PASSIVE BUILDING COOLING WITH THERMIC DIODE SOLAR PANELS

by

Juan Jose Manzano Ruiz
Ingeniero Industrial, Universidad Catolica
Caracas, Venezuela
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Signature redacted

Signature of Author...
Department of Mechanical Engineering, May, 1977

Signature redacted

Certified by...
Thesis Supervisor

Signature redacted

Accepted by...
Chairman, Department Committee on Graduate Students

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ABSTRACT

The purpose of this study is to determine the feasability of the thermic diode solar panel as a passive cooling device, from the technical and economical viewpoint. The analysis has been focused on buildings which possess large internal heat generation, like for instance, shopping centers. The study has shown that better performance can be attained in dry and warm climates. Hence, among the locations in the United States where this application can prove to be effective, we have the southwestern area of the country.

Heat transfer calculations have been carried out by means of a computer program. The input data used to run the program is hourly weather data for several American cities. The results of the simulation are analyzed to determine system's performance, after which an economical study follows. Emphasis is put to develop a general expression to represent the efficiency of the system.

The economic study has shown that current costs of panels and electric rates, as well as financing conditions, make this alternative generally unattractive, although under certain conditions the panels are economically feasible.

Thesis Supervisor: B. Shawn Buckley
Title: Associate Professor of Mechanical Engineering

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I would like to thank Professor Shawn Buckley, to whom I owe my interest in the subject, and who, as a friend, has been a source of encouragement. My second acknowledgement is due to all my friends in the Thermics group directed by Professor Buckley, for their cooperation and assistance on this study. Also, I would like to express my gratitude to Miss Barbara Rosen for the typing work of this thesis. Finally, but by no means least, I must express a word of appreciation to my wife Raiza, for her help and patience along the completion of this work.

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NOMENCLATURE

A	Non-dimensional correlation coefficient
A(I)	Wall or roof area (ft^2)
AB	Panel's outdoor face absortivity
ABSOR	Exterior absortivity of the walls
ABUIL	Building's total exterior surface (ft^2)
ALFA	Tilt angle of panels (degrees)
AFLOOR	Building's floor area (ft^2)
AREA	Face area of one panel (equal to 32 ft^2)
AZI(I)	Wall's azimuth in degrees (0°due south)
AZIMP	Panel's azimuth in degrees (0° due south)
B	Non-dimensional correlation coefficient
BETA	Ventilation flow required per unit floor area (cu ft/ft^2)
CAIR	Heat capacity of air (Btu/lbm°F) @ constant pressure
CC	Cloud cover (tenths of hemisphere)
CROOF	Roof's conducance ($Btu/ft^2h°F$)
CWALL	Wall's conductance ($Btu/ft^2h°F$)
CWAT	Heat capacity of water (Btu/lbm°F)
DCOS(I)	Factor to account for normal solar insolation
DECLI	Declination angle (radians)
DATA(K)	Each one of the input weather data
DT	Design inlet temperature gradient to rooms (°F)
EM	Panel's outdoor face emissivity
E1	Air's enthalpy entering system (Btu/lbm of dry air)

E2	Air's enthalpy after mixing with recycling air (Btu/lbm of dry air)
E5	Air's enthalpy leaving the system (Btu/lbm of dry air)
H	Exterior heat transfer coefficient (Btu/ft^{2}h°F)
HIRF	Interior heat transfer coefficient for roof (Btu/ft^{2}h°F)
HIWL	Interior heat transfer coefficient for walls (Btu/ft^{2}h°F)
HO	Combined convection and radiation heat transfer coefficient for outdoor (Btu/ft^{2}h°F)
HOAN	Horizontal sun angle (radians)
HOURS	Number of hours the building is open
HRO	Exterior radiation heat transfer coefficient (Btu/ft^{2}h°F)
KW	Panel's insulation conductivity (Btu/ft h°F)
LAMBDA	Water latent's heat (Btu/lbm)
LAT	Latitude (degrees)
LENGTH	Channel's length (ft)
L1	Largest panel's dimension (ft)
L2	Panel's width dimension (ft)
M	Panel's water flow (lbm/hr)
MA	Channel's air flow (lbm dry air/hr)
MAX	Maximum total air flow (lbm/hr)
MONTH	Month considered in the simulation
MSS	Water mass in the sensor layer (lbm)
MST	Water mass in the storage layer (lbm)
M1	Ventilation air flow (lbm/h)
PANELS	Number of panels

PCOOL	Percentage of cooling
PD	Panel's circuit pressure drop (lbf/sq.ft)/(lbm water/h)
PDROP	Total pressure drop along channels (lbf/ft^2)
PHI	Outdoor relative humidity (%)
PPERR	Number of panels per channel
POWER	Total power consumed to blow the air in the channel (kw-h)
PW	Saturated water vapor pressure (psi)
Q	Total internal heat load (Btu/h)
QAC	Cooling load of the A/C unit (Btu/h)
QAPP	Sensible and latent heat load due to appliances, lights, etc (Btu/h)
QGEN	Internal load coefficient (Btu/h ft^2)
QPEOP	Sensible & latent heat load due to people (Btu/h)
QV1	Convection heat loss at the sensor layer (Btu/h)
QV2	Convection heat gain at the storage layer (Btu/h)
QR1	Radiation heat gain at the sensor layer (Btu/h)
QR2	Radiation heat loss at the sensor layer (Btu/h)
QWALL	Total heat load through the wall and roof due to radiation and conduction (Btu/h)
RAD	Horizontal insolation flux (Btu/h ft^2)
RR	Design relative humidity inside building
SIG	Stefan-Boltzmann constant
SOALT	Solar altitude (radians)
SUM(1)	Sum of QAC
SUM(2)	Sum of QV2 (when positive)

SUM(3)	Sum of(QWALL + Q)
SUM(4)	Sum of QWALL
SUM(5)	Sum of POWER
SUM(6)	Sum of QV1
SUM(7)	Sum of QR1
SUM(8)	Sum of QV2 (when negative)
STAT	U.S. Weather Service station number
STDITT	Sum of the differences (TDI-TT)
SQR2	Sum of the heat flows QR2 at night
SQV2	Sum of the heat flows QV2 at night
TA	Outside temperature (°F)
TAVG	Average temperature TA during the month (°F)
TDESIG	Inside design temperature (°F)
TDI	Inlet temperature to the channels (°F)
THI1	Thickness of sensor layer (in)
THI2	Thickness of storage layer (in)
THI3	Thickness of insulation layer (in)
TILT	Inclination of fronts (degrees)
TIME	Number of hours of simulation
TMIN	Temperature gradient to establish the flow (°F)
TREF	Reference temperature (equal to 65°F)
TS	Weighted sensor layer temperature (°F)
TT	Weighted storage layer temperature (°F)
T1	Sensor layer temperature at top (°F)

T2	Sensor layer temperature at bottom (°F)
U	Global heat transfer coefficient in the channels ($Btu/ft^2h°F$)
V	Performance variable
WV	Wind velocity (knots)
W1	Absolute humidity at system inlet (lbm water/lbm dry air)
W2	Absolute humidity at channels inlet (lbm water/lbm dry air)
W5	Absolute humidity at system exit (lbm water/lbm dry air)
Y	Performance variable
Z	Performance variable
ZETA	Factor to account for normal solar insolation over panels

Note: a small line over a temperature signifies previous hour value (in the simulation)

$(\ )_{ann}$ singifies annual value

$(\ )_{av}$ signifies average value over the period considered.

CHAPTER 1

INTRODUCTION

It is not necessary to stress the importance of the energy problem that many countries in the world are facing today. This problem will aggravate in the coming years due to the foreseeable depletion of the principal fuels used until now. Shortages of non-renewable resources like oil, natural gas and coal, which constitute the main source of power generation, has shown the need to search for new energy alternatives.

Developed countries, like the United States, have recognized the necessity of increasing the research in these new sources of energy and also to enforce energy conservation regulations. This trend toward new energetic sources is also supported by environmental considerations. Pollution levels in industrialized countries are highest in regions near factories and big cities. Therefore, policy-makers must begin to think about how to lessen contaminates in the atmosphere, and also how to take advantage of other natural phenomena.

One way to solve this energy problem could be to decrease the power consumption spent in cooling buildings. In the United States there are regions like the South-west where some reduction in electri expenditure could be attained. This area receives high solar insolation during summer months; comfortable indoor conditions requires the use of air-conditioning (see Figure 1).

The present study intends to provide a solution for this problem through the use of the thermic diode solar panel[1] working in the cooling mode.

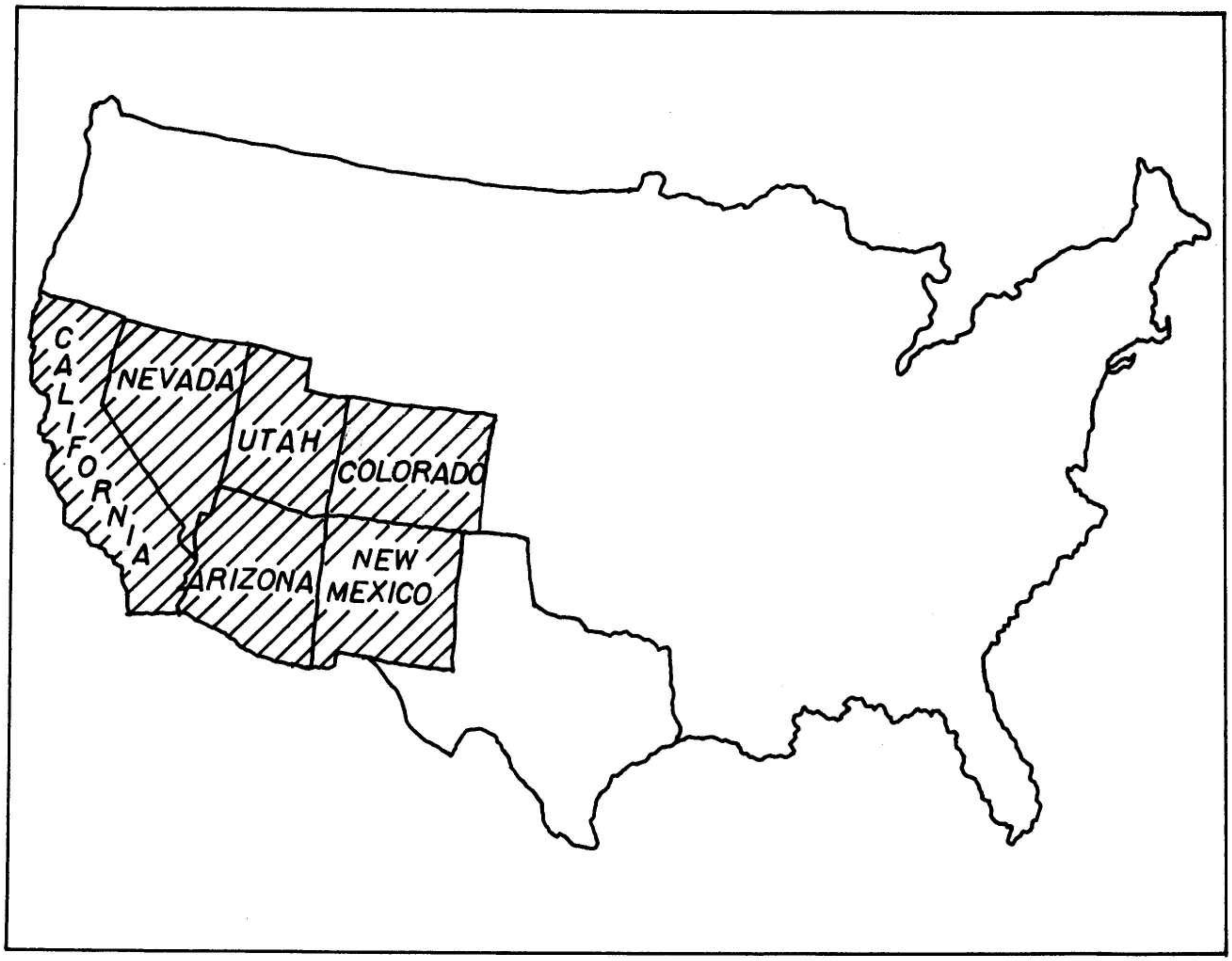

Figure 1

Regions of Interest in the United States

The main features of a thermic diode solar panel or simply, thermic panel, are shown in Figure 2. The panel is formed by two channels or layers filled with water. The water is able to circulate through these layers because they are connected at the top and at the bottom of the panel. The thin sensor layer is seen at the front. The insulation layer and the storage layer are at the back. The peculiar shape of the storage layer provides large capacity and structural stiffness at the same time, plus a large area to exchange heat with a forced-air flow behind the storage layer.

The circulation of water inside the panel is not arbitrary, but depends on natural convection or buoyancy effects due to differences in temperature between the layers. During daytime, when sunlight hits the outdoor layer (sensor layer), the water in it tends to increase in temperature to a level higher than the water stored in the interior layer (storage layer). Then, buoyancy forces due to differences in density, try to establish a flow upward in the sensor layer and downward in the storage layer. But the flow is prevented by the action of the control box. The control box connects both layers at the top of the panel; at this stage it is only needed to know that its job is the analogous to a check valve. Therefore, during daytime the valve closes, impeding the flow of warm water coming from the sensor layer and into the storage layer. Furthermore, both layers are separated by an insulation layer to reduce heat conduction losses between them. Edge losses by heat conduction are minimized by the use of an insulating frame (which also provides structural support to the system).

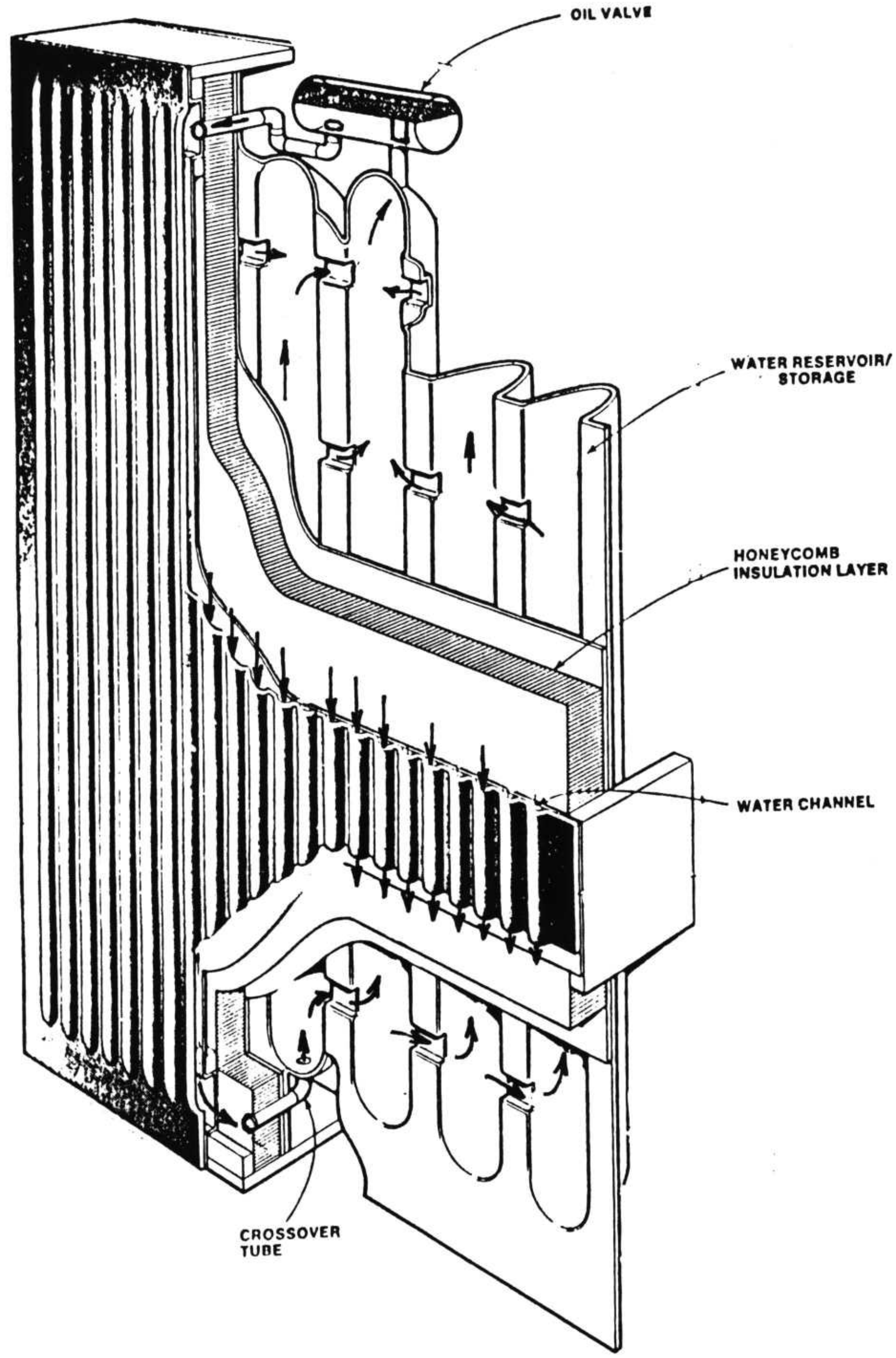

FIGURE 2

Panel's Details

At night the situation reverses. Now the water in the sensor layer can transfer energy to the environment by heat convection and radiation. The energy rejected decreases the fluid temperature in the sensor layer, making it cooler than the storage layer. From here on, the gradient in density between layers drives a flow in the panel through the control box which now works as a check valve in the positive direction. The simplified behavior of the panel in shown in Figure 3.

Good heat loss can be expected only in those regions where there is significant radiation to the sky, since heat convection is of secondary importance compared to thermal radiation. Nocturnal radiation requires that an appreciable temperature difference exists between the environment and the panel. In other words, the panel has to "see" a low temperature in the environment in order to obtain sizable heat rejection. Important factors affecting this environment temperature are cloudiness, water content of the atmosphere, and the effect of circundant elements like buildings, trees, and so forth.

The principal characteristic of the control box is its ability to respond to small pressure gradients when flow is initiated. Experimentally, as little as 1/16 inch water head difference is enough to establish the flow. If it were not so, the small buoyancy forces would not be able to compensate for the losses along the circuit. Basically the control box consists of a riser tube and a layer of mineral oil. When pressure is higher at the sensor layer because it is hotter, some oil gets into the riser balancing the pressure difference and preventing the flow. When the situation is the opposite one, the oil is pushed out of the riser by

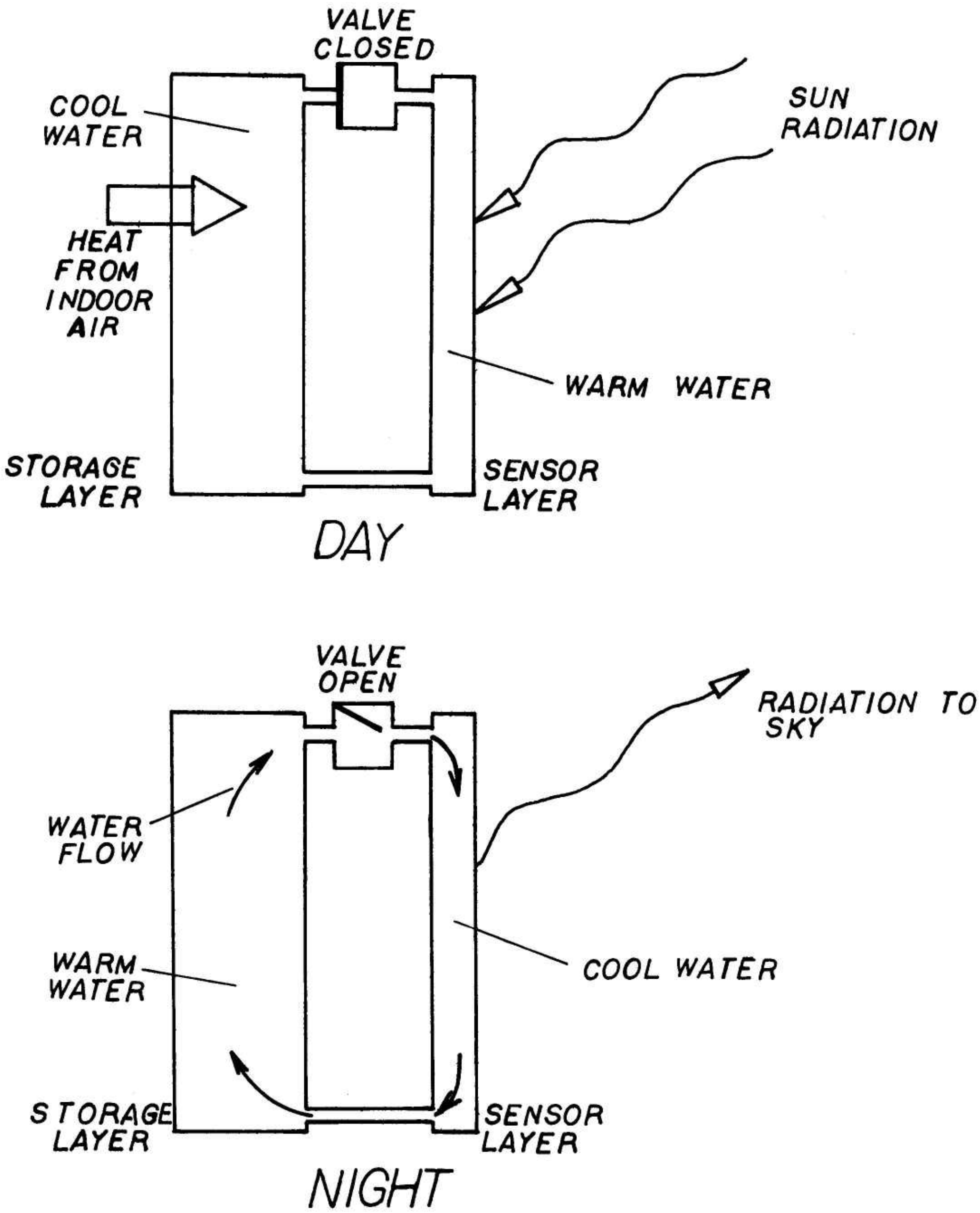

FIGURE 3

Simplified Schematic of Cooling Mode

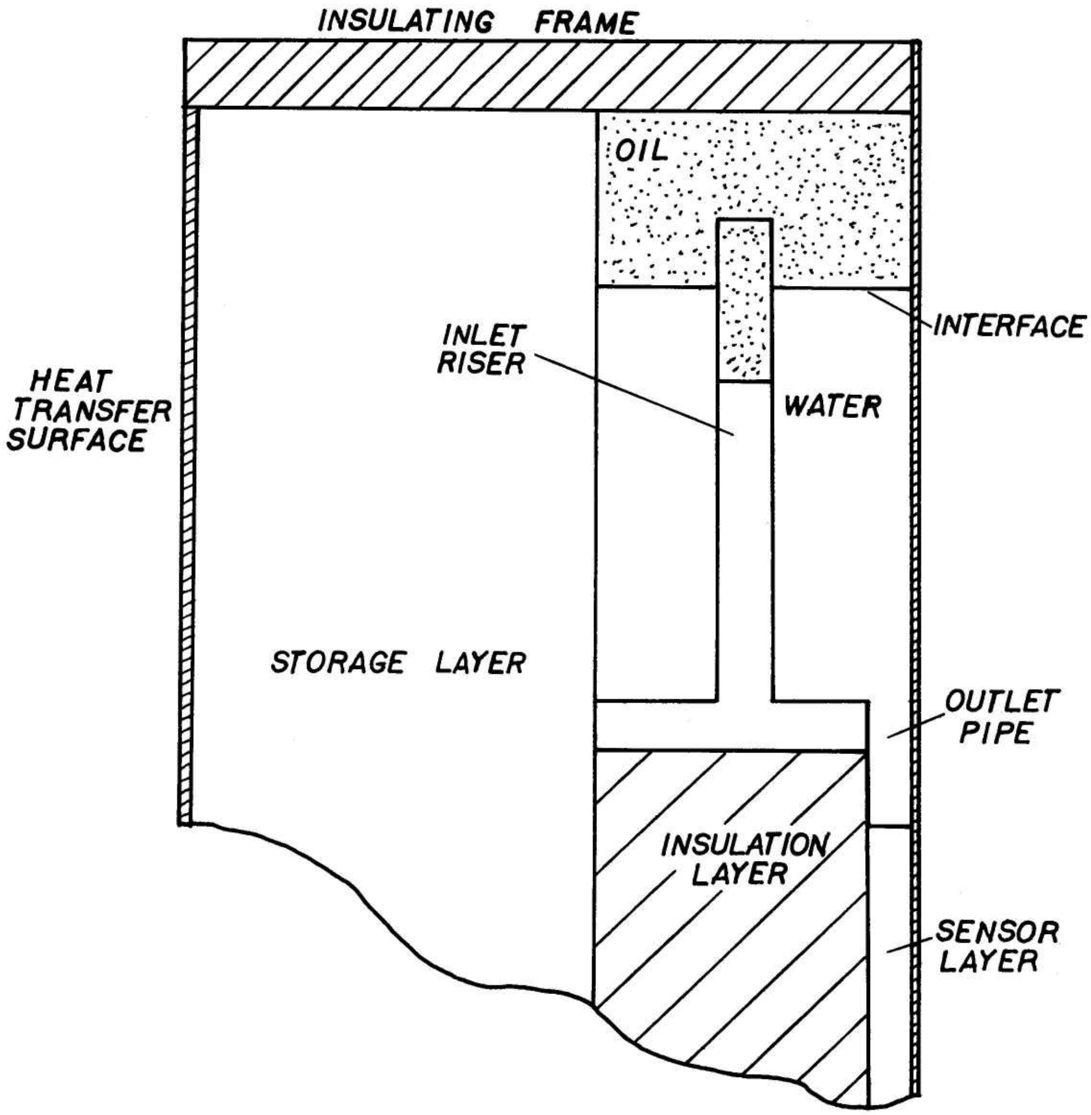

FIGURE 4

CONTROL BOX DETAILS

the warmer water coming from the storage layer to the sensor layer. Figure 4 presents a sketch of the control box.

As a conclusion we can say that cold water is stored in the storage layer to be used for cooling purposes. The water is cooled at night with the help of the control valve and the nocturnal radiation effect. Then, during daytime, the storage layer is used as a heat sink to transfer energy from the warmer forced-air stream. Air so refrigerated is ready to enter the building or an air-conditioning (A/C) unit for further cooling.

The thermic diode solar panel represents an interesting alternative to the cooling problem. It is built with inexpensive materials like aluminum, wood, paper honeycomb, fiberglass and thermoplastic resins. Its elements are very simple and rugged. It has no moving parts and constitutes a self-contained unit ready to install, and accepted by contractors and architects. Another advantage is that it replaces roof or wall elements bringing some economy.

The purpose behind this work is to find out if the thermic diode solar panel is feasible as a possible cooling device. A computer program was written to simulate the thermal dynamic behavior of a building and several panels installed upon it. Shopping centers were selected as the building type to be simulated. They are characterized by high cooling loads due both to insolation gains and to internal heat generation; they represent important consumers of electricity through A/C units. Generally speaking, shopping centers possess enough roof area to install a great number of panels. Many panels help in rejecting the heat gains of a shopping center.

The simulation program was run using real weather data; calculations are performed for each hour during summer time. Buildings with different physical dimensions as well as orientation, wall conductance and internal heat loads (electrical appliances and people contribution) can be simulated.

The results obtained from this simulation were correlated using non-dimensional parameters in order to generalize them. The correlations have been presented graphically in such a way to reduce the design calculations of buildings with thermic diode panels installed. The use of these graphs avoids computer evaluation of system efficiency, and only requires the knowledge of average environmental conditions.

Finally, an economic analysis is presented to show what savings can be obtained using the thermic diode solar panel compared to the conventional method of A/C unit.

CHAPTER 2

THE MODEL

A mathematical model was developed in order to simulate the dynamics of all the heat transfer processes taking place. This model involves several assumptions and simplifications to allow for its solution in an economical way. A very complex model not only requires large computer expense, but uncertainties in input data (parameters, weather data, etc.) reduce the accuracy of the output anyway. Therefore, the model selected was conceived to have reasonable approximations not very far from what could be obtained from more sophisticated analytical tools. Also, it had to be flexible enough to allow for many variations in parameters. Essentially, the model is a theoretical one, but some use has been made of semi-empirical relationships in order to reflect the real panel's behavior.

The mathematical model consists of two parts. The first part reproduces the thermal processes in the thermic diode solar panel. The second part simulates the behavior of the building as a whole. Obviously, both parts are coupled and the solution of the resultant system of equations has to be done simultaneously.

The assumptions included in the analysis that follows will be pointed out later in the thesis development.

Details and derivations of the equations are shown in Appendices A and B.

A. Thermic diode solar panel model

Consider Figure 5 where the principal heat flows in the panel are shown (neglecting edge losses through the panel's frame).

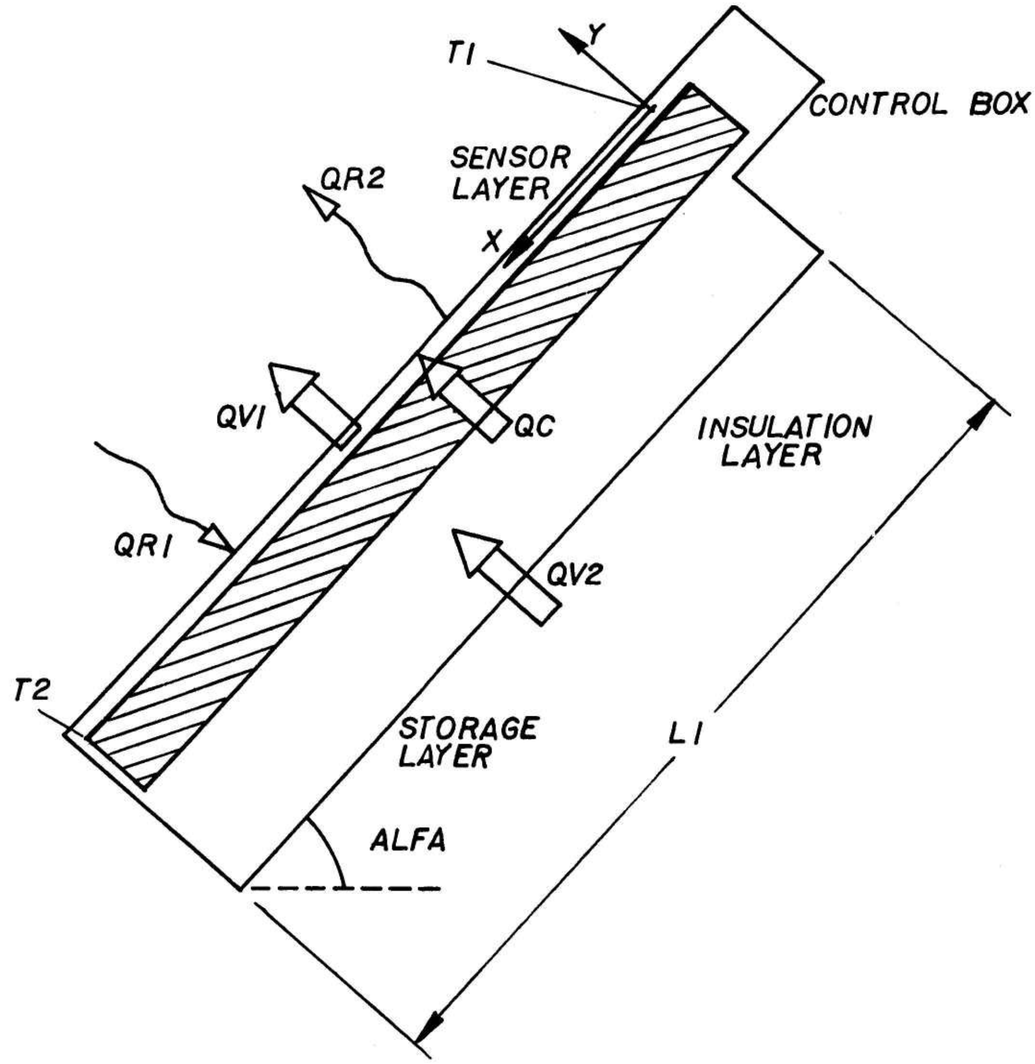

FIGURE 5

Heat Fluxes in the Panel

A.1. Daytime operation. A reasonable assumption is to consider one-dimensional heat flows through the panel. Hence, solar radiation (direct and indirect), QR1, hits the sensor layer and heats the water; it is unable to circulate because of the check valve's operation. The rise in temperature in the sensor layer allows for heat convection losses, QV1, to the nearby air layers, and also by reradiation, QR2, to the environment. In addition, there is some small leakage of heat, QC, to the storage layer by conduction through the insulation layer. These four heat flows determine the transient behavior of the sensor layer for the particular hour under consideration.

All these energy flows are evaluated using appropriate heat transfer equations.[2] The conduction heat flow, QC, is equal to the product of the conducting area, AREA; times the average temperature difference between the layers, (TT-TS); times the thermal conductivity, KW; and divided by the insulation thickness, THI3.

QC = KW * AREA * (TT-TS) / THI3 (1)

The convective loss, QV1, is the product of the heat transfer coefficient, H; times the heat transfer area, AREA; time the average temperature difference, (TS-TA).

QV1 = H * AREA * (TS-TA) (2)

The insolation flux, QR1, is equal to the product of the heat transfer area, AREA; time the total insolation flux hitting a horizontal surface, RAD; times the absorptance, AB; times the conversion factor, ZETA.

QR1 = RAD * ZETA * AREA * AB (3)

The back-radiation term, QR2, is equal to the product of the linearization variable, C1; time the area, AREA; time the average absolute temperature of the sensor layer, (TS+460).

$$QR2 = C1 * AREA * (TS+460) \tag{4}$$

The convective heat transfer coefficient, H, is calculated using:[3]

$$H = 1.09 + 0.39 * WV \quad (Btu/h\ ft^2 {}^\circ F) \tag{5}$$

The conversion factor, ZETA, represents the fraction of the solar radiation which hits the panel perpendicularly. The model assumes that most of the diffuse radiation comes from an apparent origin near the sun; in other words, the scattering of solar radiation is mostly forward scattering. This approximation is good for clear days which is generally the case for the places considered in the simulation.[3] Consequently, the conversion factor for the diffuse component of sunlight is essentially the same that for beam radiation. Further details concerning the calculation of ZETA are shown in Appendix C.

C1 is a linearization variable for equation 4. According to references 4, 5, 6 and 7, the back-radiation from the panel can be related to the temperature of the emitting surface by means of semi-empirical correlations. From that it follows:

$$C1 = (1-0.09*CC)*(1-(0.5+0.072*(PHI*0.69*PW)^{0.5}))*$$
$$EM*SIG*(\overline{TS}+460)^3 \quad (Btu/ft^2 h {}^\circ F) \tag{6}$$

The variable PW in equation 6 is evaluated using the subroutine PWS(T) with T equal to TA. The function PWS(T) is an approximation to the saturated water vapor pressure, and is given by,

$$PWS(T) = 0.6320-0.0207*T+0.0002*(T)^2 \quad (psi) \tag{7}$$

Equation 6 approximates cloud cover effects. Agreement[4] is about 20%, and does not take into account some factors like cloud structure and height, or cloud formation. Note that cloud cover data is a very rough figure and the uncertainty on this factor can be hopefully washed out by many hours of simulation. In order to solve the system of equations, the program assumes a C1 which is obtained using the previous value of TS to avoid large errors.

The sensor layer temperature for the current iteration, TS, can be found from parameters, weather data, and the previous value calculated, $\overline{TS}$. An energy balance for the sensor layer (see Figure 6), plus equations 1, 2, 3, and 4 enable us to find TS:

$$TS = (RAD*ZETA*AB+H*TA-C1*460+KW*TT/THI3+MSS*CWAT*\overline{TS}/AREA)/$$
$$(H+C1+KW/THI3+MSS*CWAT/AREA) \qquad (8)$$

Simulation results show that QC is not a very significant term. Consequently, the analysis of the storage layer can be done independently; it will be discussed later.

A.2. <u>Nighttime operation</u>. At night, when the diode valve establishes a flow there will be an enthalpy flux.

The assumptions made to get a solution are:

i) the sensor layer is in steady state conditions during each hour; at the end of which a change in heat flux and mass flow occurs that is valid for the next hour

ii) no significant heat transfer exists at the back of the panel

iii) the storage layer temperature is constant for the hour.

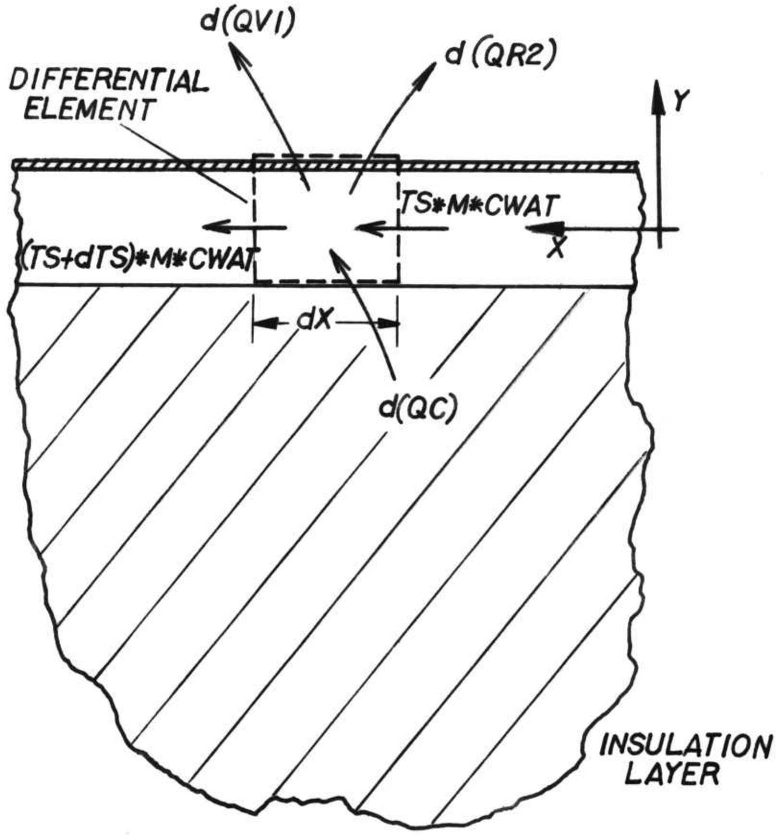

FIGURE 6

Enegy Balance for a Differential Element of the Sensor Layer

The first assumption is justified because the sensor layer is so thin that its time constant is negligible compared to the period of time analyzed. The second assumption is reasonable because there is natural convection heat transfer on the back side. This small amount of heat transfer can be neglected. Regarding the final assumption, good mixing will give an average temperature which yields a conservative approximation for the inlet temperature to the dissipator.

Using equations 1, 2 and 4, and applying the first law of thermodynamics to a differential element of the sensor layer (see Figure 6), we arrive to an expression that can be integrated to yield:

$$TS = X1/X2-(X1/X2-T1)*EXP(-X2*X/CWAT/M) \tag{9}$$

where X1 and X2 are:

$$X1 = KW*L2*\overline{TT}/THI3+H*L2*TA-C1*L2*460 \tag{10}$$

$$X2 = KW*L2/THI3+H*L2+C1*L2 \tag{11}$$

T1 is assumed to be equal to TT of the previous iteration, $\overline{TT}$, since the storage layer possesses a high thermal capacitance and there is little temperature drop across the control box. The actual value of TS is dependent on M, which in turn is a function of TS and TT, through the buoyancy forces.

If we set up a pressure head equation relating the difference in weight for both layers with the total pressure drop along the circuit, the mass flow M is determined. Thus, the resultant equation is:

$$F(M) = X3-X4*M*(1-EXP(X5/M))-100*PD*M/SIN(ALFA) = 0 \tag{12}$$

where

$$X3 = (T1-X1/X2)*L1 \tag{13}$$

$$X4 = (X1/X2-T1)*CWAT/X2 \tag{14}$$

and

$$X5 = -X2*L1/CWAT \tag{15}$$

To find M, the equation 12 must be solved by trial and error. The program uses the Newton-Raphson technique for this calculation. An initial guess for M can be found by:

$$M = -X5/(1+100*PD/SIN(ALFA)/X4) \tag{16}$$

provided that X4<30*PD/SIN(ALFA).

The variable PD is the total pressure drop per unit mass flow; an effective fluid resistance. The expression used for PD is a linearized one corresponding to an experimental resistance of the actual panel:

$$PD = 0.0039 \ (lbf/ft^2 \text{ per } lbm/h) \tag{17}$$

Knowing M, T2 follows:

$$T2 = X1/X2-(X1/X2-T1)*EXP(-X2*L1/CWAT/M) \tag{18}$$

and also $(TS)_{av}$ (valid for the hour under consideration):

$$(TS)_{av} = X1/X2-(X1/X2-T1)*(1-EXP(-X2*L1/CWAT/M))* M*CWAT/(X2*L1) \tag{19}$$

Figure 7 shows that this iterative solution is just a matching of equations obtained from equation 12, and conveniently arranged.

Initiation of flow depends on how cool the sensor layer is and how hot the storage layer is. The minimum difference necessary is about 2°F.

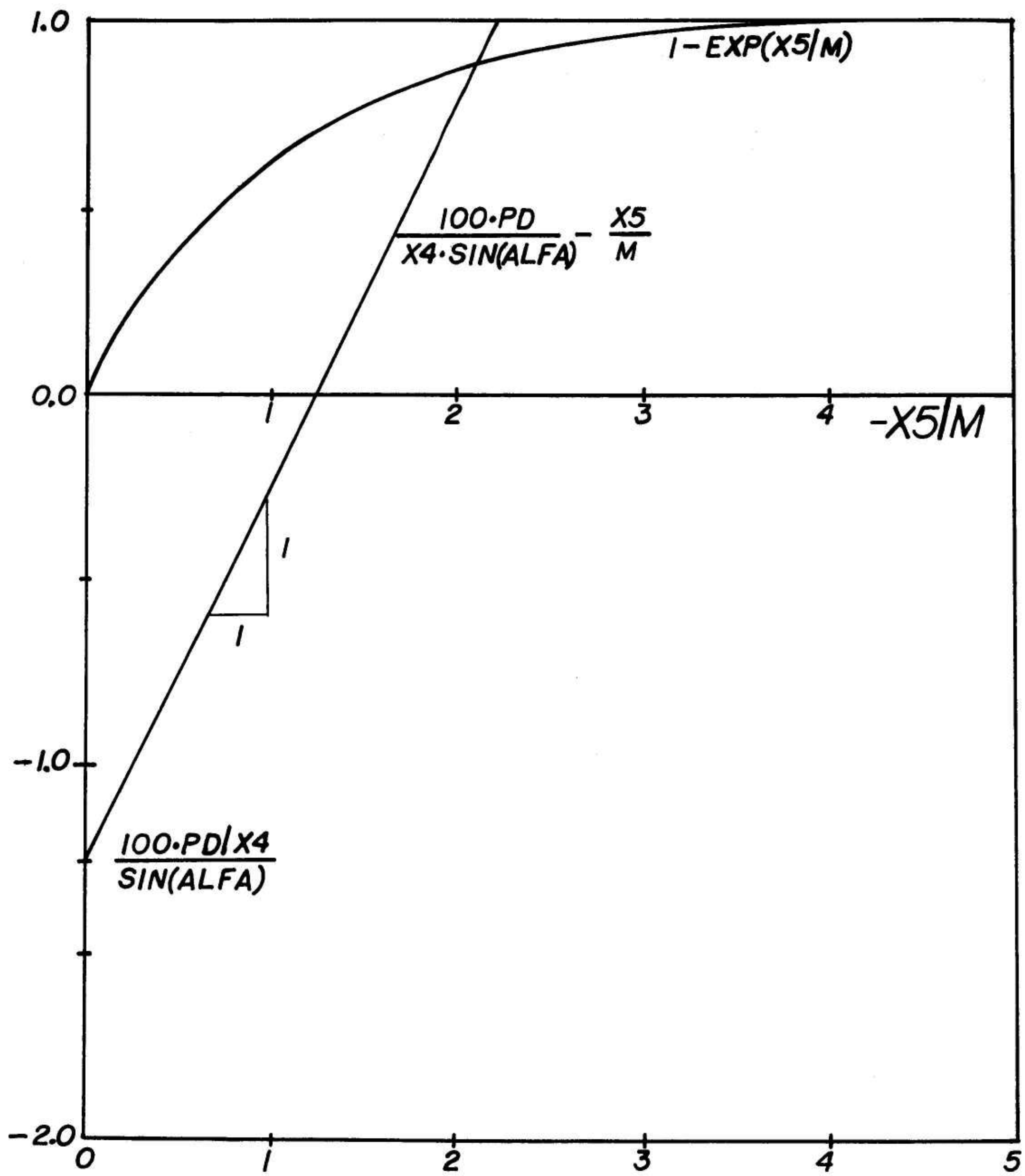

FIGURE 7

Graphic Solution of the Panel's Mass Flow

An energy balance for the storage layer determines the new value of TT:

$$\mathrm{TT = (CWAT*(T2-T1)*M+KW*AREA*TS/THI3+MST*CWAT*\overline{TT})/}$$
$$\mathrm{(MST*CWAT+KW*AREA/THI3)} \qquad (20)$$

Equations 16, 19, and 20 describe the dynamic of the panel during nighttime. They are used in conjunction with equations 1, 2, and 4 to determine the heat flows.

B. Total system model

The sketch of Figure 8 shows an overall view of the total system including the panels, A/C unit and the building itself. The "entrance" represents the mixing point of M1 and the recycled mass flow. M1 is the required[8] mass flow of fresh air; it is a function of the number of people inside the building (10 CFM/person at standard conditions of 13.33 ft^3/lb of dry air)[9]. The number of people inside the building is variable and will be considered in Chapter 3.

Rules of thumb regulate the amount of air that can be recirculated.[8] For the present analysis a maximum ratio of 2/3 of total conditioned air has been chosen. The purpose of the recirculated stream is to make the cooling process more economical because its temperature is in general lower than the outdoor air.

After the mixing process takes place, an air stream of absolute humidity W2, enthalpy E2, and mass flow rate MA (assuming uniform properties everywhere), enters the array of panels. The panels have been assumed to be installed in the roof forming triangular channels (see Figure 9). This

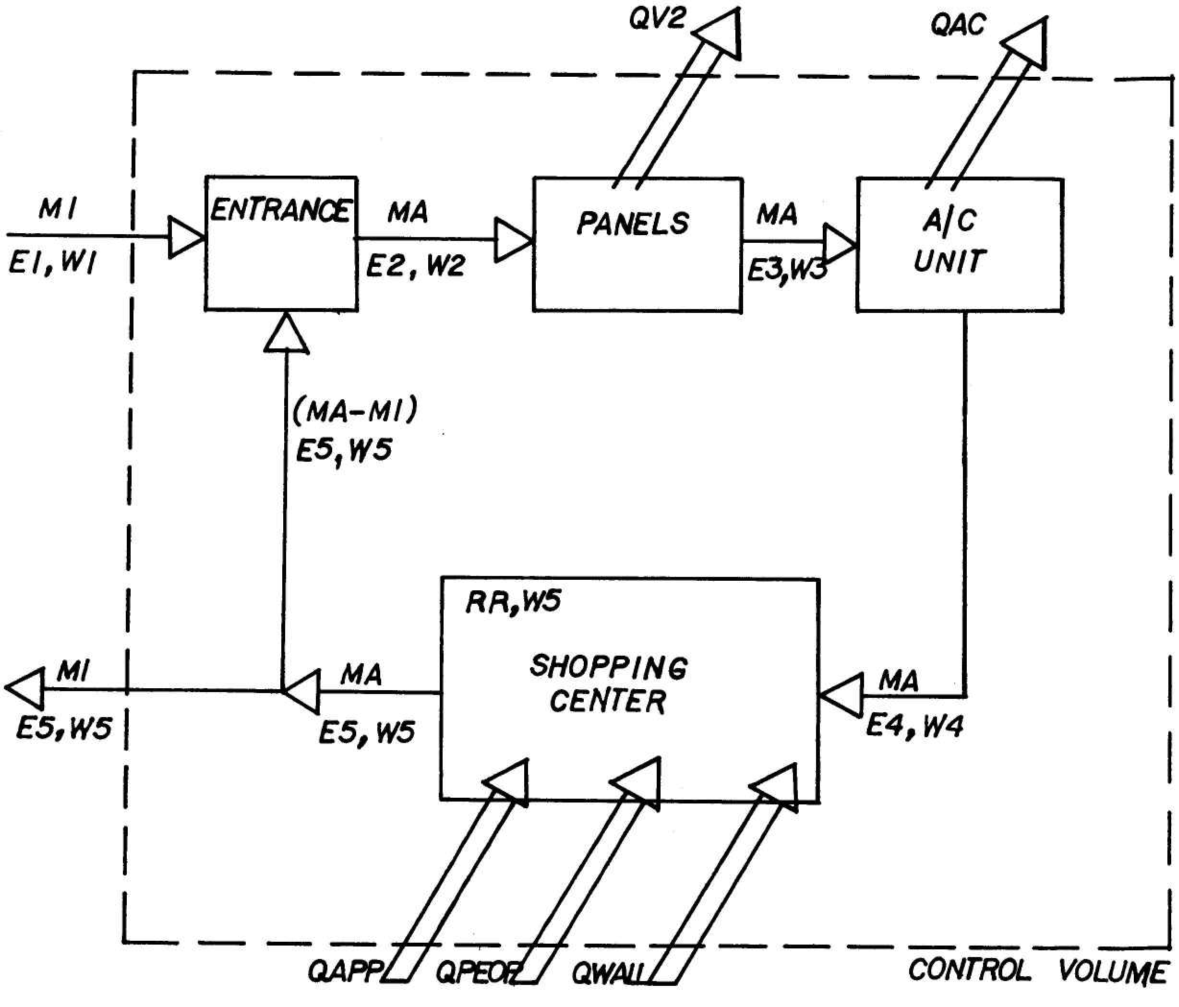

FIGURE 8

Total System Model

lets them face the sky as much as possible. The air entering these ducts will exchange energy with cool water in the storage layer through the interior surface of the panels (i.e. the "back side"). No consideration has been made in the program as to how the air is introduced to the channels by manifolding, nor for collection of the refrigerated air at the end of the channels. The computer model only determines a total pressure loss through the ducts and the associated power required. The manifolds are assumed to distribute the flow evenly through the various parallel channels. Note that after passing the panels the air enters the A/C unit. In effect the panels act as pre-chillers for the air conditioner.

The state of the air leaving the panels is described by the enthalpy, E3; the humidity, W3; and the mass flow, MA. Passing through the A/C unit, the air releases heat QAC and some water in order to meet comfort requirements before being distributed to the building. Dehumidification also represents a heat load (latent heat), and is accounted for in the overall balance. If dry air is introduced and some water is added (by means of the A/C unit) it is not considered by the model. Thus, the absolute humidity W4 is guaranteed to satisfy the comfort requirements.

The enthalpy of the air stream entering the shopping center, E4, is fixed by the humidity, W4, and the dry bulb temperature. Conventional practice[10] sets the inlet temperature 15-25°F (i.e. DT) below room temperature.

B.1. Building model. The shopping center is assumed to be a lumped element. No consideration is made considering inside air distribution or places of higher heat generation. The entire building is assumed to be at steady state[11] with dry bulb temperature TDESIG, and specified absolute humidity W5.

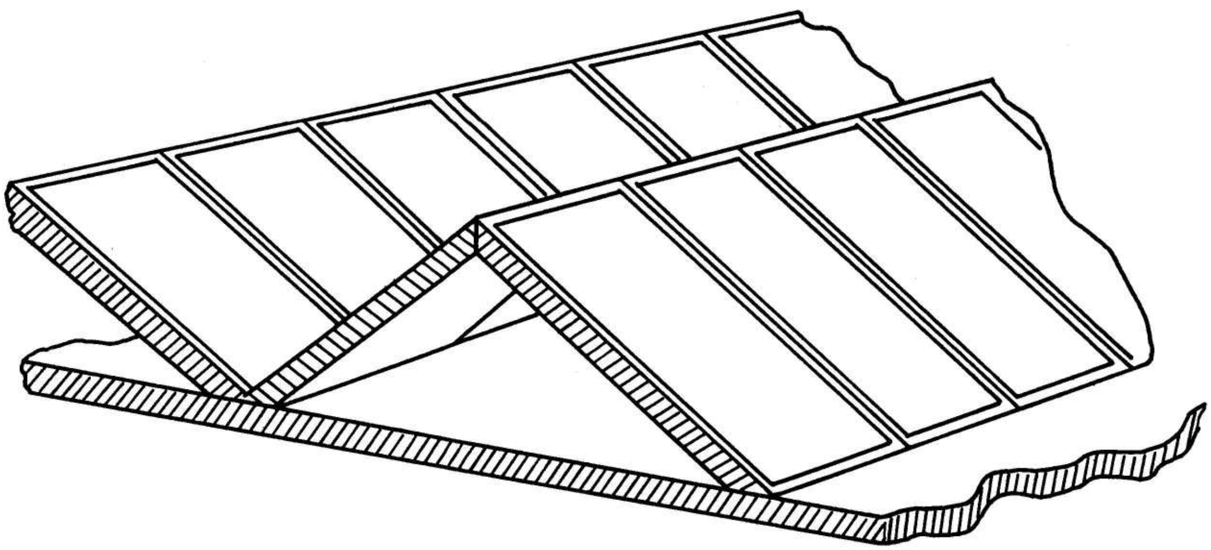

FIGURE 9

Triangular Channels Conveying Building's Air

People and electrical appliances are considered sources of internal heat generation - both, sensible and latent heat sources. Detailed calculations of these contributions are beyond the scope of this study. Therefore, simple estimates of the internal load (i.e. Q=QPEOP+QAPP), have been based on statistical data available.[8, 12, 13] In order to account for both sensible heat and moisture contribution, QPEOP and QAPP have been considered a function of the floor area. For the case of QPEOP this relation depends on the number of people per unit floor area.

Infiltration losses were initially considered. However, the simulation showed that if some care is taken to conserve energy in buildings[14] (e.g., revolving or double doors; good window sealing), the infiltration leakage is negligible compared to internal or external heat gains.

External heat gains are a consequence of the insolation and conduction heat gain. Insolation gain[9, 10, 15] is evaluated similarly to QR1. The conversion factor in this case is called DCOS(I); it is analogous to ZETA but evaluated for each wall and for the roof too. The walls' absorbtivity ABSOR and the horizontal radiation RAD is the same as used for the panel; the heat transfer area used is A(I). Not all the insolation flux hitting the walls goes into the building. Its effect is to increase the walls' surface temperature which in turn convects heat to the ambient air. Also the walls will store some energy as well as simply transmit it. To avoid the calculations[9,15] of transient phenomena in a composite slab, the model assumes that this effect can be accounted for by a slight increase in the appliances contribution.

The heat conduction contribution is due to the difference in temperature between ambient conditions and room conditions.

The combined external gain, QWALL, due to insolation and conduction is calculated by:

$$\text{QWALL} = \sum_{n=1}^{N} (\text{ABSOR*DCOS(I)*RAD+(TA-TDESIG)*HO)*A(I)/F} \qquad (21)$$

where the summation refers to all wall contributions plus roof, and HO is equal to:

$$\text{HO} = \text{H} + \text{HRO} \quad (\text{Btu/ft}^2\ \text{h°F}) \qquad (22)$$

The value of H is calculated from equation 5. The variable F in equation 21 is given by:

$$\text{F} = 1 + \text{HO} * (1/\text{HIWL} + 1/\text{CWALL}) \qquad (23)$$

The values of HRO, HIWL, and HIRF are heat transfer coefficients considered as constants in the simulation. Typical values of these coefficients were obtained from reference 9:

i) HRO = 1.05 (Btu/ft^2 h°F)

ii) HIWL= 1.74 (Btu/ft^2 h°F)

iii) HIRF= 1.37 (Btu/ft^2 h°F)

HRO represents the radiation heat transfer coefficient for the outdoor side of the walls. F is an auxiliary variable for walls and roof. For the case of the roof, equation 23 is used with the values of HIRF and CROOF replacing HIWL and CWALL, respectively.

HIWL and HIRF are the combined radiation-plus-convection indoor heat transfer coefficients for walls and roof, respectively.

CWALL represents the heat transfer conductance of the walls, and CROOF is the same for the roof.

Once the total heat load for the time period considered has been found, QWALL+Q, the amount of air (MA) required to pass through the building can be determined. This mass rate, MA, should be enough to allow for a steady state in the indoor thermal conditions, i.e. TDESIG and W5. Furthermore, assuming perfect mixing of streams implies that the exit enthalpy equals the room enthalpy, E5. This fixes MA:

MA = (QWALL+Q)/(DT*(0.24+0.444*W5))   (lbm/h)   (24)

The mass flux per channel is found by dividing the mass rate, MA, by (PANELS/PPERR). Equation 24 is the result of an energy balance for a control volume enclosing only the shopping center (see Figure 8).

Conservation of mass requires that a mass rate M1 leaves the overall system, and (MA-M1) is then the recirculated mass stream. The thermodynamic state of these streams is defined by E5, W5 and TDESIG.

B.2 <u>Channel's equations</u>. The physical meaning of equation 24 can be explained as follows. MA is the required amount of air per unit time which enters the building at a specified state and leaves at a known condition. The air stream absorbs exactly the same energy as the total cooling load of the building. Knowing MA, E2 and W2, determines the total heat exchanged with the panels (QV2):

QV2 = MA*CAIR*(TDI-$\overline{\text{TT}}$)*(1-EXP(-C4))   (Btu/h)   (25)

where

C4 = PANELS *U*AREA/(MA*CAIR)   (26)

and

$$TDI = (E2-1061*W2)/(0.24+0.444*W2) \quad (°F) \tag{27}$$

The variable U is the overall heat transfer coefficient between the storage layer and the air in the triangular duct.[16] Typical values for U are 1 - 2 Btu/h $ft^2 °F$. E2 and W2 are evaluated as it is shown in Appendix B.

B.3. <u>Storage layer equations</u>. With regard to the storage layer temperature during daytime, note that the panels arranged in the channel configuration do not have the same temperature. The channels form heat exchangers and TT at the inlet will be higher than TT at the exit. However, variations are small enough to allow a lumped model representation. In other words, the simplified model assumes that all panels in the channels are at uniform temperature TT. Nevertheless, the temperature variation along the air stream is significant and is accounted for by the model. Consequently, the value of TT will be an average one for the channel and is obtained through:

$$TT = TDI+(\overline{TT}-TDI)*EXP(U*AREA*(EXP(-C4)-1)/(C4*MST*CWAT)) \tag{28}$$

B.4. <u>Auxiliary cooling</u>. The additional cooling required is supplied by the A/C unit. In practice, the panel array never provides all the load, but some help is given by the A/C unit. The additional cooling provided is:

$$QAC = Q+QWALL-QV2+BETA*AFLOOR*(E1-E5+(W1-W5)*LAMBDA) \tag{29}$$

where QV2 is given by equation 25 and BETA is a constant which relates the number of persons per unit area and the ventilation requirements.

At night the ambient air temperature decreases. The model assumes that if this decrement in temperature is large enough, some cooling of the stored water can be achieved by blowing ambient air through the channels to help the nocturnal radiation process in cooling the storage layer.

B.5 Pumping requirements. The air flow process through the channels has associated pressure losses. The losses per channel are determined by:

$$\text{PDROP} = \text{PDR*MA}^{1.75}\text{*PPERR*LENGTH/PANELS} \quad (\text{lb/ft}^2) \qquad (30)$$

where PDR is:

$$\text{PDR} = 4.38*10^{-8}*(1+\text{COS(ALFA)})^{1.25}/\text{L1}^{4.75}/(\text{SIN(2*ALFA)})^{3} \qquad (31)$$

Details of these derivations are shown in Appendix D.

CHAPTER 3

THE DATA

So far the main features of the simulation have been presented. In addition to the equations which describe the thermal dynamics, the model requires appropriate input data. Two kinds of data are needed,

i) regional climatologic data

ii) parameters of the panel and the building

A. Weather data

The weather data is the same used by Löf and Tybout[17,18] for their economic study of solar house heating. Complete meteorological data is supplied for eight U.S. locations of different climate types during one year periods. This simulation makes use of the summer months data for two cities only.

The cities selected are:

a) Albuquerque, New Mexico. Year 1959:

The climate at this location can be described as tropical and subtropical steppe, according to the Trewartha classification.

b) Phoenix, Arizona. Year 1956:

Tropical and subtropical desert climate according to Trewartha classification.

These two cities were chosen in order to reproduce typical climatic conditions of the South-west part of the U.S., where the attention of this study is focused.

Weather data for these two cities consists of:

- station (location) code
- year of record
- month
- day
- hour
- total solar radiation on a horizontal surface, expressed in langles/h
- solar elevation angle, degrees
- extraterrestrial solar radiation, langleys/h
- wind direction, two digit code
- wind velocity, knots
- relative humidity, percentage
- sky cover, tenths of hemisphere
- visibility, miles
- ambient temperature, °F

B. Panel's data

Physical dimensions and empirical constants of the thermic panel array are described below,

- the panel is assumed to have the outdoor face covered with white paint for which the absortivity for solar radiation is about 0.3. Low temperature emissivity for the same surface is assumed to be 0.9.[10]

- tilt angle with respect to the horizontal has been set equal to 30 degrees; this lets the panels face the sky as much as possible and minimizes radiation exchange between panels. The panels are assumed to face east-west to minimize insolation.

- the physical dimensions are:
  a) length, L1, equal to 8.0 ft.
  b) width, L2, equal to 4.0 ft.
  c) sensor layer thickness, THI1, equal to 1/4 in.
  d) storage layer thickness, THI2, equal to 6.0 in.
  e) insulation layer thickness, THI3, equal to 3.0 in.
- minimum temperature gradient to start the flow, TMIN, is equal to 2°F.
- the linearized fluid resistance, PD, is:

  PD = 0.0039 (lbf/ft^2)per(lbm/h)

C. Internal load parameters

Statistical information[12] concerning the amount of people per unit area has been associated with the total (i.e., sensible plus latent) heat gain. The model assumes a total gain[9] of 450 Btu/h per person. Similarly, the total estimated heat gain from appliances[12,13] is related to electricity consumed and also to the floor area. In consequence, both internal contributions can be combined and ultimately related to the total floor area through a proportionally coefficient, QGEN. The coefficient QGEN is a very important parameter because it reflects, in general, the largest part of the total cooling load. Typical values of QGEN are in the range 2-5 Btu/h ft^2 of floor area.

D. Building parameters

Among the shopping center's parameters we have the absortivity of the walls which are assumed to be painted with a light color resulting in ABSOR equal to 0.30.

Thermal conditions for composite walls[9], CWALL, and roof, CROOF, range between 0.20 and 0.60 Btu/h ft^2°F for the simulation study.

Operations hours for the shopping centers considered are from 9 a.m. to 10 p.m., or a total of 13 hours per day; seven days a week.

Several building sizes have been considered. In Table I some of their dimensions are shown. Physical size and orientations have been selected trying to cover typical designs ranging from small to large buildings.

In addition, for each design several cases of panel arrangements and number of panels have been examined. Some of these cases are demonstrated in Table II.

Concerning design comfort conditions, they have been chosen[9, 10, 11] as

a) TDESIG = 75°F (dry bulb)

b) RR = 50% (relative humidity)

c) DT = 20°F (inlet temperature gradient)

These design parameters sensibly modify the performance of the system. It is assumed that no significant variation in their values are allowed by means of the proper automatic controls.

Typical values of the variable BETA (proportionality constant for ventilation requirements) range between 0.09 - 0.12 lbm/h ft^2.

E. Other physical parameters

There are other constants which are required by the model, among them are;

a) thermal capacity of air, CAIR, equal to 0.24 Btu/lb°F

TABLE I

PHYSICAL DIMENSIONS OF THE BUILDINGS

BUILDING	FLOOR AREA (ft^2)	LENGTH (ft)	WIDTH (ft)	HEIGHT (ft)	WALL AREA (NORTH & SOUTH FRONTS) (ft^2)	WALL AREA (EAST & WEST FRONTS) (ft^2)
#1	64,000	320	200	12	2,400	3,840
#2	125,000	500	250	14	3,500	7,000
#3	400,000	830	480	15	7,200	12,495
#4	830,000	1400	593	15	8,895	21,000

Comments:

1. For simplicity the buildings have been assumed to be square or rectangular in base.
2. Fenestration area is assumed to be 10% of the total wall area.
3. Buildings are considered to have four walls due south, east, north, and west, respectively.

TABLE II

PANELS' ARRANGEMENT IN THE BUILDINGS

BUILDING	ROOF AREA	% OF AREA COVERED	No. PANELS (TOTAL)	No. PANELS PER CHANNEL	CHANNEL'S LENGTH
#1	64,000 ft^2	87%	2016	144	288 ft
		70%	1608	134	268 ft
		61%	1400	100	200 ft
		45%	1032	86	172 ft
# 2	125,000 ft^2	89%	4032	224	448 ft
		70%	3162	186	372 ft
		49%	2208	138	276 ft
		30%	1350	90	180 ft
#3	400,000 ft^2	95%	13668	402	804 ft
		75%	10800	360	720 ft
		50%	7200	240	480 ft
		40%	5750	230	460 ft

b) thermal capacity of water, CWAT, equal to 1 Btu/lb°F

c) water's latent heat, LAMBDA, equal to 1050 Btu/lb

d) Stefan-Boltzmann constant, SIG, equal to $0.1713*10^{-8}$ Btu/h $ft^2 °R^4$

e) latitude, LAT, equal to 34 degrees north

CHAPTER 4

SIMULATION RESULTS

A. Correlation

The results of over 250 monthly simulations for the buildings considered in Table I were analyzed to be represented by generalized, non-dimensional parameters. The idea behind this representation is to find a correlation which uses simply determinable parameters in order to predict the system performance. The correlation can be found by substituting equation 26 into equation 25, and then determining the average over the time of the resultant expression. The average is obtained by integrating the equation over a period of time ΔT, long enough to decrease the importance of capacitance effect terms compared to other energy transfer terms[19]. A time span of one month is reasonable to accomplish this objective. Details of the results shown in this chapter are presented in Appendices F and G.

The percentage of cooling attained, PCOOL, is defined as the ratio of the average panel heat transfer (QV2), divided by the average heat load of the building (Q+QWALL). The correlation found for PCOOL is:

$$PCOOL = A/((Z*V)^{-1}-B) \qquad (32)$$

The coefficients A and B are parameters calculated by the program. These parameters are a function of the location and the month; they can be obtained from Table III.

The variable Z is given by:

$$Z = (1 - EXP(-C4))/C4 \qquad (33)$$

TABLE III

COEFFICIENTS A AND B OF THE FACTOR X

LOCATION	JUNE		JULY		AUGUST	
	A	B	A	B	A	B
ALBUQUERQUE	0.547	0.851	0.427	0.885	0.459	0.927
PHOENIX	0.336	0.715	0.153	0.835	0.277	0.746

which can be read from Figure 10. The variable C4 can be expressed as

$$C4 = U*AREA*PANELS*XX/(CAIR*(Q+QWALL)_{av}) \tag{34}$$

This relation can be found by substituting equation 24 into equation 26. Note that C4 represents the number of heat transfer units[2] and Z represents the effectiveness of the heat transfer in the channels.

The variable XX represents the change in enthalpy per unit mass of the air stream MA required to supply the total load, (Q+QWALL). In other words, the air entering the building has to experience an increase in enthalpy per unit mass equal to XX before leaving it (in order to absorb the heat load and assure steady comfort conditions). XX is calculated by:

$$XX = DT *(0.24+0.444*W5) \text{ (Btu/lbm-dry air)} \tag{35}$$

For the particular case where: the room temperature (TDESIG) is 75°F; the indoor relative humidity (RR) is 50%; and the room's inlet temperature drop (DT) is 20°F; then, equation 35 yields

XX = 4.88 Btu/lbm-dry air

The variable V in equation 32 can be obtained from:

$$V = U*AREA *PANELS*((TDI)_{av}-TREF)/(Q+QWALL)_{av} \tag{36}$$

The parameter V represents the ratio between the heat transferred from the air stream (if it were at the inlet temperature TDI) to the panels (if they were at the reference temperature TREF), divided by the average hourly total load of the building. The reference temperature, TREF, has been set equal to 65°F.

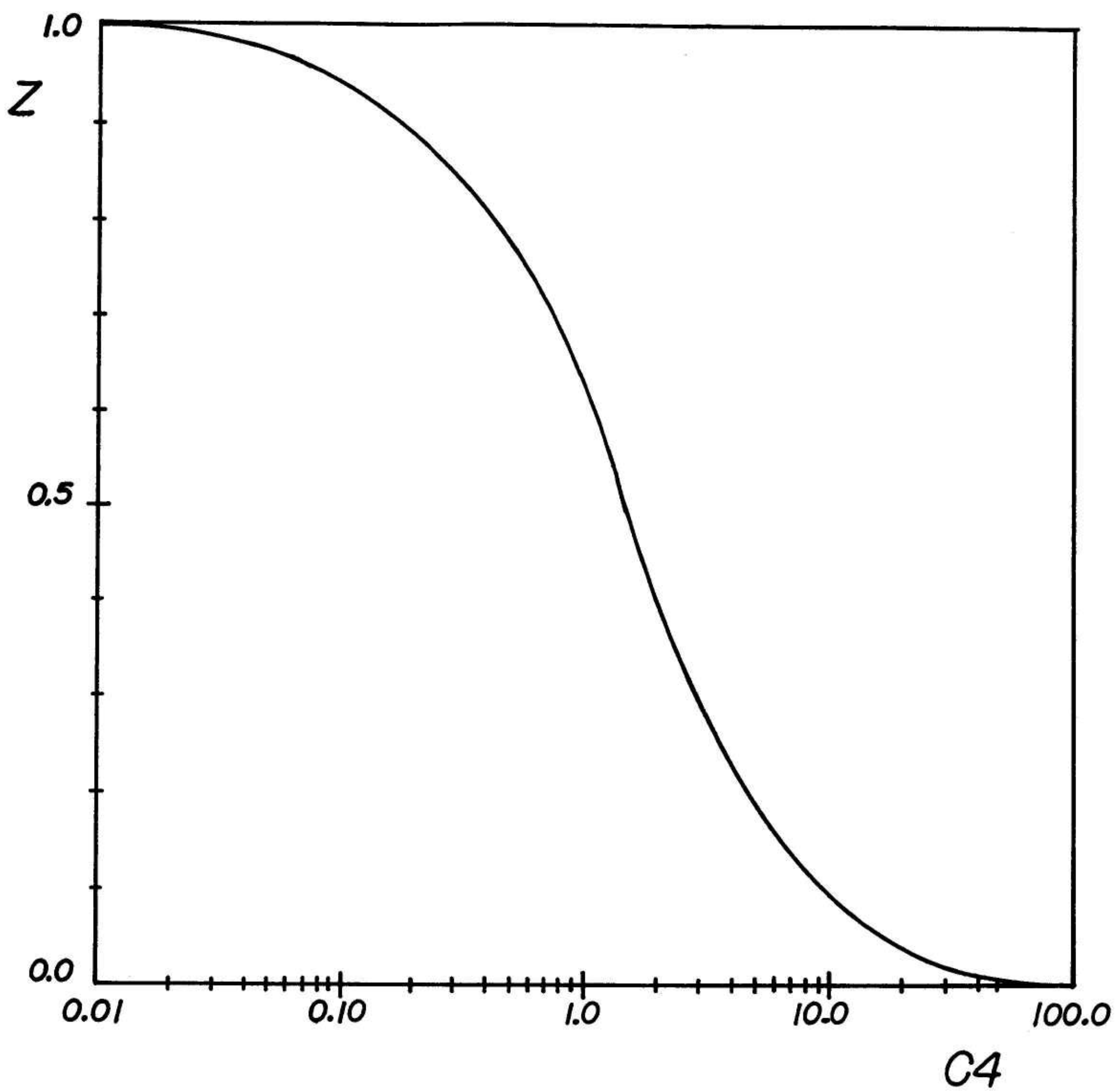

FIGURE 10

Factor Z as a Function of the Number of Heat Transfer Units

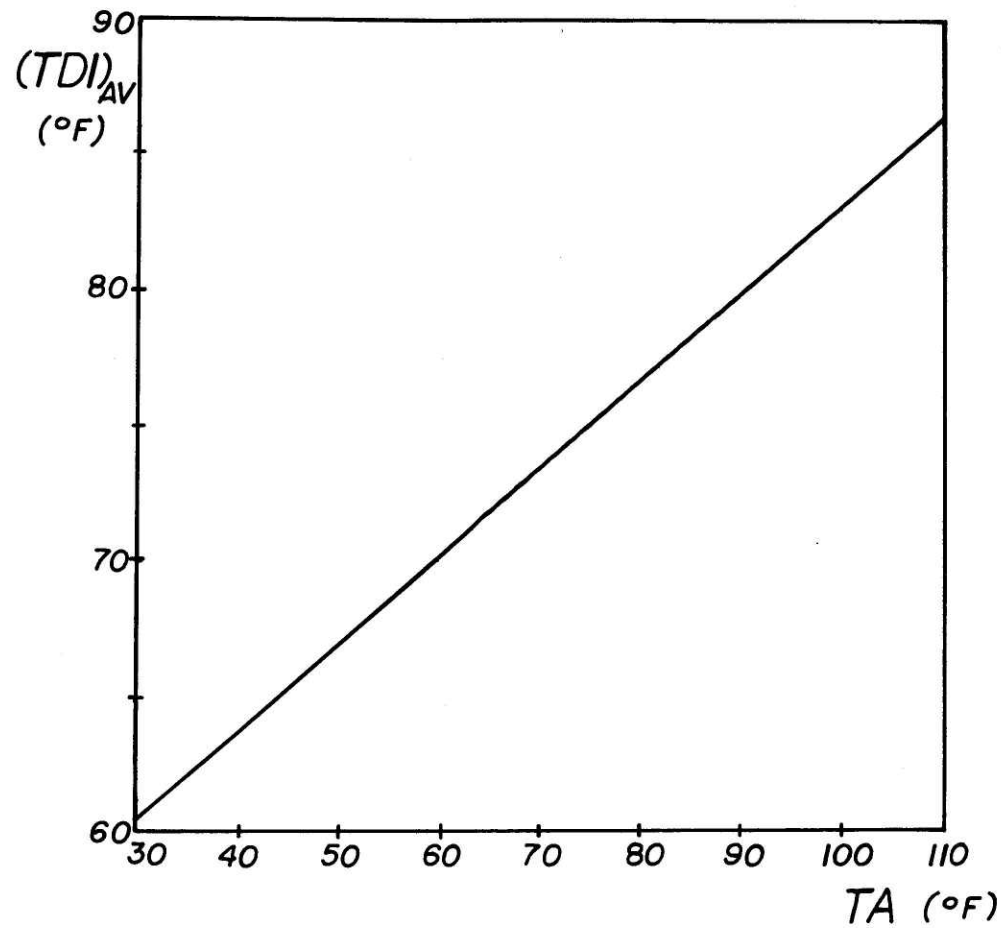

FIGURE 11

Average Channel's Inlet Temperature as a Function of Ambient Temperature

Typical values of U are in the range 1 - 2 $Btu/ft^2\ h°F$, as was mentioned in Chapter 2. The product of AREA times PANELS represents the total panels' area installed in the building. The variable W5 represents the absolute humidity of the air inside the shopping center; it is expressed in lbm-water/lbm-dry air. W5 can be calculated by the method outlined in Appendix B or by means of a psychrometric chart[9].

The value of the average hourly total load, i.e. $(Q+QWALL)_{av}$, is a parameter that has to be specified by the user of the correlation. In this study this value was calculated as it is explained in Chapter 2, using the computer program.

The channels' inlet temperature, TDI, is a function of the indoor design conditions; the outdoor relative humidity; and of the ambient temperature. Fixing the comfort conditions makes TDI an exclusive function of TA because the outdoor relative humidity, PHI, does not affect it significantly. Figure 11 presents a plot of $(TDI)_{av}$ versus TA.

Figures 12 and 13 show the plot of equation 32 for the locations considered during the summer season. PCOOL is presented in these graphs as a function of the non-dimensional parameter $(Z*V)^{-1}$. The physical meaning of the produce (Z*V) is that it represents the ratio between: the actual heat transfer that would occur if the air enters the channels at temperature TDI and the panels' temperature is TREF, and the hourly total load, $(Q+QWALL)_{av}$.

In conclusion, the steps to follow to calculate the cooling percentage are

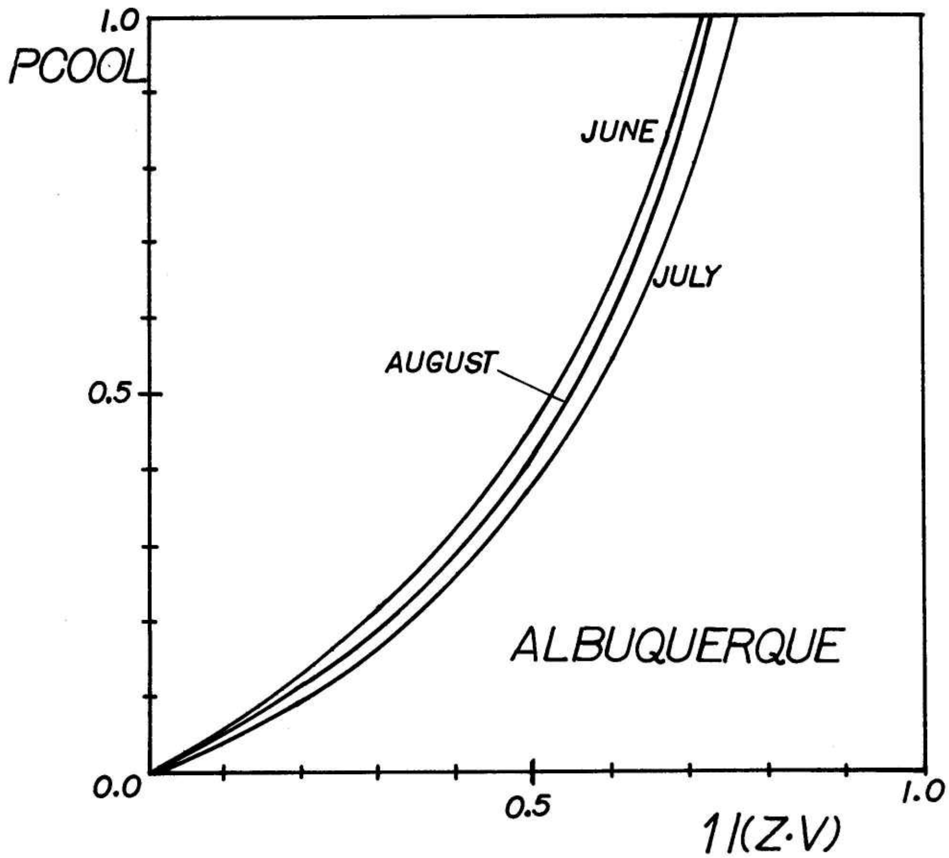

FIGURE 12

Monthly Percentage of Cooling for Albuquerque

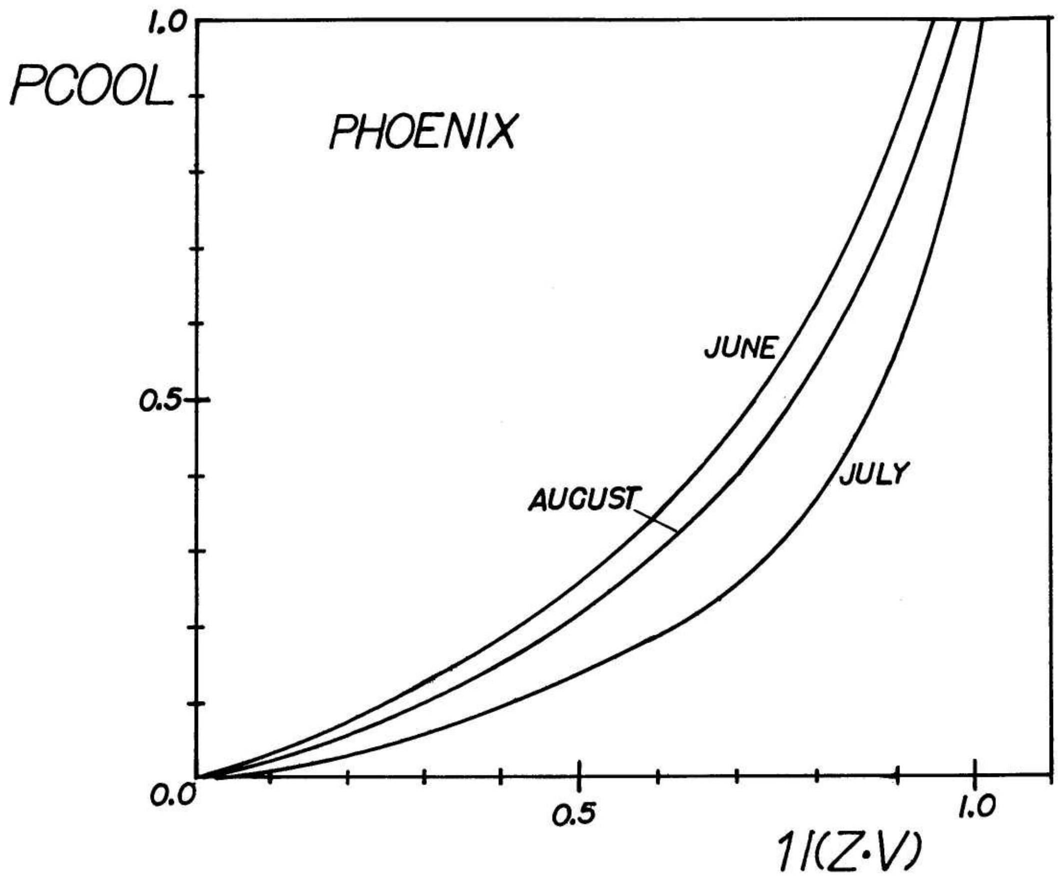

FIGURE 13

Monthly Percentage of Cooling for Phoenix

1) Determine U, $(TA)_{av}$ and $(Q+QWALL)_{av}$

2) Determine XX using equation 35

3) Find C4 with (34)

4) Determine Z from Figure 10

5) Read value of $(TDI)_{av}$ from Figure 11

6) Calculate V from equation 36

7) Compute $(Z*V)^{-1}$ and then determine PCOOL from Figures 12 or 13.

PCOOL can also be found from (32) after reading the values of A and B in Table III.

PCOOL found in this fashion represents the average percentage of cooling effect obtained for the month, i.e, the savings in total load overcome using the panels rather than relying solely on the A/C unit.

The general correlation presented yields very good agreement between results obtained from equation 32 and computer output. The maximum error detected for the cases considered is about 7% and in general the discrepancy is smaller than 5%. Needless to say, all assumptions made in Chapters 2 and 3 affect the results of the procedure outlined.

In conclusion, the advantages of the correlation are that it:

a) allows for a quick estimation of the performance

b) does not require a computer solution

c) uses very simple, easily determinable weather parameters

d) helps in the design phase because it shows the effect of changes in the parameters.

B. Output of the program

Table IV shows some of the results of the simulation for the case of building #2. In order to better understand the meaning of the figures shown, considerthe following:

- A/C load is the total amount of Btu's rejected by the air conditioning unit during the month

- TOTAL-QWALL is the radiation plus conduction heat load into the building (external load)

- TOTAL-LOAD combines all internal and external loads

- QV2 (day) is the heat rejected by the air in the channels to cool water in the storage layer

- QV2 (night) is similar to QV2(day) but for the purpose of cooling down the storage layer with cooler outdoor air. This is done occasionally, whenever ambient conditions allows; the stream is not delivered to the interior and is only done during non-working hours.

- Nocturnal radiation refers to the heat radiation at night from each panel

- PCOOL, percentage of cooling. Average for the month.

- Average value of temperature gradient (TDI-TT), in °F

- Maximum mass flow registered, in lbm/h, passing through the A/C unit.

Some other results, similar to those shown in Table IV are summarized in Appendix G.

One important remark concerning the simulation results is that in all the cases considered in this study, none of them exceeded 100 Kw-h/month of electrical power consumed to transport the air through the channels.

TABLE IV

TYPICAL RESULTS OF THE SIMULATION

LOCATION	MONTH	A/C LOAD	TOTAL QWALL	TOTAL LOAD	QV2 (DAY)	QV2 (NIGHT)
	JUNE	160.4	133.9	304.5	144.5	1.5
ALBUQUERQUE	JULY	191.2	145.8	322.1	133.3	1.5
	AUGUST	182.1	119.8	293.9	117.8	2.5
	JUNE	251.3	230.3	400.9	159.6	0
PHOENIX	JULY	337.2	224.0	400.3	95.0	0.2
	AUGUST	288.2	217.2	391.3	127.3	0

Comments:

1. The heat flows are expressed in millions of Btu.
2. The conductances CWALL and CROOF are equal to 0.30 Btu/h ft^2°F.
3. QGEN is equal to 3.50 Btu/ft^2h.

TABLE IV

(continuation)

LOCATION	MONTH	NOCTURNAL RADIATION	PCOOL	$(TDI-TT)_{av}$ (°F)	MAXIMUM FLOW
	JUNE	268.2	47.5	11.51	282.2
ALBUQUERQUE	JULY	242.8	41.4	9.94	291.7
	AUGUST	185.4	40.1	9.48	281.5
	JUNE	303.4	39.8	10.46	311.3
PHOENIX	JULY	204.0	23.7	6.00	327.6
	AUGUST	254.7	32.5	8.39	298.2

Comments:

1. The heat flows are expressed in thousands of Btu.
2. The flow is expressed in thousands of lbm/h.

Hence, as will be seen in the next chapter, it does not affect the economic performance of the system.

Some other important aspects of the results are:

a) Typical Reynold's numbers for the flow of air in the channels are in the range 15,000 - 30,000; thus, fluid velocities are about 2-5 ft/sec.

b) Nocturnal radiation heat flux is in the range 20-25 Btu/ft^2h.

c) In general, the internal heat load of the building accounts for 50-80% of the total heat load. Therefore, cooling is required most of the year.

d) The temperature gradient $(TDI-TT)_{av}$, varies from 5°F to 15°F with the most probable value near 10-11°F.

e) The effect of decreasing the design comfort temperature, TDESIG, or the design indoor relative humidity, RR, reduces the performance of the system. Same effect is obtained if the heat generation factor, QGEN, is increased.

f) There is only a slight influence of the wall conductances on the total load.

g) A lower temperature at the storage layer can be achieved if either the thickness of the storage layer, THI2, is increased, or the fluid resistance, PD, becomes smaller.

CHAPTER 5

ECONOMIC ANALYSIS

In this chapter some economical considerations will be discussed. The thermic diode solar panel can be economically feasible if the annual payments of the initial investment are lower (or at least equal) than the annual savings in electric energy. Note that no significant expenses either in channel air pumping power, not in maintenance of the panels affect this study.

Currently air-conditioning typically consumes from 20 to 40% of the total shopping center energy budget; these figures can be substantially larger for locations in the southern area of the country. Lighting and appliances consume the remaining 60-80% of the budget. In warmer climates cooling is needed year-round because of the large internal heat loads generated by lights, occupants and appliances. An energy breakdown is shown in Figure 14.

Energy density values of 7 to 10 watt/ft^2 are assigned in preliminary engineering studies of shopping centers; of this, about 5 watt/ft^2 are required by electrical loads other than air-conditioning. Shopping centers consume approximately 23 - 30 Kwh/ft^2 of electrical power annually.

All the aforementioned gives a rough idea of the power consumption using conventional cooling methods.

In order to compare annual cost versus savings obtained, building #2 of Table I has been selected to be analyzed. This building was selected because it is a mid-size shopping center. The same building with two different numbers of panels upon the roof will be separately considered.

To obtain annual savings, the computer program was run for a complete year of weather data in the two locations considered (Phoenix and Albuquerque).

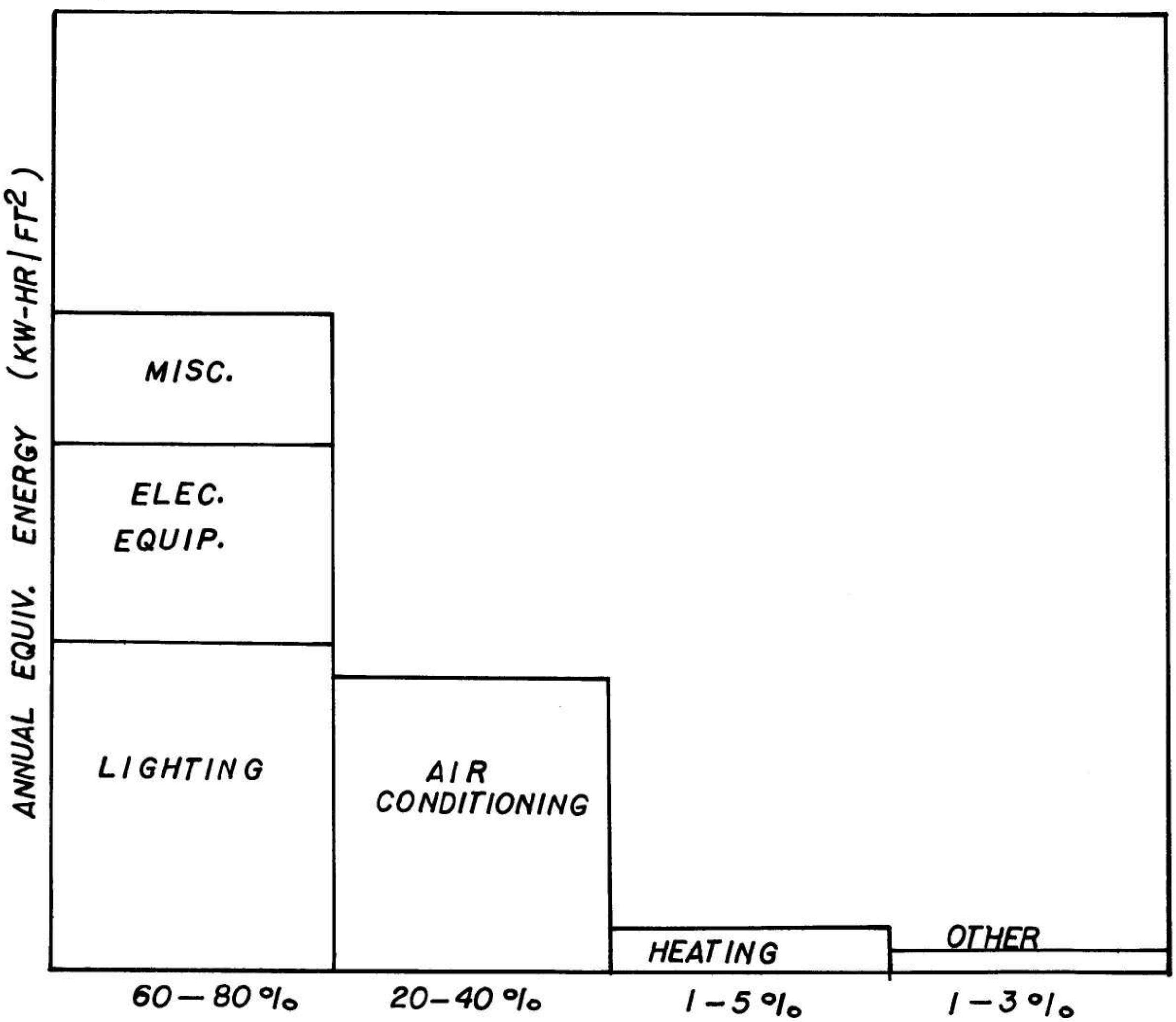

FIGURE 14

Energy Utilization Breakdown for Shopping Centers

The value of the heat generation factor, QGEN, used in the simulation for the present discussion, was chosen equal to 3.50 Btu/h ft^2. The walls' conductances CWALL and CROOF were selected equal to 0.30 Btu/h $ft^2 {}^\circ F$.

The summary of the simulation is presented on Tables V and VI. Note that the missing figures in those tables correspond to no-cooling required at all (that is, heating is required).

A. Case 1

In this case we have 2208 panels, covering almost half the roof. From the simulation we obtain that the total energy saved per year is

a) $1.03 * 10^9$ Btu, for Albuquerque

b) $1.88 * 10^9$ Btu, for Phoenix

Also, the total heat loads per year were

a) $1.69 * 10^9$ Btu, for Albuquerque

b) $3.10 * 10^9$ Btu, for Phoenix

In other words, on the average, per year, the efficiency of the panels are: 61.0% for Albuquerque and 60.7% for Phoenix.

Now if we assume a coefficient of performance (COP) of 2 for the air-conditioning units, and an electric rate of 5¢/Kwh, we have

a) savings in Albuquerque = $7544.68 per year

b) savings in Phoenix = $13,770.88 per year

On the other hand, the cost of the roof in a shopping center[12] is about $3.00 per ft^2. The actual cost of the thermic diode solar panel is $8.00 per ft^2; installation costs account for $2.00 more per ft^2. All this together means that the panel cost is $7.00 per ft^2 if we assume that it eliminates roof expenses.

TABLE V

ECONOMIC CASE No. 1

	ALBUQUERQUE			PHOENIX		
MONTH	$(TA)_{av}$ (°F)	ENERGY SAVED	PCOOL	$(TA)_{av}$ (°F)	ENERGY SAVED	PCOOL
JAN.	37.8	-	-	61.5	115.1	97
FEB.	44.2	-	-	56.8	89.1	100
MAR.	51.1	37.8	98	70.7	223.9	100
APR.	62.2	126.2	97	74.3	203.2	85
MAY	71.5	166.6	80	86.3	227.8	66
JUN.	82.8	146.9	48	95.8	161.0	40
JUL.	83.4	133.3	41	94.3	95.0	24
AUG.	82.2	119.7	40	93.3	129.4	33
SEP.	76.7	167.6	67	92.9	171.0	46
OCT.	62.3	116.5	92	77.4	189.0	73
NOV.	48.8	13.1	100	65.5	148.6	100
DEC.	40.9	-	-	59.9	106.5	100

Comments:

1. Energy in millions of Btu
2. Area of the roof covered by panels: 49%
3. No. of panels: 2208

Finally, let us assume that the interest rate per year, INT, is 7%; assume that the period of time, N, to recover the initial investment is 25 years. Therefore, the capital recovery, CR, defined as

$$CR = INT/(1-(1+INT)^{-N}) \qquad (37)$$

yields a value of 0.0858.

Using the calculated value of CR; knowing the total area of panel's surface; and applying the unit cost of panel, we obtain the annual payments:

$42,441.20 per year for the set of 2208 panels.

It is obvious that the alternative is not economically justifiable.

B. Case 2

Case B is concerned with the same building but with different number of panels installed. Now the number of panels is 1350.

The total energy saved per year is

a) $7.78 * 10^8$ Btu, for Albuquerque

b) 1.47×10^9 Btu, for Phoenix

also the total heat loads per year are

a) $1.79 * 10^9$ Btu, for Albuquerque

b) $3.43 * 10^9$ Btu, for Phoenix

In this case the equivalent or average efficiency of the system for the whole year was:

43.5% for Albuquerque, and 42.9% for Phoenix.

Assuming again a coefficient of performance equal to 2 for the air-conditioning units, and also an electric rate of 5¢/Kwh, we obtain

a) savings in Albuquerque = $5698.80 per year

b) savins in Phoenix = $10,767.65 per year

TABLE VI

ECONOMIC CASE No. 2

MONTH	ALBUQUERQUE			PHOENIX		
	$(TA)_{av}$ (°F)	ENERGY SAVED	PCOOL	$(TA)_{av}$ (°F)	ENERGY SAVED	PCOOL
JAN.	37.8	-	-	61.5	98.5	95
FEB.	44.2	-	-	56.8	70.8	100
MAR.	51.1	3.4	100	70.7	189.8	78
APR.	62.2	105.0	86	74.3	158.5	60
MAY	71.5	134.8	61	86.3	168.1	42
JUN.	82.8	110.6	32	95.8	112.8	24
JUL.	83.4	101.0	27	94.3	66.3	14
AUG.	80.2	91.5	27	93.3	93.6	20
SEP.	76.7	132.1	48	92.9	124.3	28
OCT.	62.3	99.1	86	77.4	148.1	52
NOV.	48.8	-	-	65.5	141.8	98
DEC.	40.9	-	-	59.9	89.1	100

Comments:

1. Energy in millions of Btu
2. Area of the roof covered by panels: 30%
3. No. of panels: 1350

Consider as in Case 1, that the unit cost of the panel is \$7.00 per ft^2 (eliminating roof expenses). Also consider an interest rate per year of 7% and a recovery period of 25 years. Then, CR is again equal to 0.0858 and the annual payments result:

\$25,949.10 per year for the set of 1350 panels

It is still to high to be comparative.

In conclusion, the economic feasibility of the thermic panels as used to cool a shopping center is seriously questioned. Since the panels cost so much relative to their energy savings they are not economically justifiable at present energy costs. However, this alternative will be bettter if the actual structural element replaced by the panels have a similar unit price. For instance, curtain wall modules used in vertical walls of many current buildings cost about \$16 per ft^2; actually more than the cost of the panels themselves. It means, that if panels are installed instead, not only will they be cheaper, but any cooling effect is free. Of course, the performance of the panel in a vertical orientation would be modified if it does not "see" the sky at night. Nevertheless, the dominating economic consideration is not so much the energy that the panels save, but rather on how much roof or wall they replace costs. Shopping centers with very cheap (\$3 per ft^2) roofs do not appear to be the proper application although other large buildings may be.

Appendix H gives other comments concerning the panels' economics.

CHAPTER 6

CONCLUSIONS

We can conclude that the thermic diode solar panel is an effective device to cope with cooling loads.

It has been seen that even high percentages of the total load demand can be handled during the summer season. Also, we saw in Chapter 5 for the case of two particular examples, that for the rest of the year the cooling percentage is even higher, in some cases 100%.

A correlation was found which allows the prediction of system performance, provided that we supply:

i) average ambient temperature prevailing during daytime in the lcoation, for the month considered

ii) average hourly total cooling load required

iii) average ambient relative humidity in the place under consideration

Through this correlation the need of a computer to predict performance is eliminated. Also, it allows for an easy way to determine the effect of the principal parameters on the output results; this is desirable from the design viewpoint.

Other conclusions which can be drawn from this study concern technical aspects. It was found that the air flow in the channels formed by the panels is always in the low turbulent regime range. Consequently, overall heat transfer coefficients for the air channels are of the order of 1 $Btu/ft^2h°F$.

Nocturnal radiation accounts for heat fluxes of the order of 25 $Btu/h\ ft^2$ of panel surface, for the near desert climate considered.

There is predominance of internal heat load contribution to the total load for shopping centers; this is more accentuated in cooler months. For the summer season, heat generation accounts for 50-80% of the total load. Finally, among the parameters which have the strongest influence over the panel's performance are:

a) the internal load coefficient: QGEN

b) the number of panels present: PANELS

c) the design comfort conditions: TDESIG, DT and RR

Concerning the economic analysis this study has shown that at the current costs of electric power, unit cost of roof construction, unit cost of panel, and under common financing procedures, the panels cannot compete against the more conventional methods employed. Note that this study assumes that the panels replace only the shopping center's relatively cheap roof. If they are to be installed in vertical surfaces, the economics are altered to the point where the panels are feasible. This alternative has not been contemplated because back-radiation from panels, depends on whatever conditions surrounds them; in other words, each case poses a differnt problem which requires a particular solution.

Nevertheless, even for the case of the panels placed in the roof, and for the hypothetical situation where:

i) electric rates triple

ii) Government's incentives are directed toward reducing financing interest rates, say to 4% annually

then the panels would be feasible or very close to the economic break-even point.

Lastly, this analysis has been carried out for two particular locations using given weather data. Hopefully, the data used is representative of that area and, in the long run, it is expected that the results shown are very close to the real ones. Extrapolation to other locations can be done provided that some similarity exists between parameters, like: ambient temperature, relative humidity, wind velocity, cloud cover and insolation fluxes.

APPENDIX A

PANELS' EQUATIONS

A.1. Sensor layer temperature during daytime.

During daytime there is no flow in the panel. Therefore, an energy balance over the sensor layer reads:

$$QC-QV1+QR1-QR2 = CWAT*MSS*\frac{d}{dt}(TS) \tag{38}$$

Now, using equations 1, 2, 3, and 4 into equation 38 yields:

$$\begin{aligned}&(CWAT*MSS/AREA)*\frac{d}{dt}(TS) + TS*(KW/THI3+H+C1) = \\ &(KW*TT/THI3+H*TA+RAD*ZETA*AB-C1*460)\end{aligned} \tag{39}$$

Let's use Euler's method to integrate equation 39. Then

$$(CWAT*MSS/AREA)*\frac{d}{dt}(TS) \simeq (CWAT*MSS/AREA)*\frac{(TS-\overline{TS})}{\Delta t}$$

where Δt in this case is 1 hour. Hence, the sensor layer temperature, TS, results

$$\begin{aligned}TS = \ &(KW*TT/THI3+H*TA+RAD*ZETA*AB-C1*460+\overline{TS}*CWAT*MSS/\\ &AREA)/(KW/THI3+H+C1+CWAT*MSS/AREA)\end{aligned}$$

A.2. Nighttime operation.

An energy balance for a differential element dX of the sensor layer (see Figure 6) reads

$$d(QC)-d(QV1)-d(QR2)-M*CWAT*d(TS) = 0 \tag{40}$$

after assuming steady state. Substituting equations 1, 2, and 4 into equation (40) gives:

$$KW*L2*(\overline{TT}-TS)/THI3-C1*L2*(TS+460)-H*L2*(TS-TA)= M*CWAT*\frac{d}{dX}(TS)$$

where again, $\overline{TT}$ is equal to the previous value of TT. The last expression can be rearranged to yield:

$$(M*CWAT/X2)*\frac{d}{dX}(TS)+TS = X1/X2 \qquad (41)$$

after using the definition of X1 and X2 (equations 10 and 11). The solution of equation 41 which satisfy the boundary conditions: TS (@ X=0) equal to T1, becomes

$$TS = X1/X2-(X1/X2-T1)*EXP(-X2*X/CWAT/M)$$

Consequently, we have TS as a function of the position X, some parameters X1 and X2, and the unknown mass flow M.

Momentum conservation requires that the total pressure drop of the flow in the panel matches the difference in pressure between the layers due to buoyancy forces. The difference in pressure is equal to the difference in total weight of water of each layer per unit cross-sectional area. The temperature dependence of the specific weight of water, ρ, can be approximated by

$$\rho = 62.944 - 0.010 * T \quad (lbf/ft^3) \qquad (42)$$

where the temperature T is in °F. This approximation yields a maximum error of 0.2% for the range 50 - 90°F. Then the momentum balance can be expressed by

$$\int_{o}^{L1} (\rho(TT) - \rho(TS))*SIN(ALFA)*dX = PD*M \qquad (43)$$

or, substituting equation 42 into (43):

$$SIN(ALFA)*\int_{o}^{L1} 0.01*(TT-TS)*dX = PD*M \qquad (44)$$

To solve for M, the model assumes that TT does not vary significantly from $\overline{TT}$ (see Chapter 2). Thus, introducing into equation 44: TT = $\overline{TT}$; TS given by (9); and carrying out the integration, yields

-100*PD*M/SIN(ALFA)-(X1/X2-T1)*(L1+(1-EXP(-X2*L1/

CWAT/M))*CWAT*M/L1/X2) = 0

The above equation can be simplified with the use of equations 13, 14, and 15 to obtain (12):

F(M)=X3-X4*M*(1-EXP(X5/M))-100*PD*M/SIN(ALFA) = 0 (12)

Note that the above expression, F(M)=0,can only be solved by trial-and-error. The program solves (12) for M using the subroutine FLOWM, which is the Newton-Raphson scheme.

Once M is obtained, the average value of TS for the sensor layer can be calculated:

$$(TS)_{av} = (1/L1) * \int_{o}^{L1} TS*dX =$$

X1/X2-(X1/X2-T1)*(1-EXP(-X2*L1/CWAT/M))*M*CWAT/(X2*L1)

Also, after determining M, T2 follows:

T2 = X1/X2-(X1/X2-T1)*EXP(-X2*L1/CWAT/M)

At night no significant heat transfer occurs at the back of the panel (QV2=0). An energy balance for the storage layer gives:

$$M*CWAT*(T2-T1)-QC=MST*CWAT*\frac{d}{dt}(TT) \tag{45}$$

but

$$(T2-T1) = (X1/X2-\overline{TT})*(1-EXP(-X2*L1/M/CWAT)) \tag{46}$$

hence, using equation 1 into equation 45 yields

$$TT = (CWAT*(T2-T1)*M+KW*AREA*TS/THI3+MST*CWAT*\overline{TT})/$$
$$(MST*CWAT+KW*AREA/THI3)$$

TS in this expression is given by (19). The integration of (45) has been carried out by Euler's method, where

$$\frac{d}{dt}(TT) \simeq \frac{TT-\overline{TT}}{\Delta t} \text{ and } \Delta t = 1 \text{ hour}$$

Now all the heat fluxes associated with the panel can be computed by means of equations 1, 2 and 4.

APPENDIX B

TOTAL SYSTEM CALCULATIONS

B.1. Air's humidity equation.

Air's absolute humidity is computed[9] by the function WW(X):

$$WW(X) = 0.622 * X/(14.7-X) \quad \text{(lbm-water/lbm-dry air)} \qquad (47)$$

where the dummy variable X represents the partial pressure of water vapor (expressed in psia). X is equal to the product of the relative humidity times the saturated water vapor pressure (this can be determined with equation 7).

The outdoor humidity W1 (see Figure 8), is calculated in this fashion, for which the relative humidity is PHI. For the design conditions stated in Chapter 3, the corresponding value of the indoor required humidity, W5, results:

$$W5 = 0.0093 \text{ lbm-water/lbm-dry air} \qquad (48)$$

The room's inlet humidity, W4, is essentially equal to W5 because in general the increase in humidity is negligible compared to the air's mass rate, MA.

B.2. Air's enthalpy equation.

The enthalpy is calculated[9] by the function EN(Y,W):

$$EN(Y,W) = 0.24*Y+(1061+0.444*Y)*W \quad \text{(Btu/lbm-dry air)} \qquad (49)$$

where the dummies variables Y and W represent temperature (°F) and absolute humidity (lbm-water/lbm-dry air) respectively.

The inlet enthalpy E1 is evaluated with the ambient temperature TA and the humidity W1 (see section B.1 above). E4 is evaluated using Y=TDESIG-DT (the room's inlet temperature) and the humidity W4 which is equal to W5. The room's enthalpy, E5, is evaluated with the temperature TDESIG and the humidity W5. For the comfort conditions considered in Chapter 3, E5 results to be

$$E5 = 28.04 \text{ Btu/lbm-dry air} \tag{50}$$

B.3. Internal heat load.

The internal heat load, Q, is formed by the people contribution QPEOP, and by the appliances contribution QAPP. Both contributions include sensible heat, like for instance: power dissipation in lights; human body heat generation, and so forth. Also, there is a latent heat load, i.e. water evaporated, supplied by appliances and people inside the building. It has been assumed that each person contributes with 450 Btu/h (sensible plus latent) to the total load[9]. The number of people in the shopping center can be related to the floor area; hence, QPEOP can be linked to the floor area by means of a proportionality constant. Similarly, the appliances load, QAPP, can also be considered to be a function of the floor area[13]. Therefore, the model calculates the hourly internal load, Q, by

$$Q = QGEN*AFLOOR \tag{51}$$

where AFLOOR is the total floor area and QGEN is a proportionality constant which includes the effect of both heat sources.

B.4. Exterior load calculation.

The exterior heat load of the building is formed by the conduction heat transfer plus the insolation contribution. Conduction heat transfer is due to the difference in temperature between outdoor and indoor. Significant insolation gain is present if the building's wall is hit by sunbeams.

Consider the analog electric circuit of Figure 15 for the heat transfer in the walls. The thermal resistances are 1/HO, for the outdoor convective term; 1/CWALL, for the wall itself; and 1/HIWL for the indoor convection heat transfer. HO is given by (22).

The input heat flux q''_o is the insolation flux and it is given by

$$q''_o = \text{ABSOR*DCOS(I)*RAD*A(I)} \qquad (52)$$

Solving the electric circuit for the heat flux going toward the room, q'', yields

$$q'' = (q''_o\text{+(TA-TDESIG)*HO*A(I))/(1+HO*(1/CWALL+1/HIWL))} \qquad (53)$$

which already combines radiation and conduction.

Then, QWALL results to be the summation of q'' for each wall and the roof.

B.5. Air mass flow.

MA represents the total mass flow of air flowing through the system. MA can be found from an energy balance applied on the building:

$$\text{MA * (E5-E4) = Q + QWALL} \qquad (54)$$

but

$$\text{E5-E4 = 0.24*DT+0.444*W5*DT = XX} \quad \text{(Btu/lbm)}$$

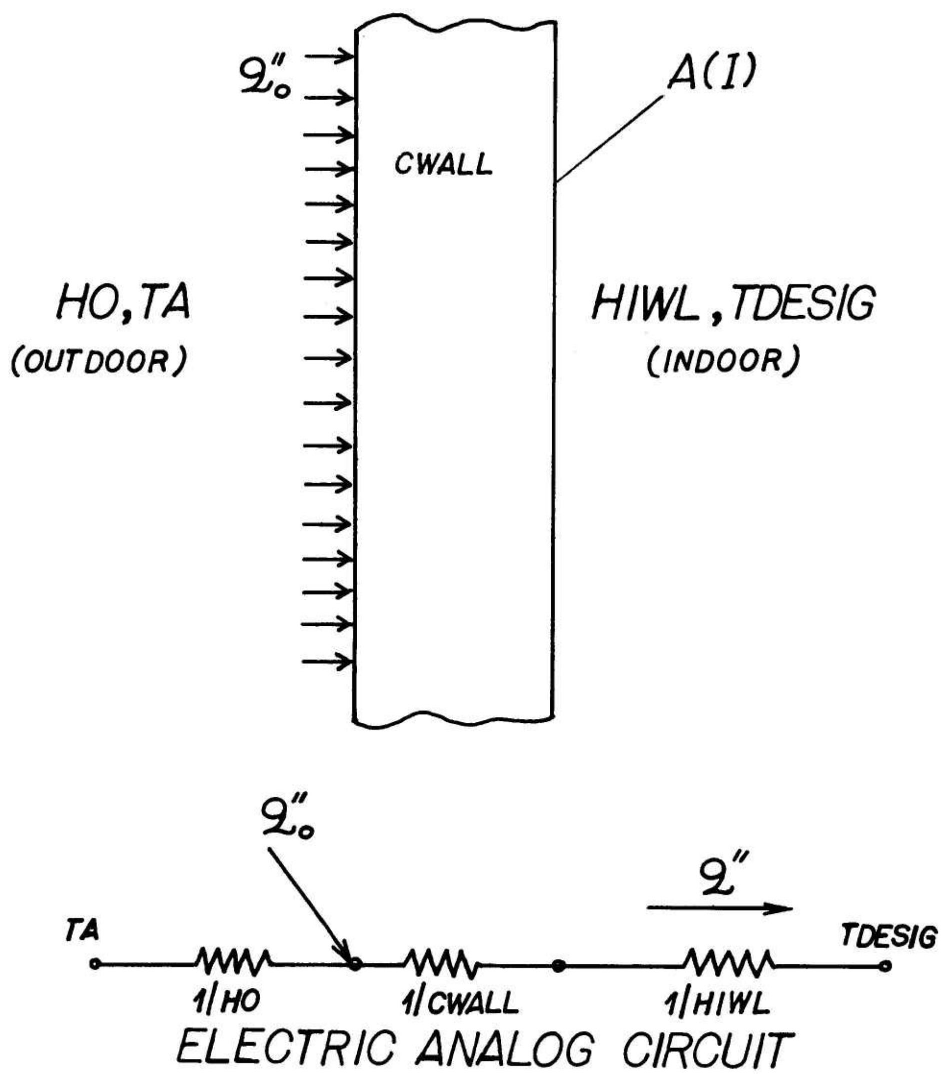

FIGURE 15

Heat Transfer at Building's Walls

after using (49) and W4=W5. Then,

$$MA = (Q+QWALL)/XX \qquad (lbm/h)$$

and MA per channel is

$$(MA)_{ch} = (Q+QWALL)/XX/(PANELS/PPERR) \tag{55}$$

B.6. Channel's inlet temperature.

The inlet temperature TDI can be expressed in terms of E2 and W2 by means of equation 49:

$$TDI = (E2-1061*W2)/(0.24+0.444*W2)$$

The enthalpy E2 and the humidity W2 can be expressed in terms of outdoor conditions (E1 and W1) and room's conditions (E5 and W5). The air stream entering the panels is the combination of fresh air plus recirculated load. The recirculation ratio is set by standards[8]; this program uses 2/3 for the calculations. Consequently

$$E2 = 2/3 * E5 + 1/3 * E1 \tag{56}$$

and

$$W2 = 2/3 * W5 + 1/3 * W1 \tag{57}$$

B.7. Channel's equations.

If U is the overall heat transfer coefficient, and if we assume a lumped parameter model for the behavior of the panels (storage layer), the heat exchanged follows:

$$\begin{aligned}(MA)_{ch} * CAIR * dT &= -d(QV2)_{ch} \\ &= -U*(T-\overline{TT})*2*L1*dX\end{aligned} \tag{58}$$

X in equation 58 represents the axial direction of the channel, and the origin is at the inlet; T represents the air temperature at the position X. Equatio 58 has been found from an energy balance for a differential element dX along the channel, and can also be written as

$$TAU1 * \frac{d}{dX}(T) + T = \overline{TT} \qquad (59)$$

where

$$TAU1 = (MA)_{ch}*CAIR/(2*U*L1) \qquad (60)$$

TAU1 is a length constant. $\overline{TT}$ is the previous hour value of TT.

The solution of (59) with the boundary condition T=TDI (@ X=0), is:

$$T = (TDI-\overline{TT})*EXP(-X/TAU1)+\overline{TT} \qquad (61)$$

Now, using (61) into (58) allows to calculate $(QV2)_{ch}$:

$$(QV2)_{ch} = \int_0^{LENGTH} 2*L1*U*(TDI-\overline{TT})*EXP(-X/TAU1)*dX$$

$$= 2*U*L1*TAU1*(TDI-\overline{TT})*(1-EXP(-LENGTH/TAU1))$$

$$= (MA)_{ch}*CAIR*(TDI-\overline{TT})*(1-EXP(-C4))$$

Note that to get the last expression we used equations 26, 55, and 60. Equation 25 is also valid to represent the total heat rejected to the panels, QV2, as a function of the total mass flow MA.

B.8. Storage layer temperature equation.

An energy balance for the storage layers for one channel reads:

$$PPERR*\dot{M}ST*CWAT*\frac{d}{dt}(TT) = (QV2)_{ch} \qquad (62)$$

Then, substituting (25) into (62) yields

$$TAU2 * \frac{d}{dt}(TT) + TT = TDI \tag{63}$$

where

$$TAU2 = (PPERR*MST*CWAT)/(CAIR*(MA)_{ch}*(1-EXP(-C4))) \tag{64}$$

The solution of equation 63 with the initial condition TT (@ t=0) equals $\overline{TT}$, is

$$TT = (\overline{TT}-TDI) * EXP(-t/TAU2) + TDI \tag{65}$$

B.9. <u>Air-conditioning load.</u>

The A/C unit provides the difference between the total load and QV2, plus the dehumidification load. Hence, it is calculated by

$$QAC = (Q+QWALL)+BETA*AFLOOR*(E1-E5+(W1-W5)*LAMBDA)-QV2$$

APPENDIX C

DIRECTION COSINES CALCULATION

This appendix is concerned with the determination of the variables DCOS(I) and ZETA. The following relationships presented are based on material drawn from references 3, 9 and 10. According to reference 3, the hour angle (solar noon being zero) can be expressed in radians by

$$\mathrm{HOAN} = 0.262 * (\mathrm{DATA}(5) - 12) \qquad (66)$$

where DATA(5) is the hour considered.

The model considers angles measured upon the horizontal plane (azimuth) to be positive clockwise with due south equal to zero degrees.

The solar azimuth is calculated accordingly to reference 10:

$$\cos(\gamma) = \mathrm{SEC(SOALT)*(COS(DECLI)*SIN(LAT)*COS(HOAN)-COS(LAT)*SIN(DECLI))} \qquad (67)$$

which is calculated by subroutine GG.

Subroutine G determines if the surface is in shadow or not. For the case of a surface which is not in shadow, the same subroutine G, after making use of subroutine GG, returns the value of the wall solar azimuth.

Finally, the direction cosine for the incident beams normal to the panel's surface is given by

$$\mathrm{ZETA} = \mathrm{(COS(SOALT)*COS(G)*SIN(ALFA)+SIN(SOALT)*COS(ALFA))/(SIN(DECLI)*SIN(LAT)+COS(DECLI)*COS(LAT)*COS(HOAN))} \qquad (68)$$

For the case of the building's walls, equation 68 reduces to

```
DCOS(I) = COS(SOALT)*COS(G)/(SIN(DECLI)*SIN(LAT)+
          COS(DECLI)*COS(LAT)*COS(HOAN))
```
(69)

Note that these conversion factors have been determined to find normal incidence on walls (or roof) when the insolation upon a horizontal plane is known.

APPENDIX D

PRESSURE DROP IN THE DUCTS

The pressure drop along the triangular ducts which convey the air has been calculated using

$$\Delta P = f * \frac{L}{D_h} * \frac{V^2}{2} * \rho_{air} \qquad (70)$$

where the friction factor has been approximated by

$$f = 0.316/Re^{0.25} \qquad (71)$$

and Re (Reynold's number) = $\frac{V*\rho*D_h}{\mu}$ (72)

The length L in equation 70 is the sum of the lengths of all the channels. In other words, L is equal to

L = PANELS*LENGTH/PPERR (73)

The density of the air has been assumed be be

$$\rho_{air} = 0.0763 \text{ lbm/ft}^3$$

and the viscosity

$$\mu_{air} = 0.043 \text{ lbm/h ft}$$

The hydraulic diameter D_h is the ratio of the cross sectional area and the "wetted" perimeter. The cross sectional area is:

CS = L1^2*SIN(2*ALFA)/2 (ft^2) (74)

and the "wetted" perimeter is obtained from:

PER = L1*SIN(2*ALFA)/4/(1+COS(ALFA)) (ft) (75)

Therefore, D_h is the ratio of CS divided by PER.

The velocity V is found using the continuity equation which yields

$$V = MA*PPERR(I)/(CS*PANELS*\rho_{air}) \qquad (76)$$

Combining all the equations presented in this appendix produces the expression for the total pressure drop along each channel:

$$PDROP = 4.4*10^{-8}*(1+COS(ALFA))^{1.25}*LENGTH*(MA_{ch})^{1.75}/ (SIN(2*ALFA))^{3}/L1^{4.75} \quad (lbf/ft^{2}) \qquad (77)$$

The total power consumed (expressed in Kw-h) to pump the air is equal to the product of the volume flow rate times PDROP. Hence, it becomes

$$POWER = PDROP*MA/\rho_{air}/2.656*10^{6} \quad (kw\text{-}h) \qquad (78)$$

APPENDIX E

COMPUTER PROGRAM

```
C
C
C
C
C
C
C
C
C         THIS COMPUTER PROGRAM HAS BEEN DEVELOPED TO SIMULATE THE THERMAL
C         DYNAMICS OF THE THERMAL DIODE SOLAR PANEL WORKING IN THE COOLING
C         MODE, IN ORDER TO OVERCOME HEAT LOADS IN BUILDINGS. THE PROGRAM
C         REQUIRES INFORMATION CONCERNING PHYSICAL DIMENSIONS AND PARAMETERS
C         OF THE PANELS.IT ALSO REQUIRES DATA OF THE BUILDING AND DESIGN
C         INDOOR CONDITIONS. THE PERIOD OF TIME OF THE SIMULATION; INITIAL
C         MONTH;AND LOCATION (STATION) CONSIDERED,ARE INPUT DATA NEEDED TOO.
C         THE PROGRAM PRINTS ALL THE INFORMATION SUPPLIED AND THEN STARTS
C         THE CALCULATIONS. THESE CALCULATIONS ARE MAINLY AIMED TO GET
C         TEMPERATURES AND HEAT FLUXES AT DIFFERENT PLACES FOR EACH
C         HOUR SIMULATED. WEATHER DATA IS NEEDED TO ACCOMPLISH THE TASK.
C         THE DATA REQUIRED SHOULD BE HOURLY WEATHER DATA FOR THE
C         LOCATION UNDER CONSIDERATION. THE PROGRAM REQUIRES SPECIFICALLY:
C         DATA(1)=STATION;DATA(3)=MONTH;DATA(4)=DAY;DATA(5)=HOUR;DATA(6)=
C         INSOLATION ON A HORIZ. SURF. (LANGLEY/H);DATA(7)=SOLAR ALT. (DEG.);
C         DATA(10)=WIND VEL. (KNOT);DATA(11)=REL. HUM. (%);DATA(12)=CLOUD COV.
C         (TENTH OF SKY);DATA(14)=AMBIENT TEMP. (F).
C         FINALLY,AFTER EACH MONTH OF COMPUTATION,THE PROGRAM PRINTS OUT A
C         SUMMARY OF THE IMPORTANT HEAT FLUXES AND AVERAGE CONDITIONS OR
C         PARAMETERS,REPRESENTATIVES OF THE PERFORMANCE OF THE SYSTEM.
C
C
C
C
C
C
C
C
```

```
C
C
C
      SPECIFICATIONS
      --------------
C
C
      DIMENSION DATA(14),AZI(5),TIL(5),A(5),DCOS(5),SUM(8)
      REAL KW,LAT,LAMBDA,LENGTH,L1,L2,M,MA,MSS,MST,MONTH,MAX
      INTEGER TIME
C
C     ------------------------------------------------------------------
C
C     FUNCTIONS
C     ---------
C
      WW(X)=0.622*X/(14.7-X)
      EN(X,Y)=0.24*X+(1061.+.444*X)*Y
      PWS(T)=0.632028798-0.020747209*T+0.000240210*T**2
C
C     ------------------------------------------------------------------
C
C      INPUT  DATA
C      -----------
C
C     PANEL  PARAMETERS
C     -----------------
C
      READ(5,20)KW,L1,L2,THI1,THI2,THI3
      READ(5,30)AB,ALFA,AZIMP,EM,PD,TMIN
C
C     NUMBER OF PANELS AND CHANNEL SPECIFICATIONS
C     -------------------------------------------
C
      READ(5,20)PANELS,LENGTH,PPERR
C
C
```

```
C          BUILDING DATA
C          -------------
      READ(5,20)HIRF,HIWL,HRO,HOURS,LAT
      READ(5,20)QGEN,CROOF,CWALL,BETA
      READ(5,20)AFLOOR
      DO 10 I=1,5
      READ(5,20)A(I),AZI(I),TIL(I)
      AZI(I)=AZI(I)*0.0175
      TIL(I)=TIL(I)*0.0175
10    CONTINUE
C
C          DESIGN SPECIFICATIONS
C          ---------------------
C
      READ(5,20)DT,RR,TDESIG
C
C          PHYSICAL CONSTANTS
C          ------------------
C
      READ(5,40)ABSOR,CAIR,CWAT,LAMBDA,SIG
C
C          OTHER SIMULATION DATA
C          ---------------------
      READ(5,50)TIME
      READ(5,20)STAT,MONTH
      DATA TREF/65.0/
C
C          INPUT FORMATS
C          -------------
C
20    FORMAT(8F10.2)
30    FORMAT(4F10.2,F10.4,F10.2)
40    FORMAT(4F10.2,E20.4)
50    FORMAT(I6)
60    FORMAT(F6.0,F4.0,3F3.0,F7.1,2F4.0,4F3.0,F9.0,F5.0)
C
```

```
C      ------------------------------------------------------------------
C
C         INPUT DATA PRINTOUT
C         -------------------
       WRITE(6,6000)
       WRITE(6,6010)AB,ALFA,AZIMP,EM,KW,L1,L2,PD,THI1,THI2,THI3,TMIN
       WRITE(6,6020)
       WRITE(6,6030)ABSOR,CAIR,CWAT,DT,HIRF,HIWL,HRO,HOURS
       WRITE(6,6040)LAMBDA,LAT,RR,SIG,TDESIG
C
C      ------------------------------------------------------------------
C
C         FORMATS ASSOCIATED TO INPUT DATA
C         --------------------------------
C
 6000  FORMAT(1H1,10X,'PANEL|S PARAMETERS IN THE SIMULATION',/,11X,'-----
      *-------------------------------')
 6010  FORMAT(1H0,/,10X,'AB    =',F8.2,//,10X,
      1'ALFA =',F8.2,' DEGREES',//,10X,
      2'AZIMP=',F8.2,' DEGREES',//,10X,
      3'EM   =',F8.2,//,10X,
      4'KW   =',F8.2,' BTU/FT-HR-F',//,10X,
      5'L1   =',F8.2,' FT.',//,10X,
      6'L2   =',F8.2,' FT.',//,10X,
      7'PD   =',F8.4,' LBF/SQ.FT./(LBM/HR)',//,10X,
      8'THI1 =',F8.2,' IN.',//,10X,
      9'THI2 =',F8.2,' IN.',//,10X,
      *'THI3 =',F8.2,' IN.',//,10X,
      *'TMIN =',F8.2,' DEGREES')
 6020  FORMAT(1H1,10X,'REFERENCE BUILDING DATA AND PARAMETERS',/,11X,'---
      *-------------------------------------')
 6030  FORMAT(1H0,/,10X,'ABSOR =',F11.2,//,10X,
      1'CAIR  =',F11.2,' BTU/F-LBM',//,10X,
      2'CWAT  =',F11.2,' BTU/F-LBM',//,10X,
      5'DT    =',F11.2,' DEGREES',//,10X,
      7'HIRF  =',F11.2,' BTU/SQ.FT.-HR-F',//,10X,
```

```
     8'HIWL  =',F11.2,' BTU/SQ.FT.-HR-F',//,10X,
     9'HRO   =',F11.2,' BTU/SQ.FT.-HR-F',//,10X,
     *'HOURS =',F11.2,' WORKING HOURS')
6040  FORMAT(1H0, 9X,'LAMEDA=',F11.2,' BTU/LBM',//,10X,
     1'LAT   =',F11.2,' DEGREES(NORTH)',//,10X,
     8'RR    =',F11.2,//,10X,
     9'SIG   =',E11.4,' BTU/SQ.FT.-F**4-HR',//,10X,
     $'TDESIG=',F11.2,' F')
C
C      ------------------------------------------------------------------
C
C         INPUT DATA SEARCH (USING TAPES)
C         -------------------------------
C
      DO 66 NS=1,70000
      READ(12,60,END=500)DATA
      IF(DATA(1).EQ.STAT)GO TO 67
66    CONTINUE
67    CONTINUE
C
C         -------------------------------
C
C         CONSTANTS' CALCULATION
C         ----------------------
C
      ALFA=ALFA*0.0175
      AREA=L1*L2
      AZIMP=AZIMP*0.0175
      HC =(1.+COS(ALFA))**0.2*0.00839/SIN(2.*ALFA)/L1**1.8
      LAT=LAT*0.0175
      PW=PWS(TDESIG)*RR
      W5=WW(PW)
      E5=EN(TDESIG,W5)
      XX=DT*(0.24+0.444*W5)
      PDR=2.288E-09*(1.+COS(ALFA))**1.25*19.03/L1**4.75/SIN(2.*ALFA)**3
      THI1=THI1/12.
```

```
      THI2=THI2/12.
      THI3=THI3/12.
      MSS=AREA*THI1*62.4
      MST=AREA*THI2*62.4
      U=1.0
      C3=U*AREA*XX/CAIR
      C6=U*AREA/CWAT/MST
C
C     ------------------------------------------------------------------
C
C
C        INITIAL CONDITIONS
C        ------------------
C
      DATA KEY,M,TS,TT/0,100.,67.,60./
70    DO 80 I=1,8
      SUM(I)=0.
80    CONTINUE
      DATA MAX,NAVG,SQV2,SQR2,STDITT,XA/5000.,0,4*0./
C
C     ------------------------------------------------------------------
C     ------------------------------------------------------------------
C
C         PRINCIPAL    LOOP
C         -----------------
C
      DO 390 JA=1,TIME
      READ(12,60,END=500 )DATA
      IF(DATA(3).NE.MONTH)GO TO 400
      MONTH=DATA(3)
      DECLI=0.41*SIN(0.0172*(DATA(4)+(DATA(3)-1.)*30.+284.))
      TA=DATA(14)
      WV=DATA(10)
      CC=DATA(12)
      PHI=DATA(11)/100.
      H=1.09+0.39*WV
```

```
      HO=H+HRO                                                          MAIN0217
      IF(DATA(6).LT.1.)GO TO 100                                        MAIN0218
      IF(DATA(6).LT.130.)GO TO 110                                      MAIN0219
100   DATA(6)=0.                                                        MAIN0220
      RAD=0.                                                            MAIN0221
      ZETA=0.                                                           MAIN0222
      GO TO 140                                                         MAIN0223
110   SOALT=DATA(7)*0.0175                                              MAIN0224
      IF(SOALT.GT.0.35)GO TO 120                                        MAIN0225
      RAD=3.69*DATA(6)                                                  MAIN0226
      GO TO 130                                                         MAIN0227
120   RAD=3.69*DATA(6)/SIN(SOALT)                                       MAIN0228
130   HOAN=0.262*(DATA(5)-12.)                                          MAIN0229
      F8=GG(DECLI,HOAN,LAT,SOALT)                                       MAIN0230
      F=G(AZIMP,HOAN,F8)*0.017453                                       MAIN0231
      ZETA=COS(SOALT)*COS(F)*SIN(ALFA)+SIN(SOALT)*COS(ALFA)             MAIN0232
      ZETA=ZETA/(SIN(DECLI*SIN(LAT)+COS(DECLI)*COS(LAT)*COS(HOAN))      MAIN0233
C                                                                       MAIN0234
C     ------------------------------------                              MAIN0235
C                                                                       MAIN0236
C        PANELS' CALCULATIONS                                           MAIN0237
C        ---------------------                                          MAIN0238
      TS=X1/X2+(X1/X2-T1)/T*(EX(T)-1.)                                  MAIN0239
      T1=TT                                                             MAIN0240
      PW=PWS(TA)*PHI                                                    MAIN0241
      C1=(1.-.09*CC)*(0.5-0.5979*SQRT(PW))*EM*SIG*(TS+460.)**3          MAIN0242
      IF(M.GT.0.)GO TO 150                                              MAIN0243
      KEY=1                                                             MAIN0244
      GO TO 160                                                         MAIN0245
150   X1=(RAD*AB*ZETA+KW*T1/THI3+H*TA-C1*460.)*L2                       MAIN0246
      X2=(KW/THI3+H+C1)*L2                                              MAIN0247
      F=X1/X2                                                           MAIN0248
      IF(F.GT.T1)GO TO 160                                              MAIN0249
      X3=(T1-X1/X2)*L1                                                  MAIN0250
      X4=(X1/X2-T1)/X2*CWAT                                             MAIN0251
      X5=-L1*X2/CWAT                                                    MAIN0252
```

```
      M=100.
      S=-0.20*X5
      M=FLOWM(M,X1,X2,X3,X4,X5,S,PD,ALFA)
      IF(M.LE.0.)GO TO 160
      T=-X5/M
      T2=X1/X2-(X1/X2-T1)*EX(T)
      GO TO 180
160   M=0.
      TS=(RAD*ZETA*AB+H*TA-C1*460.+KW*TT/THI3+MSS*CWAT*TS/AREA)/(H+C1+KW
     $/THI3+MSS*CWAT/AREA)
      IF (KEY.NE.1)GO TO 170
      KEY=0
      T=TT-TS
      IF(T.GT.TMIN)GO TO 150
170   T1=TS
      T2=TS
C
180   IF(M.EQ.0.)GO TO 190
      T=X2*L1/M/CWAT
      TS    =X1/X2+(X1/X2-T1)/T*(EX(T)-1.)
C        TEST TO DETERMINE WORKING HOURS
190   IF(DATA(5).GE.22.)GO TO 350
      IF(DATA(5).LT.9.)GO TO 350
C
C        AIR DUCT CALCULATIONS
C        ---------------------
C
      TAVG=TAVG+TA
      NAVG=NAVG+1
      N=IFIX(DATA(5)-8.)
      PW=PWS(TA)*PHI
      W1=WW(PW)
      E1=EN(TA,W1)
      E2=0.33*E1+0.67*E5
      W2=0.33*W1+0.67*W5
      TDI=(E2-1061.*W2)/(0.24+0.444*W2)
```

```
      QR1=RAD*ZETA*AREA*AB
200   QV1=H*AREA*(TS-TA)
      QR2=C1*AREA*(TS+460.)
      IF(RAD.EQ.0.)GO TO 230
      DO 220 II=1,5
      IF(II.NE.1)GO TO 210
      DCOS(II)=SIN(SOALT)
      GO TO 220
210   DCOS(II)=COS(G(AZI(II),HCAN,F8)*0.017453)*COS(SOALT)
      DCOS(II)=DCOS(II)/(SIN(DECLI)*SIN(LAT)+COS(DECLI)*COS(LAT)*COS(HOA
     *N))
220   CONTINUE
230   Q=QGEN*AFLOOR
      QWALL=0.
      F1=1.+HC*(1./HIRF+1./CROOF)
      F=1.+HC*(1./HIWL+1./CWALL)
      DO 290 II=1,5
      IF(RAD.EQ.0.)GO TO 260
      IF(II.NE.1)GO TO 240
      F2=(ABSOR*DCOS(1)*RAD+(TA-TDESIG)*HO)/F1*A(1)
      GO TO 250
240   F2=(ABSOR*DCOS(II)*RAD+(TA-TDESIG)*HO)/F*A(II)
250   QWALL=QWALL+F2
      GO TO 290
260   IF(TA.EQ.TDESIG)GO TO 300
      IF(II.NE.1)GO TO 270
      F2=(TA-TDESIG)*HO/F1*A(1)
      GO TO 280
270   F2=(TA-TDESIG)*HO/F*A(II)
280   QWALL=QWALL+F2
290   CONTINUE
300   MA=(QWALL+Q)/XX/(PANELS/PPERR)
      IF(MA.LT.5000.)GO TO 320
310   C4=C3*PANELS/(Q+QWALL)
      C5=C4*CAIR*(TDI-TI)/XX
      C7=(1.-EX(C4))/C4
```

```
      QV2=C5*C7*(Q+QWALL)                                               MAIN0325
      TT=TDI+(TT-TDI)*EX(C6*C7)                                         MAIN0326
      QAC=Q+QWALL-QV2                                                   MAIN0327
      IF(W1.LE.W5)GO TO 315                                             MAIN0328
      QAC=QAC+BETA*AFLOOR*(W1-W5)*LAMBDA                                MAIN0329
315   IF(E1.LE.E5)GO TO 316                                             MAIN0330
      QAC=QAC+BETA*AFLOOR*(E1-E5)                                       MAIN0331
316   IF(QAC.GT.0.)GO TO 330                                            MAIN0332
      QAC=0.                                                            MAIN0333
      GO TO 330                                                         MAIN0334
320   QAC=Q+QWALL                                                       MAIN0335
      IF(W1.LE.W5)GO TO 325                                             MAIN0336
      QAC=QAC+BETA*AFLOOR*(W1-W5)*LAMBDA                                MAIN0337
325   IF(E1.LE.E5)GO TO 326                                             MAIN0338
      QAC=QAC+BETA*AFLOOR*(E1-E5)                                       MAIN0339
326   QV2=0.                                                            MAIN0340
      MA=0.                                                             MAIN0341
      GO TO 316                                                         MAIN0342
C                                                                       MAIN0343
C    HEAT CALCULATIONS AND SUMMARY                                      MAIN0344
C    ------------------------------                                     MAIN0345
C                                                                       MAIN0346
330   PDROP=PDF*MA**1.75*LENGTH                                         MAIN0347
      POWER=PDROP*MA*PANELS/PPERR/202650.                               MAIN0348
      IF(MAX.GT.MA)GO TO 340                                            MAIN0349
      MAX=MA                                                            MAIN0350
340   SUM(1)=SUM(1)+QAC                                                 MAIN0351
      SUM(2)=SUM(2)+QV2                                                 MAIN0352
      SUM(3)=SUM(3)+Q+QWALL                                             MAIN0353
      SUM(4)=SUM(4)+QWALL                                               MAIN0354
      SUM(5)=SUM(5)+POWER                                               MAIN0355
      SUM(6)=SUM(6)+QV1                                                 MAIN0356
      SUM(7)=SUM(7)+QR1                                                 MAIN0357
      SUM(8)=SUM(8)+QR2                                                 MAIN0358
      STDITT=STDITT+TDI-TT                                              MAIN0359
      XA=XA+(TDI-TT)/(TDI-TREF)                                         MAIN0360
```

```
      GO TO 390
C
C     ----------------------------------------------------------------
C
C        CALCULATIONS DURING NIGHTTIME. NO AIR FLOW IN CHANNELS.
C
350   F1=T1-5.
      IF(TA.LT.F1)GO TO 360
      TT=(CWAT*(T2-T1)*M+KW/THI3*AREA*TS+MST*CWAT*TT)/(MST*CWAT+KW/THI3*
     *AREA)
      QR2=C1*AREA*(TS+460.)
      SQR2=SQR2+QR2
      GO TO 390
C
C     ----------------------------------------------------------------
C
C        CALCULATIONS DURING NIGHTTIME. THERE IS AIR FLOW IN THE CHANNELS.
C
360   MA=10000.
      TDI=TA
370   TAU1=MA*CAIR/2./L1/U
      T=LENGTH/TAU1
      F1=2.*L1*TAU1*U*(1.-EX(T))
      TAU2=PPERR*MST*CWAT/F1
      QV2=F1*(TDI-TT)*PANELS/PPERR
      F=M*CWAT*PPERR*(T2-T1)/F1
      T=1./TAU2
      TT=(TT-TDI-F)*EX(T)+F+TDI
      QR2=C1*AREA*(TS+460.)
      SQR2=SQR2+QR2
      SQV2=SQV2-QV2
C
C     ----------------------------------------------------------------
C
C        END OF THE PRINCIPAL LOOP
C
```

```
390   CONTINUE
C
C     ------------------------------------------------------------------
C
C        CALCULATIONS ASSOCIATED TO THE DATA IN THE MONTHLY SUMMARY
C
400   TAVG=TAVG/FLOAT(NAVG)
      STDIIT=STDIIT/FLOAT(NAVG)
      XA=XA/FLOAT(NAVG)
      SQV2=SQV2/1000.
      SQR2=SQR2/1000.
      SUM(1)=SUM(1)/1000000.
      SUM(2)=SUM(2)/1000000.
      SUM(3)=SUM(3)/1000000.
      SUM(4)=SUM(4)/1000000.
      SUM(6)=SUM(6)/1000.
      SUM(7)=SUM(7)/1000.
      SUM(8)=SUM(8)/1000.
      F=U*AREA*PANELS/(SUM(3)*1000000./FLOAT(NAVG))
      Y=F*STDIIT
      Z=(1.-EX(F*XX/CAIR))/(F*XX/CAIR)
      ABUIL=(A(1)+A(2)+A(3)+A(4)+A(5))/1000.
      PCOOL=SUM(2)/SUM(3)*100.
      MAX=MAX*PANELS/PPERR/1000.
C
C        RESULTS' PRINTOUT
C        -----------------
C
      WRITE(6,4010)STAT
420   WRITE(6,4030)MONTH
      WRITE(6,4040)
      WRITE(6,4050)SUM(1)
      WRITE(6,4060)SUM(2)
      WRITE(6,4070)SUM(4)
      WRITE(6,4080)SUM(3)
      WRITE(6,4090)
```

```
      WRITE(6,4100)SUM(6)
      WRITE(6,4110)SUM(7)
      WRITE(6,4120)SQR2
      WRITE(6,4130)SUM(8)
      WRITE(6,4140)SQV2
      WRITE(6,4150)SUM(5)
      WRITE(6,4160)PCOOL
      WRITE(6,4170)Y
      WRITE(6,4180)Z,XA
      WRITE(6,4190)ABUIL,TAVG,STDITT
      WRITE(6,4200)MAX
      WRITE(6,4210)(A(J),AZI(J),TIL(J),J=1,5),AFLOOR
      WRITE(6,4220)PANELS,PPERR,LENGTH,QGEN,CROOF,CWALL
430   CONTINUE
C
C     ---------------------------------------------------------------
C
C        FORMATS ASSOCIATED TO THE RESULTS' PRINTOUT
C        ----------------------------------------
C
4010  FORMAT(1H1,15X,'SUMMARY CORRESPONDING TO',F8.0,/,16X,'-----',     '
     *-------------------------------',/)
4030  FORMAT(1H0,23X,'MONTH=',F3.0)
4040  FORMAT(4X,'  ---------------------------------------------------',/,
     *13X,'(TOTAL HEAT FLOWS IN MILLIONS OF BTU)')
4050  FORMAT(1H0,5X,'LOAD ON THE A.C. UNIT',18X,'=',F9.2)
4060  FORMAT(1H0,5X,'HEAT FROM AIR TO STORAGE LAYER(DAY)       =',F9.2)
4070  FORMAT(1H0,5X,'RADIATION+CONDUCTION LOAD ON BUILDING     =',F9.2)
4080  FORMAT(1H0,5X,'TOTAL BUILDING HEAT LOAD                  =',F9.2)
4090  FORMAT(1H0,1X,'     ---------------------------------------------
     *--',/,13X,'(TOTAL HEAT FLOWS IN THOUSANDS OF BTU)')
4100  FORMAT(1H0,5X,'NET CONVECTIVE HEAT LOSSES/PANEL          =',F9.2)
4110  FORMAT(1H0,5X,'RADIATION HEAT GAIN/PANEL                 =',F9.2)
4120  FORMAT(1H0,5X,'NOCTURNAL RADIATION/PANEL                 =',F9.2)
4130  FORMAT(1H0,5X,'DIURNAL RADIATION FROM EACH PANEL         =' ,F9.2)
4140  FORMAT(1H0,5X,'HEAT FROM STORAGE LAYER TO AIR(NIGHT)     =',F9.2)
```

```
4150  FORMAT(1H0,5X,'PCWER(KW-HR)CCNSUMED TC BLOW AIR STREAM=',F9.2,/, 6
     *X,'----------------------------------------------------')
4160  FORMAT(1H0,5X,'PERCENTAGE CF COOLING EFFECT',11X,'=',F9.2)
4170  FORMAT(1H0,5X,'AVERAGE Y FACTOR',23X,'=',F9.2)
4180  FORMAT(1H0,5X,'AVERAGE Z FACTOR',23X,'=',F9.2,10X,'AVERAGE X FACTO
     *R =',F5.2)
4190  FORMAT(1H0,5X,'BUILDING AREA(THOUSAND SQ.FT.)',9X,'=',F9.2,//,
     *6X,'AVERAGE TEMPERATURE(DEGREES F)',9X,'=',F9.2,//,
     *6X,'(IDI-TT)AVG.',27X,'=',F9.2)
4200  FORMAT(1H0,5X,'MAXIMUM TOTAL AIR FLOW(LB/HR/1000)',5X,'=',F9.2)
4210  FORMAT(1H0, /,18X,'AREA(SQ.FT.)    AZIMUTH(RAD)    TILT ANG.(RAD)',/
     1,18X,'----------------------------------------------',/,6X,
     2'ROOF       =',F10.2,7X,F7.2,10X,F6.2,/ ,6X,
     3'SOUTH WALL=',F10.2,7X,F7.2,10X,F6.2,/ ,6X,
     4'WEST  WALL=',F10.2,7X,F7.2,10X,F6.2,/ ,6X,
     5'NORTH WALL=',F10.2,7X,F7.2,10X,F6.2,/ ,6X,
     6'EAST  WALL=',F10.2,7X,F7.2,10X,F6.2,/,6X,
     7'FLOOR AREA=',F10.2,/,18X,'---------------------------------------
     *-----')
4220  FORMAT(1H0,5X,'PANELS=',F7.0,5X,'PPERR=',F5.0,5X,'LENGTH=',
     *F6.0,5X,'C=', F5.2,5X,'CROOF=',F5.2,5X,'CWALL=',F5.2)
C
C     ------------------------------------------------------------------
C
C        END OF THE PRCGRAM
C
      IF(DATA(1).NE.STAT)GO TO 500
495   MONTH=MONTH+1.
      GO TO 70
500   CONTINUE
      END
C
```

```
C
C        SUBROUTINE GG CALCULATES THE SOLAR AZIMUTH. AZIMUTHS ARE
C        MEASURED CLOCKWISE WITH DUE SOUTH EQUAL TO ZERO DEGREES
C
      FUNCTION GG(V,W,X,Y)
      IF(W.NE.0.)GO TO 600
      GG=1.0
      GO TO 610
600   GG=(COS(V)*SIN(X)*COS(W)-COS(X)*SIN(V))/COS(Y)
      IF(GG.LE.1.0)GO TO 610
      GG=1.0
610   CONTINUE
      RETURN
      END
C
C
C     -----------------------------------------------------------------
C
C
C        SUBROUTINE G CONTINUES AFTER SUBROUTINE GG. G CALCULATES THE WALL-SOLAR
C        AZIMUTH FOR SURFACES WHICH ARE NOT IN SHADOW
C
      FUNCTION G(U,W,A)
      IF(W.EQ.0.)GO TO 620
      G=ABS(U-SIGN(ARCOS(A),W))*57.29578
      GO TO 630
620   G=U*57.29578
630   IF(G.LE.180.0)GO TO 640
      G=ABS(360.0-G)
640   IF(G.LT.90.0)GO TO 650
      G=90.0
650   CONTINUE
      RETURN
      END
C
C
```

```
C     -----------------------------------------------------------------
C
C
C        SUBROUTINE EX AVCIDS UNDERFLOW ERRORS IN THE CALCULATIONS WHEN
C        SOME EXPONENTIAL FUNCTION IS EVALUATED
C
      FUNCTION EX(T)
      IF(T.GE.5.)GC TO 700
      EX=EXP(-T)
      GO TC 710
700   EX=0.
710   CONTINUE
      RETURN
C
      END
C
C
C     -----------------------------------------------------------------
C
C
C        SUBROUTINE FLCWM CALCULATES THE WATER MASS FLOW IN THE PANEL
C        USING THE NEWTON-RAPHSON METHOD OF ITERATION
C
      FUNCTION FICWM(M,X1,X2,X3,X4,X5,S,PD,ALFA)
      REAL M
      DO 740 I=1,10
      IF(M.GT.S)GO TO 730
      M=X3/(100.*PD/SIN(ALFA)-X4)
      IF(M.GT.S)GO TO 730
      MNEW=M
      GO TC 75C
730   F=X3-X4*M*(EXP(X5/M)-1.)-100.*PD*M/SIN(ALFA)
      F1=-10C.*PD/SIN(ALFA)-X4*(EXP(X5/M)*(1.-X5/M)-1.)
      MNEW=M-F/F1
      F=ABS(F/F1)
      IF(F.LE.1.)GC TO 750
```

```
      M=MNEW                                                            SUBR0073
  740 CONTINUE                                                          SUBR0074
  750 FLOWM=MNEW                                                        SUBR0075
      RETURN                                                            SUBR0076
      END                                                               SUBR0077
```

APPENDIX F

CORRELATION DETAILS

The correlation shown in Chapter 4 can be found by averaging equation 25:

$$QV2 = MA*(1-EXP(-C4))*(TDI-TT)*CAIR \tag{25}$$

Equation 26 applied to (25) eliminates MA:

$$QV2 = U*AREA*PANELS*(TDI-TT)*(1-EXP(-C4))/C4 \tag{79}$$

Consequently, $(QV2)_{av}$ becomes:

$$1/\Delta t * \int_{o}^{\Delta t} QV2*dt =$$

$$U*AREA*PANELS * \int_{o}^{\Delta t} (TDI-TT)*(1-(C4))/C4*dt \tag{80}$$

It has been observed that the variability of the gradient (TDI-TT) is small for a period of one month. Hence, taking Δt = 1 month reduces (80) to:

$$(QV2)_{av} = U*AREA*PANELS*(TDI-TT)_{av} * \int_{o}^{\Delta t} \frac{(1-EXP(-C4))}{C4}\, dt$$

$$= (V)_{av}*(X)_{av}*(Z)_{av}*(Q+QWALL)_{av} \tag{81}$$

where the subscript (av) means average over one month. The parameter $(Z)_{av}$ is given by (33) and can be determined after knowing the comfort conditions and the average hourly load. The parameter $(X)_{av}$ is calculated by the program. The equation for this parameter is:

$$(X)_{av} = (TDI-TT)_{av}/(TDI_{av}-TREF) \tag{82}$$

The value of the auxiliary variable $(X)_{av}$ is a function of the percentage of cooling, PCOOL. From the simulation results it has been found that this relationship could be closely approximated by straight lines in the form

$$(X)_{av} = B*PCOOL+A \tag{83}$$

Table III shows the values of B and A during each summer month for Albuquerque and Phoenix. Physically, $(X)_{av}$ relates the monthly average temperature difference between the channels inlet and storage layer temperature, compared to the same difference when the stored water is at the fixed temperature TREF. In other words, numbers larger than one represent on the average, good cooling effect attained by the panels at night.

The parameter $(V)_{av}$ can be obtained from equation 36 making use of the average total load, $(Q+QWALL)_{av}$, and the average ambient temperature TA (to calculate TDI_{av}).

The percentage of cooling, PCOOL, is then obtained from equations 81 and 83:

$$PCOOL = A/((Z*V)_{av}^{-1} - B)$$

APPENDIX G

SUMMARY OF SOME RESULTS OF THE COMPUTER SIMULATION

TABLE VII

SUMMARY CORRESPONDING TO

ALBUQUERQUE

MONTH: JUNE

BUILDING	A.C. LOAD	TOTAL QWALL	TOTAL LOAD	QV2 (DAY)	QV2 (NIGHT)	QGEN	CWALL	CROFF
#1	79.1	58.0	170.3	91.4	0.7	4.50	0.40	0.40
#1	83.0	76.9	151.8	69.1	0.6	3.00	0.45	0.45
#1	95.7	74.9	162.3	66.8	1.1	3.50	0.30	0.30
#2	70.0	44.0	214.6	145.9	1.0	3.50	0.30	0.30
#2	129.9	86.9	306.2	176.8	1.1	4.50	0.30	0.30
#2	160.4	133.9	304.5	144.5	1.5	3.50	0.30	0.30
#2	237.7	176.1	346.7	109.5	3.2	3.50	0.30	0.30
#3	101.8	62.4	452.3	358.7	2.4	2.50	0.30	0.30
#3	237.1	84.2	708.2	473.0	2.1	4.00	0.50	0.50
#3	231.1	50.9	596.8	367.1	2.5	3.50	0.30	0.30
#3	209.7	48.3	516.3	307.9	2.0	3.00	0.30	0.30

Comments:

1. Heat flows in millions of Btu
2. Average ambient temperature: 82.8°F

TABLE VII

(continuation)

BUILDING	NOCTURNAL RADIATION	PCOOL	AVG Z	(TDI-TT)avg	MAX FLOW
#1	267.7	53.7	0.38	12.24	139.8
#1	267.2	45.5	0.39	12.88	146.7
#1	268.1	41.1	0.50	11.54	153.4
#2	266.7	68.0	0.21	14.30	150.2
#2	267.7	57.7	0.35	12.38	238.8
#2	268.2	47.5	0.46	11.51	282.2
#2	269.0	31.6	0.64	10.40	344.3
#3	265.8	79.3	0.13	15.83	288.4
#3	266.9	66.8	0.25	13.61	436.6
#3	267.4	61.5	0.31	12.90	353.5
#3	267.5	59.6	0.33	12.66	308.7

Comments:

1. Heat flows in thousands of Btu.
2. Mass flowin thousands of lbm/h.

TABLE VIII

SUMMARY CORRESPONDING TO

ALBUQUERQUE

MONTH: JULY

BUILDING	A.C. LOAD	TOTAL QWALL	TOTAL LOAD	QV2 (DAY)	QV2 (NIGHT)	QGEN	CWALL	CROOF
#1	96.2	63.3	179.3	84.4	0.5	4.50	0.40	0.40
#1	98.2	83.9	161.2	64.2	0.1	3.00	0.45	0.45
#1	111.2	81.7	171.9	61.9	0.8	3.50	0.30	0.30
#2	96.6	48.3	224.6	134.4	0	3.50	0.30	0.30
#2	163.3	94.8	321.5	160.6	0.8	4.50	0.30	0.30
#2	191.2	145.8	322.1	133.3	1.5	3.50	0.30	0.30
#2	269.5	191.7	368.0	100.9	2.0	3.50	0.30	0.30
#3	157.2	68.8	471.7	322.1	0	2.50	0.30	0.30
#3	318.3	92.8	737.6	426.8	0	4.00	0.50	0.50
#3	297.1	56.3	620.4	330.9	0.4	3.50	0.30	0.30
#3	266.9	53.5	537.0	277.7	0.6	3.00	0.30	0.30

Comments:

1. Heat flows in millions of Btu.
2. Average ambient temperature: 83.4°F

TABLE VIII

(continuation)

BUILDING	NOCTURNAL RADIATION	PCOOL	AVG. Z	(TDI-TT)avg	MAX. FLOW
#1	242.1	47.0	0.38	10.54	141.7
#1	242.1	39.8	0.39	11.01	150.6
#1	242.7	36.0	0.50	9.90	158.8
#2	241.6	59.9	0.21	12.37	148.5
#2	242.4	50.0	0.36	10.71	241.9
#2	242.8	41.4	0.46	9.94	291.7
#2	243.4	27.4	0.64	8.94	359.5
#3	240.9	68.3	0.13	13.66	286.8
#3	241.7	57.9	0.25	11.80	434.8
#3	242.1	53.3	0.31	11.19	353.3
#3	242.2	51.71	0.33	10.98	310.0

Comments:

1. Heat flows in thousands of Btu.
2. Mass flow in thousands of lbm/h.

TABLE IX

SUMMARY CORRESPONDING

TO ALBUQUERQUE

MONTH: AUGUST

BUILDING	A.C. LOAD	TOTAL QWALL	TOTAL LOAD	QV2 (DAY)	QV2 (NIGHT)	QGEN	CQALL	CROOF
#1	95.5	51.2	165.9	73.4	1.3	4.50	0.40	0.40
#1	93.9	68.2	144.6	53.8	1.2	3.00	0.45	0.45
#1	105.8	67.0	156.2	53.4	1.3	3.50	0.30	0.30
#2	102.2	38.3	212.4	116.2	1.6	3.50	0.30	0.30
#2	162.9	77.1	301.0	144.1	2.2	4.50	0.30	0.30
#2	182.1	119.8	293.9	117.8	2.5	3.50	0.30	0.30
#2	248.2	158.1	332.2	90.1	3.5	3.50	0.30	0.30
#3	179.7	53.5	451.4	291.1	2.0	2.50	0.30	0.30
#3	341.1	71.4	708.2	386.4	3.5	4.00	0.50	0.50
#3	318.2	43.0	600.2	301.3	4.0	3.50	0.30	0.30
#3	285.2	40.7	518.3	252.4	3.4	3.00	0.30	0.30

Comments:

1. Heat flow in millions of Btu
2. Average ambient temperature: 80.2°F

TABLE IX

(continuation)

BUILDING	NOCTURNAL RADIATION	PCOOL	AVG. Z	(TDI-TT)ave	MAX. FLOW
#1	185.1	44.3	0.37	10.09	139.7
#1	184.7	37.3	0.37	10.81	146.6
#1	185.3	34.2	0.48	9.56	153.24
#2	184.3	54.7	0.20	11.81	149.8
#2	185.1	47.9	0.34	10.10	238.1
#2	185.4	40.1	0.44	9.48	281.5
#2	185.9	27.1	0.62	8.53	343.5
#3	183.7	64.5	0.13	12.84	290.6
#3	184.5	54.6	0.25	11.06	440.8
#3	184.8	50.2	0.31	10.47	357.2
#3	184.9	48.7	0.33	10.27	312.8

Comments:

1. Heat flows in thousands of Btu
2. Massflow in thousands of lbm/h.

TABLE X

SUMMARY CORRESPONDING

TO PHOENIX

MONTH: JUNE

BUILDING	A.C. LOAD	TOTAL QWALL	TOTAL LOAD	QV2 (DAY)	QV2 (NIGHT)	QGEN	CWALL	CROOF
#1	116.5	103.7	216.0	104.6	0	4.50	0.40	0.40
#1	121.4	135.4	210.2	94.0	0	3.00	0.45	0.45
#1	144.1	129.4	216.8	77.8	0	3.50	0.30	0.30
#2	90.5	84.1	254.7	174.2	0	3.50	0.30	0.30
#2	188.9	153.8	373.2	194.2	0	4.50	0.30	0.30
#2	251.3	230.3	400.9	159.6	0	3.50	0.30	0.30
#2	366.2	299.1	469.7	113.4	1.0	3.50	0.30	0.30
#3	133.8	125.0	515.0	413.1	0	2.50	0.30	0.30
#3	308.1	171.6	795.4	519.4	0	4.00	0.50	0.50
#3	295.3	106.3	652.2	388.9	0	3.50	0.30	0.30
#3	276.5	102.1	570.1	325.5	0	3.00	0.30	0.30

Comments:

1. Heat flows in millions of Btu.
2. Average ambient temperature: 95.8°F.

TABLE X

(continuation)

BUILDING	NOCTURNAL RADIATION	PCOOL	AVG. Z	(TDI-TT)avg	MAX. FLOW
#1	302.7	48.4	0.45	11.28	152.9
#1	302.9	44.7	0.48	11.00	164.2
#1	303.7	35.9	0.58	10.12	169.8
#2	300.8	68.4	0.24	13.87	153.4
#2	302.4	52.1	0.41	11.67	257.1
#2	303.4	39.8	0.54	10.46	311.3
#2	304.6	24.2	0.71	9.22	383.0
#3	298.9	80.2	0.15	16.09	313.9
#3	300.9	65.3	0.28	13.47	476.3
#3	301.5	59.6	0.34	12.69	379.1
#3	301.8	57.1	0.36	12.35	344.4

Comments:

1. Heat flows in thousands of Btu.
2. Mass flow in thousands of lbm/h.

TABLE XI

SUMMARY CORRESPONDING

TO PHOENIX

MONTH: JULY

BUILDING	A.C. LOAD	TOTAL QWALL	TOTAL LOAD	QV2 (DAY)	QV2 (NIGHT)	QGEN	CWALL	CROOF
#1	170.1	100.7	216.8	63.0	0.1	4.50	0.40	0.40
#1	168.7	131.5	208.9	56.6	0.1	3.00	0.45	0.45
#1	186.4	125.9	216.2	46.1	0.1	3.50	0.30	0.30
#2	178.0	81.6	258.0	111.9	0	3.50	0.30	0.30
#2	289.8	149.6	376.2	118.4	.2	4.50	0.30	0.30
#2	337.2	224.0	400.3	95.0	.2	3.50	0.30	0.30
#2	432.9	291.0	467.3	66.3	.1	3.50	0.30	0.30
#3	342.5	121.3	524.3	286.5	0	2.50	0.30	0.30
#3	583.8	166.2	811.0	329.2	0	4.00	0.50	0.50
#3	527.7	103.1	667.2	241.6	0	3.50	0.30	0.30
#3	483.9	99.0	582.6	200.7	0.3	3.00	0.30	0.30

Comments:

1. Heat flows in millions of Btu.
2. Average ambient temperature: 94.3°F.

TABLE XI

(continuation)

BUILDING	NOCTURNAL RADIATION	PCOOL	AVG. Z	(TDI-TT)avg	MAX. FLOW
#1	203.7	29.1	0.44	6.54	157.5
#1	203.7	27.1	0.47	6.36	171.2
#1	204.2	21.3	0.57	5.76	179.1
#2	202.7	43.4	0.24	8.58	160.9
#2	203.5	31.5	0.40	6.85	265.5
#2	204.0	23.7	0.53	6.00	327.6
#2	204.6	14.2	0.70	5.20	406.5
#3	201.6	54.6	0.15	10.78	312.6
#3	202.8	40.6	0.28	8.24	478.2
#3	203.1	36.2	0.33	7.61	380.9
#3	203.2	34.5	0.36	7.35	336.9

Comments:

1. Heat flows in thousands of Btu.
2. Mass flow in thousands of lbm/h.

TABLE XII

SUMMARY CORRESPONDING

TO PHOENIX

MONTH: AUGUST

BUILDING	A.C. LOAD	TOTAL QWALL	TOTAL LOAD	QV2 (DAY)	QV2 (NIGHT)	QGEN	CWALL	CROOF
#1	141.5	97.5	212.1	83.0	0	4.50	0.40	0.40
#1	142.2	127.3	203.7	73.9	0	3.00	0.45	0.45
#1	161.4	122.1	211.2	62.2	1.0	3.50	0.30	0.30
#2	138.1	79.0	253.2	139.2	0	3.50	0.30	0.30
#2	238.2	145.0	368.8	154.8	0	4.50	0.30	0.30
#2	288.2	217.2	391.3	127.3	0	3.50	0.30	0.30
#2	388.3	282.2	456.3	92.1	1.0	3.50	0.30	0.30
#3	262.5	117.4	515.4	330.1	0	2.50	0.30	0.30
#3	460.4	160.6	797.3	414.2	0	4.00	0.50	0.50
#3	422.7	99.7	656.9	311.5	0	3.50	0.30	0.30
#3	389.8	95.8	573.3	260.8	0	3.00	0.30	0.30

Comments

1. Heat flowsin millions of Btu.

2. Average ambient temperature: 93.4°F.

TABLE XII

(continuation)

BUILDING	NOCTURNAL RADIATION	PCOOL	AVG. Z	(TDI-TT)avg	MAX. FLOW
#1	254.2	39.1	0.44	9.01	146.6
#1	254.3	36.3	0.47	8.84	155.8
#1	254.9	29.4	0.57	8.14	162.6
#2	252.8	55.0	0.24	11.12	157.0
#2	254.0	42.0	0.40	9.33	248.6
#2	254.7	32.5	0.53	8.39	298.2
#2	255.5	20.2	0.70	7.47	365.8
#3	251.4	64.1	0.15	12.83	304.1
#3	252.9	51.9	0.28	10.69	462.4
#3	253.4	47.4	0.33	10.06	370.7
#3	253.6	45.5	0.36	9.81	326.2

Comments:

1. Heat flows in thousands of Btu.
2. Mass flow in thousands of lbm/h.

APPENDIX H

ECONOMIC CONSIDERATIONS

This appendix deals with the development of some equations to determine the economic breakeven point of the panels alternative.[21] The procedure followed is to transform the initial capital invested in panels to a uniform payment series, compounded annually for a determined service life. The resultant amount should be equal to the savings obtained in electric power, computed using regional electric costs.

H.1. Equations of the economic analysis.

The annual charges, ACH (expressed in \$ per year), can be found from:

$$ACH = UPC * PANELS * AREA * CR \qquad (84)$$

where CR, the capital recovery factor, is given by (37). The variable UPC represents the sum of the charges for installation ($\$/ft^2$), plus the unit cost of the panel ($\$/ft^2$), minus the savings obtained in construction costs replaced.

On the other hand, the annual electricity savings, AES, expressed also in \$ per year, are obtained by:

$$AES = (QV2)_{ann} *EC/(COP * 3413) \qquad (85)$$

where EC is the electrical cost in \$/Kw-h, and COP is the coefficient of performance of the A/C unit. $(QV2)_{ann}$ refers to the annual cooling savings expressed in Btu.

Now, combining equations 24 and 25 we get another expression for C4:

$$C4 = -Ln(1-XX*PCOOL/CAIR/(TDI-TT)) \quad (86)$$

The annual savings in cooling can be approximately determined in the following way:

$$(QV2)_{ann} = U*AREA*PANELS*(TDI-TT)_{av}*(1-EXP(-C4))* TIME/C4 \quad (87)$$

where TIME refers to the number of hours of the year for which cooling is needed. The other variables in (87) are annually estimates of the typical performance of the system.

Using(86) and (87) into (85) yields:

$$AES = -EC*U*AREA*PANELS*TIME*XX*PCOOL/(COP*CAIR*3413*Ln (1-XX*PCOOL/CAIR/(TDI-TT)_{av})) \quad (88)$$

Hence, the breakeven point for electricity cost, BEEC, can be found by matching AES and ACH (equations 84 and 88):

$$BEEC = -3413*COP*CR*UPC*Ln(1-XX*PCOOL/CAIR/(TDI-TT)_{av})* CAIR/(U*TIME*XX*PCOOL) \quad (89)$$

H.2 Typical analysis.

Consider the following typical case:

i) UPC= $\$7.00/ft^2$

ii) $(TDI-TT)_{av} = 12°F$

iii) COP = 2

iv) U = 1 $Btu/ft^2h°F$

v) TIME = 3600 hours

then, the resultant equation (89) is:

$$BEEC = -0.654*Ln(1-1.695*PCOOL)*CR/PCOOL \qquad (90)$$

Figure 16 shows the equation 90 plotted. We can easily see the requirements that have to be met in order to be economically feasible, concerning electricity cost and financing rates (CR).

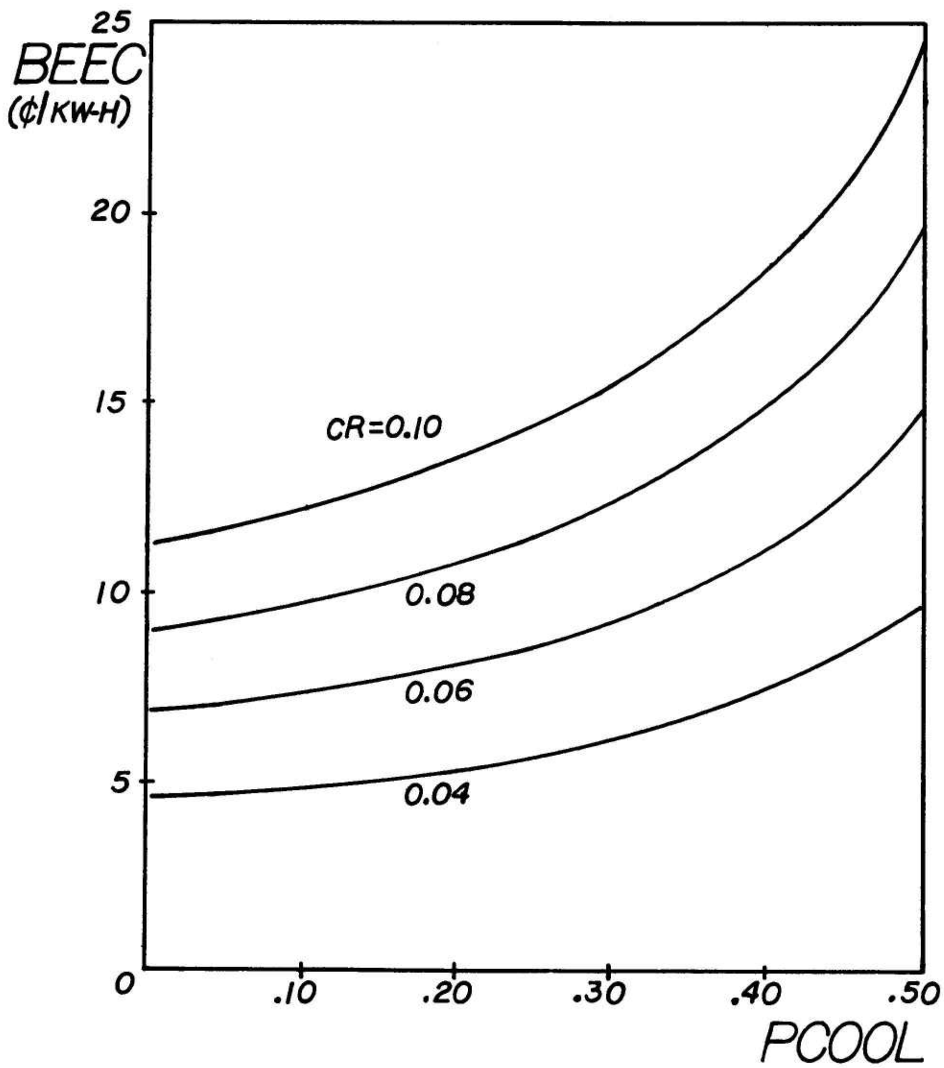

FIGURE 16

Economic Breakeven Point of Electricity Cost

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