,NR5,AP1,AP2,AP3,AP4,NL1,NL2,NL3,NL4,NL5
READ(5,26) PRATE,FLCWS,FLCWR,TRCCM,TCS,FRSCN,VPLNM,VLAMP,CAPRUG,CNR
1UG
READ(5,27) UDS,UCR,ACS,ACK,TRANS,AESCFE,CT1,CELT,ERS,ELS,XLAMPS,HCS
READ(5,23) (AR(I),I=1,NR5)
READ(5,23) (AP(I),I=1,NP4)
READ(5,26) (AL(I),I=1,NL5)
DO 50 I=1,NR5
50 READ(5,25) (SSRT(I,J),J=1,NR5)
M = NR4 + 1
DO 51 I=M,NR5
51 READ(5,25) (SSRS(I,J),J=1,NR5)
DO 52 I=1,NP4
52 READ(5,25) (SSPT(I,J),J=1,NP4)
DO 53 I=1,NL4
53 READ(5,25) (SSLT(I,J),J=1,NL4)
DO 54 I=1,NL1
54 READ(5,25) (SSLS(I,J),J=1,NL4)
DO 55 I=1,NL5
55 READ(5,26) (COND(I,J),J=1,NL5)
READ(5,26) (CAPL(I), I=1,NL5)
READ(5,21) (CNR(I),I=1,NR2)
READ(5,21) (CNP(I),I=3,NP3)
READ(5,21) (WR(I), I=1,NR2)
READ(5,21) (WP(I), I=3,NP3)
READ(5,21) (DEAR(I), I=1,NR2)
READ(5,21) (DENP(I), I=3,NP1)
READ(5,21) (CPR(I), I=1,NR2)
READ(5,21) (CPP(I), I=3,NP1)
C-----HEAVY SURFACE CONVECTION COEFFICIENT ESTIMATES
READ(5,24) (HR(I),I=1,NR2),(HP(I),I=1,NP3)
20 FORMAT(20I5)
21 FORMAT(1CF10.4)
22 FORMAT(11F11.3)
23 FORMAT(15F5.1)
24 FORMAT(15F5.3)
25 FORMAT(11F7.3)
26 FORMAT(15F5.2)
27 FORMAT(12F6.3)
28 FORMAT(10F8.1)
29 FORMAT(' ')
30 FORMAT(3X,'TIME',5X,'FINPUT',4X,'CTOUT',5X,'CROOM',6X,'RAOT',5X,'?
1PLNM',5X,'TLAMP',2X,'? TC RCCM',2X,'? TCIAL')
31 FORMAT(' TEMPERATURES OF NON-LUMINAIRE RCCM AND FLENLM SURFACES')
32 FORMAT(' TEMPERATURES OF LUMINAIRE SURFACES')
C-----INITIALIZING SOME VARIABLES AND SETTING CONSTANTS.
TRANS = TRANS/ERS
SIG = 1.714E-05
NDAY = 1
NUMBER = 1
TIME = 0.
DO 100 I=1,NR5
100 TR(I) = TRCCM

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```

      DO 105 I=1,NP4
105  TP(I) = TRCCM
      DO 110 I =1,NL5
110  TL(I) = TRCOM
      TLAMP = TRCCM
      TPLNM = TRCCM
      TSUPPLY = TRCOM
      TDS = TRCCM
      TDR = TRCCM
      TGUT = TRCCM
      TIN = TRCCM
      TRUG = TRCOM
      QDS = 0.
      QDR = 0.
C-----FINDING THE SUPPLY AIR WEIGHTING FACTOR - ZDS
      IF(NP1 .EQ. NP2) GO TO 112
      N = NP2
      CUP = (AP(N) + ADS)/2.
      HDS=(1./ADS)/(1./((LDS*ADS)-(WP(N)/(CLP *CAP(N))+1./((HP(N)*AP(N))))
      PUD = HDS * ADS/(.018 * FLOWR)
      ZDS = (1. - EXP(-PLC))/PLC
112  CONTINUE
C-----FINDING THE RETURN AIR WEIGHTING FACTOR - ZDR
      IF(NP2 .EQ. NP3) GO TO 113
      N = NP3
      CUP = (AP(N) + ADR)/2.
      HDR=(1./ADR)/(1./((UDF*ADR)-(WP(N)/(CLP *CAP(N))+1./((HP(N)*AP(N))))
      PUD = HDR * ADR/(.018 * FLOWR)
      ZDR = (1. - EXP(-PLC))/PLC
113  CONTINUE
C-----FINDING THE HEAT CAPACITY OF PLENUM AIR - CPLNM
      IF(NCP .LT. 2.5) CUP = VPLNM
      IF(NCP .EQ. 3) CUP = FLOWR * DELT
      IF(NCP .EQ. 4) CUP = FLOWR * DELT
      IF(VPLNM .GT. DUP) CLF = VPLNM
      CPLNM = CUP * .018
C-----FINDING THE HEAT CAPACITY OF LAMP AIR - CLAMP
      DUP = VLAMP
      IF(NGL .EQ. 2) DUP = FLOWR * DELT
      IF(NGL .EQ. 3) DUP = FLOWR * DELT
      CLAMP = DUP * .018
C-----LAYER SIZING FOR HEAVY ROOM SURFACES
      N1 = 1
      DO 120 I=1,NR2
      ZH = HR(I)
      ZW = WR(I)
      ZA = AR(I)
      ZD = DENR(I)
      ZC = CPR(I)
      ZK = CNR(I)
C-----THE SURFACE IS DIVIDED INTO N FULL LAYERS
      N = (.5 * ZH*ZW/ZK) * (SQRT(1. + ZK*ZC*ZC/(.5 * ZH*ZH * DT1))- 1.)
C-----ONLY CEILING MAY HAVE N<1, ELSE PROGRAM MALFUNCTIONS
      IF((N .LT. .99) .AND. (I .LT. 1.5)) GO TO 118
      ZN = FLCAT(N)
      CNR(I) = ZK * ZA * ZN/ZW
      CAPR(I) = ZA * ZW * ZC * ZC/ZN
      NLR(I) = N
      GO TO 120
C-----SETTING VALUES FOR LIGHT CEILINGS

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```

118 N1 = 2
   CAPR(1) = ZA * ZW * ZC * ZC * .5
   CNR(1) = ZK * ZA/ZW
   PUD = CAPR(1)/(ZA * FF(1) + CNR(1))
   IF(PUD .GT. DELT) GO TO 120
   CAPR(1) = CAPR(1) * 2.
   CNR(1) = C.
120 CONTINUE
C----- LAYER SIZING FOR HEAVY PLENUM SURFACES
C----- BE SURE ALL NCDES ARE HEAVY ENOUGH
   IF(NCF .EQ. 1) GO TO 131
   DO 130 I = 3, NP1
   ZH = FP(I)
   ZW = WP(I)
   ZA = AP(I)
   ZC = CENP(I)
   ZK = CPP(I)
   CNP(I) = ZK * ZA * ZN/ZW
   CAPP(I) = ZA * ZW * ZI * ZC/ZN
   NLP(I) = N
130 CONTINUE
131 CONTINUE
C----- INITIALIZING LAYER TEMPERATURES
   DO 129 I = N1, NR2
   N = NLR(I) + 1
   DO 129 J = 1, N
129 TLR(I, J) = TRCCM
   IF(NOP .EQ. 1) GO TO 199
   DO 139 I = 3, NP1
   N = NLP(I) + 1
   DO 139 J = 1, N
139 TLP(I, J) = TRCCM
199 CONTINUE
C----- 199 BEGINS THE TOTAL PROGRAM LOOP
C----- DT IS HALF AS LARGE JUST AFTER POWER CHANGES
   DT = DT1
   IF(TIME .LT. 1.9) DT = .5 * DT1
   PUD = HRSCN - .1
   DUP = HRSCN + 1.9
   IF((TIME .GT. PUD) .AND. (TIME .LT. DUP)) DT = .5 * DT1
C----- INITIALIZING VALUES
   DO 150 I = 1, NR4
150 QR(I) = C.
   IF(NCF .EQ. 1) GO TO 161
   DO 160 I = 1, NP1
160 CP(I) = 0.
161 GROCM = C.
   QTOUT = C.
   RACT = 0.
   DELTT = 0.0001
   PINPUT = C.
C----- HEAVY SURFACE CONVECTION COEFFICIENTS
C----- EQUATIONS MUST BE INSERTED TO MATCH SURFACES
   VR = 20. * FLOWS/AR(2)
   VP = .2
   HR(1) = .0115 * SQRT(VR/6.) + .38
   PR(2) = .0115 * SQRT(VR/6.) + .83 * (.1 + TR(2) - TRCCM)**.12

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HR(3) = .0115 * SCRT(VR/5.) + .50 * (.1 + TR(3) - TRCCM)**.12
HR(4) = .0115 * SCRT(VR/5.) + .50 * (.1 + TR(4) - TRCCM)**.12
HR(5) = .0115 * SCRT(VR/3.) + .20 * (.1 + TR(5) - TRCCM)**.25
HP(1) = .0115 * SCRT(VP/6.) + .38
HP(2) = .0115 * SCRT(VP/6.) + .38
IF(TP(2) .LT. TPLNM) HP(2) = HP(2) + .83*(.1+TPLNM-TP(2))**.12-.38
HP(3) = .0115 * SCRT(VP/4.) + .50 * (.1+ABS(TP(3)-TPLNM))**.12
HP(4) = .0115 * SCRT(VP/6.) + .50 * (.1+ABS(TP(4)-TPLNM))**.12
200 CONTINUE
C-----200 BEGINS THE LIGHTWEIGHT SURFACE SECTION OF THE PROGRAM
C-----RESETTING THE LAMP CUTPLTS
C-----EQUATIONS MUST BE INSERTED TO SIMULATE LLMINAIRES
POWER = PRATE * (.8 + .2 * SIN((TL(1) - 520.) * .0393))
IF(TIME .GT. HRSCN) POWER = 0.
POWER = POWER/XLAMPS
PB = .2 * POWER
PT = POWER + PB
PINPLT = (PT * CELT * XLAMPS) + PINPUT
PLS = .1 * PRATE * (.1 + SIN(.357 * SCRT(TL(1) - 525.) - .894)) / (X
1LAMPS * FLCAT(NL1))
IF(TIME .GT. HRSCN) PLS = 0.
C-----RESETTING LIGHTWEIGHT SURFACE CONVECTION COEFFICIENTS
C-----EQUATIONS MUST BE INSERTED TO MATCH SURFACES
HP(5) = .0115 * SCRT(VP/3.) + .30 * (.1+ABS(TP(5)-TPLNM))**.33
HP(6) = .0115 * SCRT(VP/1.) + .30 * (.1+ABS(TP(6)-TPLNM))**.33
HP(7) = HP(6)
HP(8) = HP(6)
HP(9) = HP(6)
HP(10) = HP(6)
HL(1) = .25 * (.1 + ABS(TL(1) - TLAMP) * 4.)**.25
HL(2) = HL(1)
HL(3) = HL(1)
HL(4) = HL(1)
HL(5) = HL(1) * 1.5
HL(6) = HL(1)
HL(7) = HL(1)
HL(8) = HL(1) * .5
HL(9) = HL(1) * .5
C-----INITIALIZING LIGHTWEIGHT SURFACE HEAT VALLES
IF(N1 .EQ. 1) GO TO 207
QR(1) = 0.
QP(1) = 0.
207 K = NR4 + 1
DO 206 I=K,NR5
206 QR(I) = 0.
K = NP1 + 1
DO 208 I=K,NP4
208 QP(I) = 0.
DO 209 I=1,NL5
209 QL(I) = 0.
C-----CONVECTION EXCHANGE TO LAMP AIR
IF(NCL .EQ. 2) TLAMP = TRCCM
IF((NOL .EQ. 3) .AND. (NCP .NE. 4)) TLAMP = TIN
IF((NCL .EQ. 3) .AND. (NCP .EQ. 4)) TLAMP = TPLNM
S1 = 0.
S2 = 0.
DO 210 I = 1,NL5
CUP = HL(I) * AL(I)
S1 = S1 + CUP
210 S2 = S2 + TL(I) * CUP

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      S3 = S2/S1
      PUD = S1 * DELT/CLAMP
      TBAR = S3 + (TLAMP - S3) * (1. - EXP(-PLD))/PLD
      CLAMP = C.
      DO 215 I=1,NL5
      QX = (TBAR - TL(I)) * FL(I) * DELT * AL(I)
      QL(I) = QL(I) + QX
215  CLAMP = CLAMP - QX
      TLAMP = TLAMP + CLAMP/CLAMP
      IF(NOL .EQ. 3) QRCOM = QRCOM + (TLAMP - TSLPLY) * CLAMP
C-----CONVECTION FROM LAMP SURFACES TO FCCM AIR
      K = NR4 + 1
      DO 220 I = K, NR5
      QX1 = AR(I) * HR(I) * (TRCCM - TR(I)) * DELT
      QR(I) = QR(I) + QX1
220  QRCOM = QRCOM - QX1
C-----SHORT WAVELENGTH RADIATION EXCHANGE IN THE LAMP
      DO 230 J=1,NL4
      QX = C.
      DO 231 I=1,NL1
231  QX = QX + PLS * SSLS(I,J) * DELT/(ELS * AL(I))
C-----RAD = LIGHT ABSORBED INTO THE LENS
      IF(J .EQ. NL4) RAD = QX
230  QL(J) = QL(J) + QX
C-----LONG WAVELENGTH RADIATION EXCHANGE IN THE LAMP
      DO 245 I=1,NL4
      QX1 = C.
      T = TL(I)
      T4 = T*T*T*T
      DO 240 J=I,NL4
      IF(J .EQ. I) GO TO 240
      T = TL(J)
      QX = SSLT(I,J) * SIG * (T4 - T*T*T*T) * DELT
      QL(J) = QL(J) + QX
      QX1 = QX1 - QX
240  CONTINUE
245  QL(I) = QL(I) + QX1
C-----LONG WAVELENGTH RADIATION FROM LAMP SURFACES IN THE ROOM
      N = NR4 + 1
      DO 255 I=N,NR5
      QX1 = C.
      T = TR(I)
      T4 = T*T*T*T
      DO 250 J=1,NR5
      IF(J .GE. I) GO TO 250
      T = TR(J)
      QX = SSRT(I,J) * SIG * DELT * (T4 - T*T*T*T)
      QR(J) = QR(J) + QX
      QX1 = QX1 - QX
250  CONTINUE
255  QR(I) = QR(I) + QX1
C-----LONG WAVELENGTH RADIATION FROM LIGHTWEIGHT PLENUM SURFACES
      IF(NCP .EQ. 1) GO TO 266
      N = NP1 + 1
      DO 265 I=N,NP4
      QX1 = C.
      T = TP(I)
      T4 = T*T*T*T
      DO 260 J=1,NP4
      IF(J .GE. I) GO TO 260

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      T = TP(J)
      QX = SSPT(I,J) * SIG * DELT * (T4 - T*T*T*T)
      QP(J) = QP(J) + QX
      QX1 = QX1 - QX
260  CONTINUE
265  QP(I) = QP(I) + Q>1
266  CONTINUE
C-----CONDUCTION EXCHANGE BETWEEN LAMP SURFACES
      DO 275 I=1,NL5
        QX1 = 0.
        T = TL(I)
        DO 270 J=I,NL5
          QX = CCAD(I,J) * DELT * (T - TL(J))
          QX1 = QX1 - QX
        270  QX1 = QX1 + QX
        275  QX1 = QX1 + Q>1
C-----CORRECTING FOR LIGHT TRANSMISSION
        RAD = TRANS * RAD
        RADT = RADT + RAD * XLAMPS
        CL(NL4) = CL(NL4) - FZC
C-----SHORT WAVELENGTH RADIANT EXCHANGE BETWEEN FCCM LAMP SURFACES
        N = NR4 + 1
        PUD = XLAMPS * RAD/(ERS * AR(NR5))
        DO 277 I=N,NR5
          277  QR(I) = QR(I) + SSR5(NR5,I) * PUD
C-----PLENUM AIR SECTION
        IF(NCP .EQ. 1) GO TO 299
        F = 0.
        IF(NCP .EQ. 3) F = FLCWR
        IF(NCP .EQ. 4) F = FLCWS
        T = TSUPPLY
        IF((NCP .EQ. 3) .AND. (NCL .EQ. 1)) T = TRCCM
        IF((NCP .EQ. 3) .AND. (NCL .EQ. 2)) T = TLAMP
        TPLNM = (T * F * DELT + TPLNM * (VPLNM - F * DELT))/VPLNM
        IF((F * DELT) .GT. VPLNM) TPLNM = T
C-----FINDING THE MEAN PLENUM AIR TEMPERATURE
        S1 = 0.
        S2 = 0.
        DO 280 I=1,NP4
          DUP = HP(I) * AP(I)
          S1 = S1 + DUP
        280  S2 = S2 + TP(I) * DUP
        S3 = S2/S1
        PUD = S1 * DELT/CFLNM
        TBAR = S3 + (TPLNM - S3) * (1. - EXP(-PUD))/PUD
C-----CONVECTION TO THE PLENUM AIR
        QPLNM = 0.
        IF(NP2 .GT. NP1) QPLNM = QPLNM + QP(NP2)
        IF(NP3 .GT. NP2) QPLNM = QPLNM + QP(NP3)
        DO 282 I=1,NP4
          QX = (TBAR - TP(I)) * HP(I) * AP(I) * DELT
          QP(I) = QP(I) + QX
          QPLNM = QPLNM - QX
        282  CONTINUE
        TPLNM = TPLNM + QPLNM/QPLNM
        IF(NCP .EQ. 3) QTCUT = QTCUT + (TPLNM - TRCCM) * DELT * F * .C18
        IF((NCP .EQ. 4) .AND. (NCL .EQ. 1)) QRCOM = QRCOM + (TPLNM - T) *
          1DELT * F * .C18
C-----SUPPLY AIR CONVECTION
        IF(NP2 .EQ. NP1) GO TO 296

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T = TDS + (TSUPPLY - TDS) * ZDS
QDS = (T - TDS) * FDS * ACS * DELT
TIN = TSUPPLY - QDS/(FLWS * .018 * DELT)
IF(NGL .EQ. 3) QRCOM = QRCOM - QDS
TDS = T + (TPLNM - T) * UCS/FCS
TP(NP2) = TPLNM - (TPLNM - T) * ACS * UCS/(HF(NP2) * AP(NP2))
298 CONTINUE
C-----RETURN AIR CONVECTION
IF(NP3 .EQ. NP2) GO TO 299
IF(NCL .EQ. 2) TT = TLAMP
IF(NCL .NE. 2) TT = TRCCM
T = TDR + (TT - TDR) * ZDR
QDR = (T - TDR) * FCR * ACR * DELT
TOUT = TT - QDR/(FLWR * .018 * DELT)
QTOUT = QTOUT + (TOUT - TRCCM) * FLWR * .018 * DELT
TDR = T + (TPLNM - T) * UCR/FCR
TP(NP3) = TPLNM - (TPLNM - T) * ACR * UCR/(HF(NP3) * AP(NP3))
299 CONTINUE
C-----LIGHTWEIGHT CEILING SECTION
IF(N1 .EQ. 1) GO TO 285
C-----CONVECTION TO THE RCCM
284 T = TR(1)
CX1 = AR(1) * HR(1) * DELT * (TRCCM - T)
QROOM = QRCOM - CX1
C-----CONDUCTION THRU THE CEILING
CX2 = (T - TP(1)) * DELT * CNR(1)
CX1 = CX1 - CX2
C-----RADIATION TO THE RCCM SURFACES
T4 = T*T*T*T
DO 285 J=2,NR4
T = TR(J)
CX = SSRT(1,J) * SIG * DELT * (T4 - T*T*T*T)
CR(J) = CR(J) + CX
285 CX1 = CX1 - CX
C-----RADIATION TO THE PLENUM SURFACES
T = TP(1)
T4 = T*T*T*T
DO 286 J=2,NP1
T = TP(J)
CX = SSPT(1,J) * SIG * DELT * (T4 - T*T*T*T)
CP(J) = CP(J) + CX
286 CX2 = CX2 - CX
C-----SHORT WAVELENGTH RADIATION, HEAT TOTALS, AND INCREMENTATION
CX1 = CX1 + SSRS(NR5,1) * XLAMPS * RAD/(ERS * AR(NR5))
IF(CNR(1) .LT. .001) GO TO 288
TR(1) = TR(1) + (CR(1) + CX1)/CAPF(1)
TP(1) = TP(1) + (CP(1) + CX2)/CAPR(1)
GO TO 285
288 TR(1) = TR(1) + (CR(1) + CP(1) + CX1 + CX2)/CAPR(1)
TP(1) = TP(1)
289 CONTINUE
C-----TOTALS FOR CORRESPONDING LAMP - RCCM OR PLENUM SURFACES
C-----MATCHING EQUATIONS MUST BE INSERTED
QL(2) = QL(2) + CP(7)/XLAMPS
QL(3) = QL(3) + CP(8)/XLAMPS
QL(4) = QL(4) + CP(9)/XLAMPS
QL(6) = QL(6) + CP(6)/XLAMPS
QL(7) = QL(7) + CP(5)/XLAMPS
QL(8) = QL(8) + CP(10)/XLAMPS
C-----TOTALING THE ENERGY TO LIGHT TUBES OR BULES

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      PUC = FLOCAT(NL1)
      DO 291 I=1,NL1
    291 QL(I) = QL(I) + (POWER/PUC - PLS) * DELT
C-----ENERGY TO THE BALLAST
      QL(NL5) = QL(NL5) + PE * DELT
C-----INCREMENTING THE LAMP SURFACE TEMPERATURES
      DO 295 I=1,NL5
    295 TL(I) = TL(I) + QL(I)/CAPL(I)
C-----RESETTING THE CORRESPONDING LAMP - ROOM OR PLENUM TEMPERATURES
C-----MATCHING EQUATIONS MUST BE INSERTED
      TP(7) = TL(2)
      TP(8) = TL(3)
      TP(9) = TL(4)
      TP(6) = TL(6)
      TR(5) = TL(7)
      TP(10) = TL(8)
      DELTT = DELTT + DELT
      IF(DELTT .LT. DT) GO TO 200
C-----START OF THE NON-LUMINAIRE SECTION OF THE PROGRAM
      TIME = TIME + DT
C-----FINDING THE MEAN CATHODE TEMPERATURE
      IF(CAPRUG .LT. 1.) GO TO 312
      T = TR(2)
      PUC = C.
      CUP = C.
      DO 310 I=N1,NR4
C-----RADIANT COMPONENTS ARE LINEARIZED
    310 PUC = PUC + SSRT(I,2) * SIG * T*T*T
      DO 311 I=N1,NR4
      TT = TR(I)
    311 CUP = CUP + SSRT(I,2) * SIG * TT*TT*TT*TT
      S1 = AR(2) * HR(2) + CNRLG + PLC
      S2 = AR(2) * FR(2) * TRCCM + CNRLG * TLR(2,1) + CUP + SSRS(NR5,2)
      S3 = S2/S1
      PUC = S1 * DT/CAPFUG
      TR(2) = S3 + (T - S3) * (1. - EXP(-PLC))/PLC
      TRUG = TR(2)
    312 CONTINUE
C-----CONVECTION EXCHANGE IN THE ROOM
      DO 300 I=N1,NR2
      CX = AR(I) * HR(I) * DT * (TRCCM - TR(I))
      CR(I) = CR(I) + CX
    300 QROOM = QROOM - CX
C-----LONG WAVELENGTH RADIATION EXCHANGE IN THE ROOM
      DO 325 I=N1,NR4
      CX1 = C.
      T = TR(I)
      T4 = T*T*T*T
      DO 320 J=I,NR4
      IF(J .EQ. I) GO TO 320
      T = TR(J)
      CX = SSRT(I,J) * SIG * DT * (T4 - T*T*T*T)
      CR(J) = CR(J) + CX
      CX1 = CX1 - CX
    320 CONTINUE
    325 QR(I) = QR(I) + CX1
C-----LONG WAVELENGTH RADIATION EXCHANGE IN THE PLENUM
      IF(NOP .EQ. 1) GO TO 346
      DO 345 I=N1,NP1

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      T = TP(I)
      T4 = T*T*T*T
      DO 340 J=1,NP1
      IF(J .EQ. 1) GO TO 340
      T = TP(J)
      QX = SSPT(I,J) * SIG * DT * (T4 - T*T*T*T)
      CP(J) = CP(J) + QX
      QX1 = QX1 - QX
340  CONTINUE
345  CP(I) = CP(I) + QX1
346  CONTINUE
C-----SHORT WAVELENGTH RADIATION EXCHANGE IN THE ROOM
      PUD = RADT/(ERS * AR(NR5))
      DO 370 I=N1,NR4
      CX = SSRS(NR5,I) * PLC
C-----CORRECTING FOR TRANSMISSION THRU WINDOWS
      IF(I .GT. NR3) CX = CX * ABSCH2
370  QR(I) = QR(I) + QX
C-----HEAVY ROOM SURFACE CONDUCTION SECTION
C-----CARPET CONDUCTION
      IF(CAPRUG .LT. 1.) GO TO 404
      CX = (TLR(2,1) - TRUG) * DT * CNRLG
      QRUG = QR(2) + QX
      QR(2) = -CX
      TRUG = TRUG + QRUG/CAPRUG
      TR(2) = TLR(2,1)
404  CONTINUE
C-----SETTING BOUNDARY CONDITIONS
C-----EQUATIONS MUST BE INSERTED TO SIMULATE BOUNDARY HEAT TRANSFER
      X(1) = QP(1)
      X(2) = QP(2)
      M = NLR(3) + 1
      X(3) = .9 * DT * AR(3) * (TR(3) - TLR(3,M))
      X(4) = .9 * DT * AR(4) * (TR(3) - TLR(4,M))
C-----LAYER TEMPERATURE CHANGES
      DO 410 I=N1,NR2
      M = NLR(I)
      N = M + 1
      PUD = CNR(I)
C-----BOUNDARY LAYERS ARE HALF AS THICK, SO DOUBLE HEAT VALUES
      Q(1) = (QR(I) + (TLR(I,2) - TR(I)) * DT * PUD) * 2.
      Q(N) = (X(I) + (TLR(I,M) - TLR(I,N)) * DT * PUD) * 2.
C-----CONDUCTION AND INCREMENTATION
      IF(M .LT. 1.5) GO TO 430
      DO 420 J=2,M
      K = J - 1
      L = J + 1
420  Q(J) = (TLR(I,K) + TLR(I,L) - 2. * TLR(I,J)) * DT * PUD
430  CONTINUE
      DO 440 J=1,N
440  TLR(I,J) = TLR(I,J) + Q(J)/CAPR(I)
      IF((NCF .GT. 1.5) .AND. (I .LT. 2.5)) TP(I) = TLR(I,N)
410  TR(I) = TLR(I,1)
      IF(CAPRUG .GT. 1.) TR(2) = TRUG
C-----CONDUCTION IN HEAVY PLENUM SURFACES
      IF(NOP .EQ. 1) GO TO 465
C-----SETTING THE BOUNDARY CONDITIONS
C-----MATCHING EQUATIONS MUST BE INSERTED FOR BOUNDARY HEAT LOADS
C-----CEILING AND FLOOR ALREADY DONE IN THE ROOM SECTION
      M = NLP(3) + 1

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      X(3) = .9 * AP(3) * CT * (TR(3) - TLF(3,M))
      X(4) = .9 * AP(4) * CT * (TR(3) - TLF(4,M))
C----- LAYER TEMPERATURE CHANGES
      DO 460 I=3,NP1
      M = NLP(I)
      N = M + 1
      PUD = CNP(I)
C----- BOUNDARY LAYERS ARE HALF AS THICK, SO DOUBLE HEAT VALUES
      Q(1) = (QP(1) + (TLP(1,2) - TP(1))) * CT * PUD * 2.
      Q(N) = (X(1) + (TLP(1,M) - TLP(1,N))) * CT * PUD * 2.
C----- CONDUCTION AND INCREMENTATION
      IF(M .LT. 1.5) GO TO 480
      DO 470 J=2,M
      K = J - 1
      L = J + 1
      470 Q(J) = (TLP(I,K) + TLF(I,L) - 2. * TLP(I,J)) * CT * PUD
      480 CONTINUE
      DO 490 J=1,N
      490 TLP(I,J) = TLP(I,J) + Q(J)/CAPP(I)
      480 TP(I) = TLP(I,1)
      485 CONTINUE
C----- THE HEAT TO LIGHTWEIGHT SURFACES GOES TO THE ROOM AIR
      IF(NR2 .EQ. NR3) GO TO 510
      N = NR2 + 1
      DO 509 I=N,NR3
      TR(I) = TROOM + QP(I)/(AR(I) * HR(I) * CT)
      509 QROOM = QROOM + QP(I)
      510 CONTINUE
C----- HEAT TRANSFER THRU WINDOWS
      IF(NR3 .EQ. NR4) GO TO 515
      M = NR3 + 1
      DO 514 I=M,NR4
      QROOM = QROOM + QP(I) * .5 + (TOS - TROOM) * AR(I) * DT/(1./HR(I)
      1+1./HCS)
      514 TR(I) = (TOS * HCS + TROOM * HR(I))/(HCS + HR(I))
      515 CONTINUE
C----- TOTALS AND CLIPIT
      QTOUT = QTCUT + QROOM
C----- ENERGY TO ROOM AND TOTAL AS A %AGE OF RATED LIGHTING
      QX1 = QROOM/(PRATE * CT)
      QX2 = QTCUT/(PRATE * CT)
C----- SUPPLY TEMPERATURE CALCULATIONS ARE VARIABLE
      TSUPPLY = TROOM - QROOM/(FLOWS * CT * .C18)
      IF(NUMBER .NE. 4) GO TO 888
      WRITE(6,30)
      WRITE(6,21) TIME,FINFLT,QTCUT,QROOM,RACT,TPLNM,TLAMP,QX1,QX2
      WRITE(6,31)
      WRITE(6,22) (TR(I),I=1,NR4),(TP(I),I=1,NP3)
      WRITE(6,32)
      WRITE(6,22) (TL(I),I=1,NL5)
      WRITE(6,29)
      NUMBER = 0
      888 NUMBER = NUMBER + 1
      IF(TIME .LT. 23.9) GO TO 199
      NDAY = NDAY + 1
      IF(NDAY .GT. NDAY5) GO TO 999
      TIME = 0.
      GO TO 199
      999 STOP
      END

```

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VITA

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Thesis: A MATHEMATICAL MODEL FOR CALCULATING TRANSIENT HEATING
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A MATHEMATICAL MODEL FOR CALCULATING
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by

DANIEL JOSEPH GREEN

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AN ABSTRACT OF A MASTER'S THESIS

submitted in partial fulfillment of the
requirements for the degree

MASTER OF SCIENCE

Department of Mechanical Engineering

KANSAS STATE UNIVERSITY
Manhattan, Kansas

1980

ABSTRACT

The purpose of this thesis was to develop a mathematical model that could be used to accurately simulate the transient transfer of lighting energy. The model was built around a transient finite difference analysis of a building section that was divided into many surfaces. The calculation of radiation, convection, and conduction heat transfer was done using basic equations, but it was not oversimplified. The model was developed into a computer program and the results were compared with previously published transient and steady state experimental data. The computer predictions closely matched the experimental data, both for steady state and for transient situations. The predictions appear to be accurate enough and the model flexible enough so that more complete analysis of the transfer of lighting energy can be accomplished using this model.