MASTER

DOE/CS/341 1-4

Project Report

The SunCatcher and Cool Pool

Living Systems

JONATHAN HAMMOND

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

Project Report

The SunCatcher and Cool Pool

DISCLAIMER

This book was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise, does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

Living Systems

JONATHAN HAMMOND

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

JHS

This work has been supported by the
Solar Heating and Cooling Research
and Development Branch, Office of
Conservation and Solar Applications,
U.S. Department of Energy, under
Grant #EG-77-G-04-4153.

Author: Jonathan Hammond with
James Plumb
Karen Crowther
David Batts

Editing: Caesar Adams
David Batts

Graphics: James Plumb
Ann Wickham

Hardware Research & Development:
Dennis Long
Bruce Melzer
David Batts
Rod Hobbs

Data Acquisition System:
David Batts
Bruce Maeda
Paul Grant

Mathematical Consultant:
Bruce Maeda
Karen Crowther

Grant Administration:
Mary Oak

Report Production:
Ann Wickham

March 1, 1981

TABLE OF CONTENTS

1. <u>Introduction</u>	1	5. <u>Appendix II - 1979 Math Models</u>	80
2. <u>The Suncatcher</u>	4	Thermosiphoning	81
Seasonal Performance	9	Radiation into the Water Column	82
Performance Estimates	11	Evaporation	89
Construction	14	Convection into the Roof Pond	90
Architecture and Aesthetics	15	Conduction	91
Sacramento Suncatcher	16	Net Radiation into Pool	92
A Davis Duplex	18	Building Heat Load	92
The Porter House	19	Computer Simulation	93
The Sunrise Suncatcher	20	Conclusion	98
The Maeda House	21	Sizing for Different Climates	98
Suncatcher Potential	23	Fortran Program	99
Conclusion	24	Wet-Bulb Temperature Map	100
3. <u>The Cool Pool</u>	26	Cost Estimate	101
1977 Component Development			
and Performance	31		
Experiments #1-6	34		
Conclusions	40		
1978 Test Building Experiments	41		
Experiments #1-3	43		
Conclusions	52		
1979 Test Building Experiments	52		
1980 Test Building Experiments	56		
Detailing	59		
4. <u>Appendix I - 1978 Math Models</u>	70		
Interior to Water Column	70		
Column to Pool Heat Transfer	72		
Pool to Ambient	72		
Rules of Thumb	74		
Definition of Terms	77		

Project Report

The SunCatcher and Cool Pool

Introduction

The natural heating and cooling of buildings is a relatively widespread and accepted technique for conserving energy, especially in private residences. More recently, large scale buildings of various types have also begun to use solar and other innovative techniques to provide some of their energy needs. The assurance of an end to fossil fuel supplies and the continuing political uncertainties which surround even the short term supply of conventional energy resources reinforce the need to increase and improve this use of effective, new energy conserving strategies. The following report describes two devices invented by Jonathan Hammond which we believe represent such a refinement of existing alternative technologies.

The Suncatcher is a solar concentrator which can be used with a variety of thermal storage elements in either private housing or commercial scale buildings. Because the Suncatcher provides solar access through the roof of a structure, it gives designers considerable flexibility in the design of passively heated buildings. By raising the collector area, the Suncatcher also makes solar heating possible for buildings where south wall passive systems will not work. The Cool Pool is the Suncatcher's complement for natural cooling.

The Cool Pool, we believe, has great promise, especially in hot, arid climates where nighttime ventilation alone is not adequate for cooling. The Cool Pool is a shaded evaporative roof pond which thermosiphons cooled water directly into storage areas for interior cooling. Initial tests indicate that it is a powerful, passive cooling device suitable not only for private residences but for commercial scale buildings as well.

Together the Suncatcher and Cool Pool provide some interesting, new possibilities for designers of passive buildings. Both devices are relatively easy to build using conventional materials and construction techniques; their installed costs compare favorably with existing passive systems; and their flexibility makes them appropriate for numerous architectural applications, even applications where site constraints, climate or building uses make passive heating or cooling difficult with more conventional techniques. Either component can be used separately or both can be combined into an integrated system.

The report which follows describes the development of the Suncatcher and the Cool Pool and the results of various experiments on their performance.

THIS PAGE
WAS INTENTIONALLY
LEFT BLANK

The SunCatcher

Passively heated and cooled buildings require substantial thermal mass, solar access, and some way to regulate the amount and timing of sunlight striking their thermal storage areas. Since south facing glass offers the most direct sunlight, thermal storage elements are usually placed in front of south windows, where they effectively intercept sunlight, but also block views and hinder natural lighting. These requirements can restrict design and, where south solar access is impeded, preclude or limit the use of passive heating systems.

The Suncatcher, a simple solar concentrating device developed by Living Systems in 1976, responds to these problems in several ways. The Suncatcher captures light, which enters clerestory windows, and directs it into a house. Light entering a house in this way shines directly onto thermal storage elements that can be placed at the back of a south facing living space. This leaves south wall windows open for views, for natural lighting or for additional heat gain. The high, clerestory windows are also less likely to be shaded by nearby trees, buildings or other solar obstructions than ground level windows. This makes the Suncatcher promising for the passive heating of buildings with limited solar access. By solving this problem of solar access simply and directly, the Suncatcher also allows designers considerable architectural flexibility.

The Suncatcher cone design was the extension of an idea of Jon Hammond's for concentrating heat onto a domestic hot water heater. While thinking about the heater problem, Hammond decided that its small, concentrating cone device could be enlarged and made part of a building's passive collector system. Further work by Living Systems on the cone's aperture configuration and window size resulted in the basic Suncatcher design and operation now in use on several buildings.

The Suncatcher was named by the British consul to Guatamala during a game of pool with Hammond while they were traveling to the temple of the Sun at Tikal.

The Suncatcher's cone shape was extensively studied to maximize its performance in both summer and winter. We found that a cone designed to capture low angle winter sun also reflected away the high summer sun. The Suncatcher thus effectively regulated itself in meeting the varying needs of winter and summer. The cone developed for the latitude and climate of the Sacramento area was a 19° cone raised 22° from horizontal.

Following development of the cone, the Suncatcher was next integrated directly into the roof line of a hypothetical structure and all the system's elements studied. We considered the size of the Suncatcher's windows, materials and methods for constructing the cone, the kind of thermal mass to be used and its placement within the building, the system's seasonal requirements, and methods for controlling heat gains and losses with insulating drapes and shutters.

After considering various size Suncatcher windows, 4' windows were chosen as appropriate. The cone itself is constructed of standard building materials and requires no special fabricating skills. Aluminum sheet metal roofing material which lines the cone provides a durable, exterior reflective surface. For thermal storage, Living Systems uses the water-filled galvanized steel culvert pipe. The 16 gauge galvanized metal and cylindrical shape have sufficient structural integrity to resist bursting pressure, seismic loads and corrosion. Properly installed and properly placed, these cylinders are effective heat sinks for both solar heating and natural cooling.

The Suncatcher's design regulates its seasonal operation by partially controlling unwanted summer heat gains. For their daily operation, passive systems also require insulating shutters and drapes. Living Systems had developed a specially designed drape, sealed on all sides. The drape is made of 2 layers of reflective foil fabric sewn so that an insulating air space is maintained between layers when the drape hangs. This drapery system uses standard hardware and materials and has a low installed cost.

Seasonal Performance

The Suncatcher's winter performance was studied by graphically projecting sun angles from November through March.

In the figure below, the 26° angle at noon on December 21 was used. The amount of sunlight received through the Suncatcher's clerestory windows at that hour will be approximately 1.5 the amount received by an equivalent south facing window. Put another way, a 4' Suncatcher window will receive (at noon) the same amount of insolation as a conventional south facing 6' window. This concentration of usable sunlight provides additional design flexibility and increased heating efficiency.

WINTER DAY

During the day, the shutter is opened to allow the low angle sun in.

WINTER NIGHT

In the evening, the shutter is closed to prevent the loss of heat out the Suncatcher windows.

SUMMER DAY

The shutter is closed to prevent heat gains. The window is shaded by the exterior overhang.

SUMMER NIGHT

The shutter is opened in the evening and the house is opened to evening breezes to cool the thermal mass.

High angle summer sun (72.5° at noon on July 21) will be rejected by the cone's geometry and direct sunlight entering the cone will be reflected away. Indirect radiation striking window glass can be prevented from entering interior spaces by the insulated shutter with its outward facing, reflective surface.

Performance Estimates

The following table, Multiplier for Suncatcher Radiation, shows the per cent of Suncatcher insolation over conventional windows.

MULTIPLIER FOR SUNCATCHER RADIATION

% of Normal South Facing
Radiation for January

East Clerestory	138%
Middle Clerestory	149%
West Clerestory	146%

Until more refined hour by hour formulas are available, it can be assumed that winter radiation through a Suncatcher window is 40-50% higher than through an ordinary window.

Although the Suncatcher is designed to provide shading to the clerestory window in June, the reflective roof surface will increase the diffuse radiation entering the window. Even so, 80% of the summer solar radiation is still rejected by the cone's geometry. Reflective insulated shutters can be installed to further reduce this unwanted heat source in the summer.

The following chart shows the Suncatcher's winter performance. The data shown was taken from the Maeda house (described on pp.21-2) under a monitoring program sponsored by the Department of Energy. During the winter monitoring period, November 1977 to May 1978, weather conditions were unusual. Temperatures were warmer but cloud cover more extensive than normal. The Suncatcher window system dramatically increased the

amount of insolation received by the house. Compared to other houses in Davis, including other solar houses in the subdivision where the Maeda house is located, the Maeda Suncatcher performed well.

Minimum outdoor temperatures during the winter were in the thirties and forties; indoor temperatures averaged 68°F during the day. At night the temperature dropped only slightly, to an average low of 62.5°F. For only five days throughout the 1978 winter did indoor temperature drop below 60°F. The lowest inside temperature recorded was 58°F. These lower temperatures occurred after a day of very low outside temperatures and reduced insolation. Despite these conditions, the Maeda house achieved its daily daytime average of 69°F without insulating drapes and shutters. Performance should be even better with the shutters and drapes installed and operating.

The figure below, Solar Contribution, shows the calculated required heating load, the amount of auxiliary heating used and the per cent of solar heating. The calculated load is less than the actual load since the building heat loss coefficient assumed insulated windows. (The window insulation was not installed until spring.) With window insulation the solar contribution is expected to be about 10% higher.

Preliminary data for another Suncatcher house, the Tarkington Suncatcher in Davis, California, indicate a solar fraction of 75%. The Tarkington house is being extensively monitored by IBM under a subcontract from the Boeing Corporation for HUD.

The first houses built using the Suncatcher located their thermal mass 4' to 5' behind the line of clerestory windows. Later designs, based on data taken from these actual installations, move the water-filled storage cylinders farther back from the windows. Sunlight striking the lower parts of a water-filled container is more effective in heating the whole cylinder than light striking only its top.

Analysis indicated that only the top four feet of a ten foot tall cylinder received direct sunlight at noon from December through March. Graphic analysis of sun angles indicates that a cylinder placed 11'0" behind the windows will receive direct sunshine at noon in those same months across all 10' of its length.

The stratification chart below shows the effect of concentrating sunlight on only the tops of storage cylinders. Bathing the entire length of the tube in light helps distribute heat over the entire length and reduces stratification.

Construction

To date, six contractors have built Suncatcher houses. The contractor's supervision of various subcontractors must be careful since roofers and window installers are not used to working together, as they must in constructing the Suncatcher and installing its windows. According to our experience, problems are likely to occur in installing the flashing under the windows. Otherwise, neither special skills, materials nor equipment are required for building the Suncatcher house.

Architecture and Aesthetics

The dramatic effect of the Suncatcher on a building's roofline is apparent. One or several of the elevated cones will announce the building's function as a sun catching structure. The building's aesthetics are thus grounded in its form and function which are made apparent in its exterior design.

Interior spaces are similarly designed to highlight the house's operation as a passive building. Dramatically lit spaces set deep within living spaces, sun-filled atriums and light wells characterize some of the various interior designs possible in Suncatcher houses.

Some of these designs, for buildings already constructed, appear on the following pages.

Sacramento Suncatcher

The floor plan shown here was developed for a PG&E demonstration house, designed to meet 80% of its heating and 100% of its cooling with its passive system. The Sacramento Suncatcher admits light through its clerestory windows and other south facing glass. Heat is stored in the house's exposed concrete slab and in a row of water-filled cylinders which screen the living room from the dining room.

Additional thermal mass is provided by eight sculptural concrete screens placed on the west wall of the entryway, by four short, water-filled tubes forming a half-wall opposite the screen and by various pieces of thermal mass furniture.

All large window areas, except the Suncatcher windows which use shutters, have insulating drapes.

The following chart, based on data provided by the Pacific Gas and Electric Company, shows the Sacramento Suncatcher's performance on the last of five days during which temperatures consistently registered above 100°F, peaking on the last day at 109°F. Without mechanical cooling, the house remained within the 70°F to 80°F range, except for the 109°F day when interior temperatures peaked at 82°F. On this particular day PG&E and SMUD customers broke all previous electrical power consumption records, primarily for air conditioning. In addition, during the five-day period, more than usual numbers of visitors toured the house, including over 350 on one day.

Despite these exceptionally heavy heating loads, the house remained comfortable without any back-up mechanical air conditioning.

The Sacramento Suncatcher took only 2 1/2 months to build, a short time for any house and especially for one with unconventional requirements. Careful coordination of building schedules and providing for early installation of the solar cylinders and other solar features made it possible to build this house in less time than that required by conventional housing.

Sacramento Suncatcher Summer Performance Data

A Davis Duplex

This Davis duplex, designed as low income farmworker housing, was the first design to use the Suncatcher system. The Suncatcher and the solar cylinders are placed in the middle of the house to form the back wall of the living room. This central sunny space receives natural light

on both its north and south sides. The main entryway opens directly into the high space below the Suncatcher; circulation to other rooms passes through this central area. Additional shorter tubes are placed directly behind south glass to provide thermal storage, while opening up views and letting light into interior spaces.

Since this Davis project required several duplexes, the design had to be flexible enough for placement on either the north or south side of the street. Slightly changing the entry made such siting possible. The building also had to be low cost so that the budget limited the choice of materials. Everything except the cylinders, the Suncatcher shutters and the insulated drapes in the duplexes is of conventional, low cost materials. As a result, architectural interest relies exclusively on the overall design of the building and the form, color and natural lighting of its interior spaces.

Suncatcher Solar Duplex

Designed for the City of Davis

The Porter House

The Porter house plan resembles that of the Davis duplex but is much larger (1872 sq. ft.). This house is set on an east/west lot, the most difficult lot orientation for a passive solar house. The Suncatcher solves this orientation problem by providing solar access above trees and adjacent houses.

As in the Davis duplex, the entry is directly into a hallway below the Suncatcher. There are, however, two significant differences. The Porter residence introduces another, smaller clerestory over the living room to provide more light deep within the house. The Porters wanted the living room to double as a meeting place; its natural day lighting works well with this intention. In addition, the thermal storage cylinders have concrete infill which makes them into a solid wall. The heat stored in the cylinder wall radiates to rooms on both the north and south sides for more even heating of the entire house. The exposed concrete aggregate of the infill provides a textural counterpoint to the smoothly surfaced, fluted spirals of the solar cylinders.

The Sunrise Suncatcher

The Sunrise Suncatcher, designed for Steven Kurtz, applies the experience gained from previously completed Suncatcher houses. By widening the entry hall of the Kurtz house to 11'0", the Suncatcher space becomes a dining room and hearth area. Instead of being merely a tall, narrow hall the Kurtz Suncatcher space is transformed into a sunny atrium. Setting the cylinders farther back in this atrium space allows the sun to strike them lower. This improves their performance and partially solves the problem of stratification. The Kurtz cylinders also use concrete infill to form a wall like that in the Porter house.

To dramatize the house's roofline, the roof over the bedrooms is raised at the north end. Since the house is set on a hill, the raised roofline complements the sloped topography of the site.

The Maeda House

In this house for Bruce T. Maeda of Davis, California, the placement of rooms is distinctly different from the previously described houses. Living spaces are located on the north and bedrooms on the south. This arrangement provides south glass and thermal mass for each room. The north facing living room also takes advantage of views towards a greenbelt.

Data from the Maeda house indicate that it performs differently than other Suncatchers because several windows open directly into north rooms. Temperatures in the north facing living room were warmer than north bedrooms in other Suncatchers, but south bedrooms were more likely to overheat than south living spaces in other designs.

The following chart compares the energy consumption of the Maeda house to the average energy consumption of Davis houses, houses in selected, nearby communities and other houses in the solar subdivision where Maeda lives.

Monitoring of the Maeda house, the most complete to date, clearly indicates its superior performance, even when compared to other solar and energy conserving houses.

Suncatcher Potential

To evaluate the Suncatcher's potential, various cone configurations were analyzed for three different latitudes. Generally, we found that the cone needs to become wider and to be raised from the horizontal at lower latitudes. In the Gulf Coast area, with its high temperatures and humidity, the use of the Suncatcher is not recommended because cooling is the significant need. In areas with high snowfall, more than two feet annually, snow build-up in the cone would also create difficulties.

Sun angle studies of Suncatcher houses for three latitudes, 32°, 40° and 48° North, are discussed below.

32° North Latitude

In the southern states, where sun angles vary widely between 9 and 3 o'clock, the cone is enlarged to 27°. This design will block neither the low morning sun nor the higher sun at noon. Since the sun is also higher in the sky at lower latitudes, the cone is raised higher, to 31.5°.

40° North Latitude

In the middle regions of the United States where snowfall is minimal, a Suncatcher cone of 19° raised 22° from the horizontal is appropriate. All the houses described in this report use this design.

48° North Latitude

In northeast climates, with low winter sun angles, the cone is lowered to 16° above horizontal. This results in a flat roof so that drainage can run away from the Suncatcher's windows.

In areas with low cooling demands, the overhang can be shortened to enhance heating throughout the spring and fall.

32° NORTH LATITUDE

40° NORTH LATITUDE

48° NORTH LATITUDE

Conclusion

The Suncatcher is a valuable addition to the art of passive design. It solves specific problems of solar access and provides a powerful mechanism for concentrating sunlight and enhancing natural heating. The Suncatcher's geometry regulates its performance well enough so that a simple shutter system is all that is needed to control heat gains or losses.

Because it is a flexibly located, simple and powerful solar collector, the Suncatcher can be successfully applied under various building and site constraints.

As an integral part of a building's form, the Suncatcher also offers exciting, new possibilities for developing a distinctive solar architecture which makes explicit the connection between solar access and thermal storage. The public response to Suncatcher houses so far indicates that their visible, simple, reliable energy conserving technology is not only acceptable, but appealing as well.

THIS PAGE
WAS INTENTIONALLY
LEFT BLANK

The Cool Pool

Most buildings with substantial interior mass use ventilation, primarily at night, for passive cooling. In climates like those of Davis in California's Central Valley, where nighttime temperatures by 8:00 PM will usually drop to 78°F, and as low as 56°F at dawn, night ventilation works very well. This is especially true for residences where properly designed natural ventilation can easily provide the 10-15 air changes per hour required for effective cooling. Providing such large numbers of air changes has, however, inherent problems, the first and most obvious being available cool air. Large portions of the Central Valley and other western regions have climates where summer temperatures even at night rarely reach the comfort range, much less the cooler temperatures required for effective, natural cooling. For example, the mean low temperature in Bakersfield in August is 70°F and 76°F in Indio.

Even with cool air available, the volume of air required creates additional problems, especially for commercial scale buildings. For instance, airborne dirt can create maintenance problems; the need for open windows and doors complicates building security; and ventilating large spaces can compromise other building requirements. For these and other reasons, ventilation alone is not always adequate for passive, natural cooling.

The Skytherm is one passive cooling system that can be used where ventilation is either impossible or inadequate. For its operation, however, the Skytherm requires expensive movable

insulation. The Sherwood house, illustrated above, uses a modified Skytherm system for cooling.

Another solution for natural cooling is the system installed in the Dickinson house. This house in Chico, California uses a 12'X 12' movable lid that opens over a skylight. In summer

the skylight is removed to expose water-filled cylinders directly to the cold night sky for cooling.

Unlike the Skytherm system or the system described above, the Cool Pool requires neither movable insulation nor manual operation, and it has the ability to cool large spaces effectively, even in climates where nighttime ventilation is limited.

The Cool Pool conceptualization and development spanned five years from 1976 through 1980. Work done under D.O.E. funding spanned the four years from 1977 through 1980. First year D.O.E. work included component development and associated performance testing. Second year D.O.E. work involved component integration on the Suncatcher/Cool Pool module building and testing of the integrated system. In 1979 mathematical modelling of the Cool Pool was developed through further testing--particularly thermosiphon testing. The fourth year, 1980, the cool pool components were made modular to a certain extent, and were engineered for ease of construction and maintenance and for cost effectiveness. Performance testing showed that the new construction designs had no adverse effects on performance, and may have actually improved performance of the Cool Pool.

The following section describes the development of the Cool Pool from conceptualization through current state of the art.

The Cool Pool is an evaporating, shaded roof pond which thermosiphons cool water into water-filled columns within a building. The system works primarily by evaporation. A shade screens the pool's water from direct sunlight, and is designed so that free air movement over the pool's surface promotes evaporation and cools the water.

This cool water can be used in several ways: the bottom of the tank can rest (as in the Skytherm system) on the ceiling of the space to be cooled; or the pool's cool water can be connected directly to heat exchangers within the building. The cool water can then be pumped or thermosiphoned from the pool to the exchanger.

While the Cool Pool has significant advantages over the Skytherm, its invention in 1975 by Jonathan Hammond was inspired by Hay's work on the Skytherm and by Loren Neubauer's work on north sky radiation. Neubauer discovered that shading a dry object from direct sunlight while allowing it to radiate to the cool north sky resulted in the object's remaining a few degrees cooler than ambient air both during the day and at night.

AFTER NEUBAUER, ET AL. 1964

Hammond constructed a shade like Neubauer's but substituted a 12" deep, open pool of water for Neubauer's dry object. The water at the bottom of the tank stayed 35°F cooler than the maximum ambient air temperature. Marshall Hunt, under a HUD innovative projects grant, replicated this experiment in Indio, California, under even more severe conditions, but with similarly encouraging results.

As a result of these experiments, Living Systems built a demonstration Cool Pool room, under a grant from the California Energy Commission, for California's 1976 State Fair in Sacramento. The room used the bottom of a Cool Pool for its ceiling. In addition, 1' diameter pipes were to be attached to the pool to provide additional radiative surface--an idea whose time had not quite come, given the constraints of the Fair's set-up schedule for exhibitions. Despite several all night efforts, we found it impossible to create a watertight seal between the pipes and the tank. As a result, just the roof pond in essentially the Skytherm configuration was used for cooling the exhibition room.

Even with this limited system and an average 200 visitors daily, the room maintained temperatures below 80°F during 110°F weather. These very encouraging results inspired the DOE financed research described below.

1977 COMPONENT DEVELOPMENT AND PERFORMANCE

The success of the Cool Pool room used at the California State Fair prompted the research program undertaken by Living Systems in the summer of 1977 for the Solar Research Division of the Department of Energy. The program's goals were to develop the Cool Pool for use in actual installations by establishing its cooling potential and predicting its performance so that it could be sized to meet the various cooling needs of buildings in different climates. The pool's architectural requirements also had to be evaluated and appropriate hardware developed to make the system cost effective and reliable.

Developing a Shade

From August to October 1977 Denny Long and Jim Plumb of Living Systems conducted a series of experiments to test the effects of different shade configurations on the Cool Pool's performance. Data was also gathered on the relative effects of radiation to the north sky and of evaporation, using both free and forced convection. To test these variables, experiments using styrofoam ice chests as Cool Pool tanks were designed, and various shade configurations constructed to provide differing amounts of north sky view and evaporation.

Purpose of the Preliminary Experiments

We already had preliminary data for an optimally designed shade, but that shade was too cumbersome to use on a full-scale building. What we wanted from the experiments was data we could use to design practical, full-scale shades for potential installations.

Procedure

Two different size styrofoam boxes were used as water containers. The smaller container, used on most of the tests, held approximately 20 lbs of water, was 1/2" thick and had interior dimensions of 8"X 13" with a 5" depth of water. Later, a tank holding 200 lbs of water was used. The larger tank had 2" thick walls, measured 20"X 26", and had a water depth of 1 foot.

To test the effects of radiative cooling to the north sky, two containers, one covered with plastic to prevent evaporation and one open, were placed within various shades. All the shades were constructed of 3/8" plywood painted white on the outside and black on the inside. All shades had south-side louvers or were open on the south to allow cross-ventilation.

Each test was run for several days and, whenever possible, during weather over 100°F. During the day temperatures were recorded at two hour intervals; wet bulb temperatures, wind and cloud cover observations were also taken. Recording hygrothermographs at the test area provided the ambient temperature curves, and a sling psychrometer or calculations using dry bulb temperatures and humidity provided wet bulb temperatures. Temperatures were taken from locations at the top, middle and bottom of each tank. Only the temperature from the pool's bottom appears here since those temperatures are the coolest and come from the water to be used for interior cooling. The results of six of the most interesting experiments are presented below.

Experiment #1

Object--To test the effect of limiting north sky view on both evaporating and closed tanks, three shading devices that completely shaded the two tanks from direct solar radiation on June 21 were tested. Angle X (shown in the test shade drawing) was 45°, 60° and 90° for these three devices.

Results--The performance of the evaporating tanks changed little, seldom more than one degree, as a result of varying the amount of sky view. The evaporating tanks, in contrast, were consistently 8°F lower than the closed tanks.

Comparing the closed tanks, which eliminated evaporation, makes evident the differences in radiative cooling. On August 30 the maximum difference between the shades was 4°F.

Conclusion--The small differences in temperature caused by varying the amount of north sky view are offset by difficulties in building shades with maximum sky view. For example, the 90° shade has to be 5'10" tall to shade a tank 2' wide. The 45° shade is only 2'8" tall for the same size tank.

Experiment #2

Object--Experiment #2 was designed to test a low profile, louvered shade with reduced sky view.

On warmer days the covered tank was 10° warmer and on cooler days only 6°-8°F warmer than the open tank.

Results--The figure below compares the results of a louvered shade with vanes set at 60° to the 60° shade without louvers used in the previous experiment. The maximum difference was 4°F for both evaporating and closed tanks.

Conclusion--The louvered shade with its low profile solves specific architectural problems but requires sacrificing some of the Pool's cooling power.

Since the spacing and material of the louvers could affect cooling, further tests on these two variables were carried out in Experiment #3. Varying the shade's configuration had a maximum effect of 4°F.

Experiment #3

Object--A variety of shading configurations intended to enhance the pool's performance by augmenting radiative or evaporative cooling were tested.

Results--Various shades (a 45° saw tooth shade, a 45° louvered shade, and a 45° shade with south scoop), tested with open tanks of water, performed with little difference, $\pm 2^\circ\text{F}$ separating them.

Conclusion--None of these attempts to enhance natural air circulation had any significant effect.

Experiment #4

Object--Experiment #4 used forced convection to increase evaporation in an attempt to improve the pool's performance.

Results--A 6" diameter fan was placed 9" above the water level of the open tank within the 60° shade shown in the test shade drawing. The shade with the fan was compared to a 45° shade without a fan. The fan was turned on around 6:00 PM each evening, run overnight and turned off at about 8:00 AM.

Forced convection achieved much more dramatic results than the natural ventilation described in Experiment #3. Typically, the pool with the fan was 5°-8°F cooler. On September 9 the device with the fan achieved a water temperature of 46°F after a day with a 93°F daytime maximum and a 45°F wet bulb nighttime minimum. During the day, the water in both devices reached similar maximums: 74°F, for instance, on a day with a 104°F maximum ambient air temperature. Running the fan during the day warmed the small amount of water in the pool's tank.

Conclusion--Using the fan at night improves the system's performance but increases first costs and life-cycle costs because of the fan's purchase price and the cost of energy consumed in running it. Despite these additional costs, the fan device is promising for use in severe climates or as a "back-up" on unusually hot nights in gentler climates.

Increased wind velocity generated by the fan had two effects: it increased the heat transfer coefficient from the air to the pool and it also increased the evaporation rate. The first effect resulted in warmer water during the day if the fan was used when air temperature was high, but cooler water at night when air temperatures were lower. Increased evaporation, the second effect, was beneficial day and night.

Since the first effect, increasing the heat transfer coefficient, dominated, the net result of using the fan in the day was warmer pool water.

Experiment #5

Object--To test the possibility of completely eliminating the shade and relying instead on thermal mass stored in cylinders within the building, Experiment #5 compared a Cool Pool with a 45° shade to a completely unshaded tank.

Results--During the day the water in the unshaded tank was often 20°F warmer than the water in the shaded tank, but typically 3-4°F cooler at night. This ability to reject heat at night was attributed to the unshaded tank's unimpeded night sky view and unimpeded east-west breezes.

Conclusion--The unshaded tank is less expensive to build, and despite daytime water temperatures as high as 94°F, its ability to reject heat at night gives it interesting possibilities for some applications. For example, if enough internal storage provides daytime cooling, this configuration (without a shade) could be used for nighttime cooling only.

Experiment #6

Object--Experiment #6 was designed to compare directly the systems tested in the previous five experiments on a scale more appropriate to an actual installation. The size of the tank was increased to accommodate 200 lbs of water, rather than the 20 lbs used in the original five tests.

Results--The results of the following four alternatives were evaluated:

- The typical Cool Pool configurations (shown in the test shade drawing on page 35) using a simple shade with no moving parts.
- The typical Cool Pool with a 6" diameter fan placed 9" above water level run at night between 6:00 PM and 8:00 AM.
- A tank of water with a canvas shade removable day or night to provide maximum night sky view and access to breezes.
- A Skytherm system with a removable lid closed during the day and opened at night.

Conclusion--Because of their additional thermal mass, the larger tank's temperatures fluctuated less from day to night.

The device with the fan was consistently the coolest, cooling more at night and holding its coolness through the following day.

The removable canvas shade system improved the simple Cool Pool's performance by only 1°-2°F.

The Skytherm system started the day with warmer water but eventually settled out: within 2°F warmer than the device with the fan and 2°F cooler than the Cool Pool without fan.

The Cool Pool with a simple shade performed well, fluctuating between 54° and 56°F in daytime ambient air temperatures of 83° to 93°F, for a + 30°F differential. This compares favorably with the systems requiring more extensive hardware and operation and with the fan-enhanced system, which requires an external power source. The increased performance of one of the other systems might be appropriate for severe climates or for specific architectural designs. Otherwise, the simple shaded Cool Pool is adequate.

Conclusions for the Six Preliminary Experiments

In general, the more north sky exposure a shade allows the cooler the water, but not more than 4°F whatever shade is used. Different shade configurations also affected the differences among maximum, late afternoon temperatures and among minimum, nighttime temperatures. For example, in Experiment #2 maximum temperatures were 2°F to 4°F apart and the minimum temperatures only 1°F to 2°F apart on the hottest days. With non-evaporating pools, the maximum separation occurring in the afternoon was 4°F. Since the system's performance depends on the minimum temperature achieved, the different shade configurations tested had little, important effect on this key variable.

The closed tanks in all four configurations were consistently 10°F warmer than the evaporating tanks. Varying a shade made a 2°F difference while adding a fan lowered water temperature in an evaporating tank by 5° to 8°F. Thus, the difference between a closed, non-evaporating tank and a fan assisted, evaporating tank could be 18°F. The dominant effect of evaporation is explained theoretically in the Appendix (pp. 73).

Comparing the Cool Pool to the Skytherm indicates that the Skytherm runs 2° to 3° cooler than the simple Cool Pool, while the Cool Pool with a fan to enhance evaporation is 1° to 2° cooler than the Skytherm model.

The larger 200 lb tanks used in the final series of experiments had day to night temperature fluctuations of 5°F at the pool's bottom as compared to the 10° to 15° differentials of the shallower tanks. The increased mass better retains the pool's cool temperature.

Shade Design Criteria

The preliminary experiments indicated that the best shade design had unimpeded north sky view, good ventilation, complete summer shading, a low architectural profile and low cost. Since the requirements for this best of all possible shades conflicted, the design chosen for the next series of experiments balanced these factors.

Summary of 1977 Work

The experiments clearly indicated that many alternative shade and pool configurations were workable. These early experiments were by no means exhaustive and other avenues of research would clearly be worth investigating.

1978 TEST BUILDING EXPERIMENTS

During the summer of 1978, a Cool Pool prototype was tested on a specially constructed test building designed to incorporate a Suncatcher and a Cool Pool as they might be integrated on an actual building. The experiments were conducted on a heavily insulated 144 sq. ft. test building near Winters, California. The 12'X 12'X 12' wood frame structure has R-19 wall, R-30 roof and R-5 insulation at the perimeter of its concrete slab floor. The 24 sq. ft. of south window is insulated with an R-4 shutter. The roof pond consisted of a 60 sq. ft. X 14 inch deep galvanized steel pan, located on the northern portion of the test building roof. The pond was covered by a 3/8" plywood shade having 45° fixed louvers which prevent direct radiation from striking the pool. The bottom of the pool was insulated from the test building by R-19 fiberglass batts.

Problems with the data acquisition system and unusual winter weather conditions made testing of the Suncatcher installed on the test building difficult. (A Fluke 2200B Data Logger was eventually obtained to replace the former troublesome data acquisition system.) As a result, only the Cool Pool system on the test building was evaluated. (Suncatcher data reported in the previous chapter came from actual residential buildings.)

The Cool Pool was tested in three ways: connected to four water-filled cylinders, connected to two water-filled cylinders and connected by means of a heat exchanger to the two cylinders. All of the cylinders were 8' in height x $1\frac{1}{2}$ ' in diameter.

For all three test modes the pool was plumbed into water-filled cylinders placed within the test building. A $1\frac{1}{2}$ " diameter 8' long ABS pipe connects the cooled water at the pool's bottom with the bottom of the water-filled cylinders. A $1\frac{1}{2}$ " plastic tube, serving as a riser, connects the top of the cylinder with the top of the pool. Water circulates through the system by thermosiphoning. (A detailed mathematical analysis of the thermosiphoning process is included on p. 72.)

Data was recorded for inside and outside air temperature and humidity, and for cylinder temperatures at three levels: pool top, pool bottom and slab. In addition, outside wind speed and wind speed over the pool's surface were measured as well as net radiation over the test building pool and over an unshaded pool.

Infiltration was estimated to be high during the day (one air change per hour) since the door was often opened. Since the door remained closed at night and the test building was tightly constructed, $1/2$ air change per hour was used for nighttime estimates. Both day and night loads were calculated using ASHRAE methods. Included were infiltration, internal loads and conduction through the building envelope. All experiments were run without nighttime ventilation so that all cooling would be provided by the Cool Pool.

Test Building Experiment #1

Object--Experiment #1 was designed to verify the Cool Pool's ability to cool a building interior by thermosiphoning cool water into four interior water-filled cylinders.

Results--The experiment lasted six days. For five of these days, outside temperatures peaked above 100°F (wet bulb temperatures peaked at 79°F). The following figure compares a typical day's outside temperature to indoor temperatures recorded at the 3 1/2 ft. level. Even after three days of 100°F plus weather, the test building's interior temperature remained between 68°F and 77°F, with humidity ranging between 59% and 79%. The maximum outside temperature during this experiment was 106°F (wet bulb 78°F).

According to ASHRAE standards, inside conditions were always within the comfort range for low air velocities, light clothing and light activity. The ASHRAE standards, applied to these test building experiments, were developed for buildings which radiate at or near the interior air temperature. Since the cylinders within the test building are at temperatures below that of ambient air, occupants will feel cooler and more comfortable than even the ASHRAE standards suggest.

The chart below compares the estimated building load and the hourly cooling in BTUs provided by the Cool Pool for Experiment #1. An estimated cooling load was calculated using Sol-Air temperatures based on actual outside temperatures and a specified interior temperature of 78°F. The actual cooling load was similarly calculated except that measured inside temperatures were substituted for the 78°F temperature. Actual and estimated cooling loads were calculated in this way for each of the test building experiments.

The hourly cooling load for the test building was calculated according to ASHRAE's Sol-Air temperature equation which accounts for radiation gains and losses as well as convection and conduction from the outside air. The internal heat gain from the data acquisition system was included in the hourly cooling load.

Since there was no nighttime ventilation of the test building, virtually all cooling can be attributed to the thermosiphon process. Despite unusually hot weather, the Cool Pool was able to provide more than 100% of the test building's required cooling.

Conclusion--The Cool Pool's thermosiphon system works well even under exceptionally demanding cooling conditions.

Object---Since the results of Experiment #1 suggested that the Cool Pool system may have been oversized, two of the four interior cylinders were blocked and the pool connected only to the remaining two cylinders.

Results—The following figure compares inside and outside conditions on a typical day. With only two columns, the inside temperature also remained within the ASHRAE comfort range. Inside temperatures were between 70°F and 75°F, while interior humidity ranged from 53% to 59%, even though outside temperatures reached 94°F (68°F wet bulb). These results indicate that the Cool Pool system using four cylinders was indeed oversized for cooling the test building.

The chart below shows the estimated load and hourly cooling provided by the Cool Pool for a typical day during Experiment #2. Since the outside temperatures were lower in this second experiment, less total cooling was required. The lower total hourly cooling shown for Experiment #2 (as compared to the previous experiment) does not indicate impaired performance, since more than 100% of the cooling load was again supplied by the Cool Pool.

Conclusion--The Cool Pool connected to only two of the interior cylinders was still oversized for cooling the test building. Obviously, even with only two cylinders thermosiphoning there was more than adequate circulation of cool water to remove the heat from the building's interior.

Test Building Experiment #3

Object--To circumvent the problem of contamination entering the cylinders, where it could cause corrosion and reduce heat transfer efficiencies, the exposed water of the Cool Pool needed to be isolated within the system by a heat exchanger.

The heat exchanger must present little or no frictional resistance to water flow, since thermosiphoning operates with very low pressure, less than 1 psi head. Such low pressure rules out the use of conventional piped heat exchangers since the pipe lengths and turns necessary for adequate heat exchange at typical temperature differences, about 4°F, would introduce enough friction to reduce thermosiphoning drastically.

Results--A six mil thick sheet of plastic, attached to the pool's sides and extending 1" below the water's surface, isolated the water circulated to the cylinders from the environment without adding to frictional pressure losses. The water above the plastic evaporates and cools the bulk of the isolated water below.

The following figure compares inside and outside temperatures for a typical day during this experiment with the heat exchanger. Despite outside temperatures up to 95°F, interior temperatures remained within the comfort range, between 68°F and 72°F at the 3 1/2' level with humidity ranging from 59% to 64%.

The chart below shows the estimated load and actual hourly cooling in BTUs. Outside conditions resembled those which occurred in Experiment #2, and again the Cool Pool provided more than 100% of the test building's required cooling.

The following figure compares the water temperature rise above wet bulb at the top of the pool for Experiments #2 and #3. The lower curve shows that the pool with the heat exchanger was cooler relative to wet bulb temperature than the pool without a heat exchanger. This is probably an effect of the storage of cooled water mass in the columns and the pool.

Conclusion--The heat exchanger does not impair Cool Pool performance although the heat exchanger's effect on a pool with reduced surface area was unknown.

Comparison of Shaded and Unshaded Pools

Additional experiments were also conducted on a small unshaded pool of the same depth as the test building pool. The following figure compares net radiation (BTUs/hr/ft²) to the shaded pool on the test building against the net radiation of the unshaded pool on August 15.

During the day the shaded pool received only one tenth the net radiation of the unshaded pool. At 1:00 PM the unshaded pool received 168 BTUs/hr/ft² more radiation. Radiation into the shaded pool is comprised of diffuse and reflected insolation, as well as radiation from the sun-warmed shades.

At night the warmed shade continues to radiate into the pool, causing a small positive net radiation gain (4 BTU/ft²) while the unshaded pool loses about -16 BTU/ft² to the night sky. After midnight as the shade cools, net radiation for the shaded pool is -5 BTU/ft² compared to the unshaded pool's -14 BTU/ft². The unshaded pool lost 158 BTU/ft² more than the shaded pool at night but gained an excess 1527 BTU/ft² during the day. The daytime heat gain of an unshaded pool offsets its slightly better performance at night.

Temperature Stratification in Cylinders

The following two figures show temperature stratification within a water cylinder at midnight and noon on August 15 and 29. Thermosiphoning provided significant daytime cooling, but since no consistent pattern characterized the difference between column temperatures and air temperatures, more testing and data is needed to study this aspect of the system's performance.

On August 29, when four columns were connected to a pond with a heat exchanger, the cylinder temperatures were significantly cooler than on August 15 when only two columns were connected to a pool without a heat exchanger. Outside dry bulb temperatures were similar, but on August 29 wet bulb temperatures were lower. When only two cylinders were connected, they operated at a higher temperature which increased thermosiphoning to compensate for the loss of the two blocked-off cylinders.

The Cool Pool system is very responsive to the natural forces which operate it. Higher column temperatures create increased thermosiphoning which enhances the system's cooling efficiency.

Cooling by Radiation, Convection and Evaporation

The following figure shows the results of calculating the shaded pool's performance on the test building for a typical August day in Winters, California.

The net radiation (mostly diffuse and reflected insolation) was the most important means of heating the pool during the day. Convection warmed the pool during the day about the same amount as it cooled it at night, for a net contribution close to zero. Evaporation was the largest cooling method: eight times more effective than night convection. These figures are not experimental; they are based on the analysis shown in the Math Models on page 70. The theoretical, estimated nighttime evaporative cooling was measured over an open pool of water.

equivalent to the nighttime radiative cooling measured over an open pool of water.

General Conclusions

The results of the experiments and analysis discussed above indicate that the Cool Pool in all three configurations provides effective, evaporative cooling without introducing additional water vapor into a building's interior.

About 42 cubic feet of water would be lost to evaporation during a typical cooling season for the 6'X 10'X 1' pool used in our experiments. This represents the amount of water an electrical generating plant's cooling tower would use in producing 630 KWH of electricity, approximately one month's electrical energy for an air conditioner.

Like the Suncatcher for heating, the Cool Pool is a simple, self-regulating passive cooling device suitable for both houses and commercial scale buildings. Together the Suncatcher and Cool Pool provide a powerful tool for passive designers.

1979 TEST BUILDING EXPERIMENTS

The 1979 experimental test set up was basically the same as the previous year's set up. Differences are noted. The original troublesome data acquisition system was replaced with a Fluke model 2200B Data Logger. The Cool Pool module building was monitored to verify the accuracy of the heat transfer relationships and the computer program that were developed from 1978 experimental and research data.

Temperatures were measured with 30 gauge Omega-Constantan thermocouple wires which were welded together and coated with epoxy to guarantee accurate readings under water. Since temperature changes were slow, sensor response time was not an important factor. The thermocouples were constructed from a single spool of wire and calibrated with an HP digital voltmeter in a Rosemount controlled ice bath and hot oil bath. The thermocouples showed uniform readings (± 0.001 mV at most). They agreed with the platinum resistance thermometer to well within 0.05°C at 10°C and 20°C . They were uniformly biased at 30°C by $+0.1^{\circ}\text{C}$ and at 40°C by $+0.2^{\circ}\text{C}$.

The thermosiphoning rate was measured by injecting dye into a rubber sleeve which connected a 4 foot (1.22 m) long, 1 1/2" (3.8 cm) I.D. clear tube to the riser outlet. The Boussinesq relation gives:

$$\text{length for fully developed flow} = .03 \text{ Re} \cdot D$$

where: Re = Reynolds number
 D = pipe diameter

For the velocities measured, this length was between 2 and 3 feet. Therefore the injected dye was timed between the 3 ft. and 4 ft. marks on the clear tube. Flow was always laminar so the centerline velocity was timed and divided by 2 to yield the average velocity.

Net radiation into the roof pond was monitored with a one year old Weather Measure R422 Net Radiometer. The factory calibration constant was used. Outside wet bulb temperature was monitored with an aspirated, platinum resistance wet bulb thermometer and double-checked with a recording hygrothermograph. Sensor readings were processed and recorded with a Fluke 22003 data logger. Temperatures were measured at 5 heights within the water column, 3 heights within the downcomer, 8 heights in the interior air. A thermocouple was taped to each wall, two places on the roof, and on the floor. Measurements were not taken until the system was set up for at least three days.

Thermocouple Placement

Before the Cool Pool computer model was developed the various heat and mass transfer processes were studied individually. The main heat and mass transfer paths for the Cool Pool system are shown here.

Heat and Mass Transfer

Heat is transferred into the water column by convection from the air and radiation from the walls, people, furniture, windows, etc. In this system the windows were shuttered and no insolation fell on the columns.

Water is transported from the pool, through the downcomer and column and out into the pool via the riser. Heat is conducted and convected into the downcomer from the column water.

The pool water is primarily cooled by evaporation. Depending on outside conditions, it is either cooled or heated by convection, radiation and conduction.

Cool Pool sizing experiments and the resulting computer program are presented in Appendix II on p. 93.

1979 Test Building Set Up

SECTION OF TEST BUILDING

1980 TEST BUILDING EXPERIMENTS

The same module was used as in previous experiments. The entire shade, insulated parapet, and pool were removed from the module. A finished roof was constructed as a pool base. This was to eliminate awkward construction and rain leakage problems. An additional shed roof was constructed to the north to provide a work platform.

The original Cool Pool tank was sheet metal, fabricated at a heating and air conditioning shop. Recently, we drew up plans for two sheet metal pools, one with an integral sheet metal heat exchanger, and sent them out for bids. The bids came in so high (\$17.72 per sq. ft. and \$18.90 per sq. ft. of evaporating surface) that we decided to try alternative tank designs to bring the cost within reason.

One pool design we are considering is a cross linked polyethylene tank designed for livestock feed. Unfortunately, the tank is not exactly the right dimensions for our uses. The sides are somewhat too high - 23" versus 16", and the tank area is limited to 4' x 4' or 4' x 8' sizes. Nevertheless, the relatively low base price of \$6.24 per sq. ft. of pool area means that even with shipping and installation of a heat exchanger and plumbing, this cleanly finished tank could come in at around \$10 per sq. ft.

We have built two wooden pools that are lined with waterproof material. These pools are of standard house stud frame construction, and are designed to be built in place on flat roofs at construction sites.

One of the pools is lined with a resin-fiberglass mixture that is used in the manufacture and repair of fiberglass liquid handling tanks and is used to line pools. A coat of gel-coat over the fiberglass ensures a good resin cure and seals any fiber ends that might stick out. Prices of application vary depending on complexity, size of job, etc., but we are looking at an on-site application cost of perhaps \$3 - \$5 per sq. ft. of pond area on a commercial size Cool Pool installation. The work is performed by professional contractors who specialize in fiberglass resin application of this sort. The other pool is lined with 30 mil fiber reinforced hypalon.

The steel culvert water columns were modified and moved. 10 ga. steel plates were cut to fit the culvert tops. Two holes were cut in the plates and a 1 1/2" and a 2 1/2" threaded pipe flange were welded into the holes in each plate. The plates were then spot welded to the tops of the culverts. Gaps were filled with a mixture of epoxy and glass fiber. Before this set up, the entire seam was taped with fiberglass cloth tape, and epoxy was pressed into it with a putty knife. After the epoxy set, rough spots were filed and a polyester gel coat was applied. One of the two former pipe connections on the side of the culvert was plugged. The other (bottom) one received a reducer and a hose bibb valve to facilitate system drain down.

The culverts were placed directly on the concrete slab floor. Subsequent tests demonstrated that coupling to the slab did not offer cooling to the water columns, as the water temperatures at the bottom of the columns were consistently cooler than the slab temperatures. The water column taps were placed at a lower level than the evaporating pool tops (approx. 1 1/2' lower).

The culverts were placed at the south wall in the east and west corners of the building. The connecting pipes between the culverts and pools were about six feet long. The downcomer connecting pipes were insulated with about R-3 insulation-wrapped fiberglass. It appears that the R-3 insulation was not enough, as thermosiphoning stopped at one point during an experiment--apparently because of overheating of the downcomer connector. In the future, it is recommended to use a much shorter connector (vertical columns adjacent to pools) or more substantial insulation.

1980 Test Building Set Up

DETAILING

Biological Organisms and Debris

Mosquitos, algae, and debris problems may develop in any open water. Mosquitos must be completely controlled as they are not only a nuisance but present a health hazard. Algae and debris are problems only if they obstruct the plumbing or aid in corrosion of the pool container.

Three main controls for these problems have been investigated: biological, chemical and barrier.

The biological control method was tried in the 6'x10'x1' deep shaded Cool Pool located on the roof of the test building. The chemical and barrier experiments were conducted in 14" diameter 4" deep galvanized steel pans located under a plywood shade.

Biological Set Up

Biological

Mosquito fish, *Gambusia Affinis*, eat mosquito larvae. They thrive in water which is between 40°F and 100°F and can even survive the winter under a layer of ice. These fish grow to be about 2 inches long. They bear live young about three times per summer.

Mosquito fish were introduced into the Cool Pool test building water on June 11, 1979. As of June 30, 1979 they were alive and no mosquitos had developed although mosquitos are present in this area.

Algae was growing in a thin layer on the Cool Pool bottom. After 2 years of operation no corrosion of the metal pan was visible under the algae and the algae did not inhibit the flow through the pipes. It does not seem to present any problem.

Chemical

Two chemical alternatives were investigated including copper Sulphate and Ph control.

Copper Sulphate (also known as blue stone) is an algaecide which, in the recommended dosage (1 to 1.5 ppm), is relatively harmless to most animals including humans and livestock. It is toxic in greater concentrations. Since it does not evaporate with the water one dose should last the season. However, when the winter rains cause runoff, an overflow to the sewer system or septic tank should be provided to avoid the build-up of copper sulphate in the soil. Copper Sulphate may control mosquitos by depriving the larvae of food. It tends to oxidize ferrous materials.

Agricultural Lime (CaCO_3) was used to induce an alkaline Ph of about 8.3. This appeared to be the maximum solubility achievable. It was hoped that this basic environment would inhibit algae growth and thus starve the mosquito larvae. Preliminary results indicated that the lime enhances corrosion.

The control pan achieved a Ph of 8.3 from evaporation and concentration of naturally occurring salts without corrosion.

Volatile chemicals such as chlorine were rejected because the pool water would need to be constantly tested and the chemical replenished. In addition, chlorine is an oxidant and corrodes steel.

Oil or kerosene is sometimes used to control mosquitos. However a coating of these substances on the water surface would decrease evaporation and decrease the pool's ability to reject heat.

Barrier

Ordinary polyester window screening was fastened over one test pan. The screen was located $1\frac{1}{2}$ inches above the water surface and the edges were carefully sealed to the pan. Although it is possible that the mosquitos may have laid their eggs through the screen, the new mosquitos should be trapped under the screen and die. During the 3 weeks the screen has been in place no larvae have been observed in the water.

Conclusion

The effectiveness of the different methods for mosquito control have been tested. It appears that lime aids corrosion and should not be used where it may come into contact with steel. Algae and debris do not seem to present a problem. They do not appear to enhance corrosion nor do they inhibit the thermosiphoning.

The Cool Pool Shade

The Cool Pool shade is a fixed shade. Numerous experiments and calculations have shown that:

1. An unshaded pool of water loses 158 BTU/ft² more than a shaded pool during the night, but gains an excess of 1527 BTU/ft² during the day.
2. Net radiation (diffuse and reflected) is the most significant means of heating the pool during the day except for the building load.
3. North sky and night sky radiation from the pool do not seem to be critical factors.
4. Evaporation from the pool is by far the most significant cause of net cooling.
5. The ideal shade, a vertical wall to the south of the pool, would have to be about 40 feet high to shade a 6' wide pool at noon. A 45° single wall shade would have to be about 5' high. These designs would not be easy to build, and would generally be architecturally unacceptable.

The problem then, is to design a shade that is low profile, blocks all direct solar radiation, and does not inhibit evaporation of the pool below.

A New Cool Pool Shade

A new shade prototype has been designed and constructed. It has been installed and its performance monitored. It's performance has been compared to the performance of the shade already installed on the test module.

The original module shade did not adequately block the direct insolation from the north from striking the pool in the morning and evening during the hottest part of the summer. At sunrise and sunset on June 21st at 38°N. lat., the sun respectively 30°N. of East and 30°N. of West at sunrise and sunset. A shade has been designed to shade the north of the pool from this sunlight. It consists of a frame holding vertical slats with 6" spacing between the slats (not 6" centers). The slats are each 3 1/2" to 4" wide.

This shade presents a minimal obstruction to air flow. It presents a facia which is more architecturally acceptable than previous attempts at a north shade.

When working with low profile flat angled louvers, there is always some time of day that light can pass between the louvers. One solution to direct radiation infiltration is the use of chevron louvered shades. These may also reduce air flow above the pond somewhat, but can span greater widths than 45° shades, and do not require auxiliary vertical shades at right angles to keep east and west sun out. Unfortunately, the cost of chevron louvers is too high to consider them a cost effective

No sunlight can pass directly through this shade.

component of a passive cooling system. Prices ranged from \$12 per square foot to \$22 per square foot not including shipping or installation.

The solution was to use standard attic vent louver panels horizontally. These louvers cost only about \$3.75. Since these 45° louvers only face one direction (south in this case), East and West morning and afternoon sun coming from the North can pass between the louvers to a certain extent. The solution was to hang galvanized sheet steel shades between the rows of louvers. The louver supports each are made of two pieces of 1 x 4 redwood. Each of the sheet metal shades is sandwiched between the two 1 x 4's, so that it hangs down from them. The sheet metal shades are to be painted flat black. The louver tops are to be painted glossy white.

The shade support is constructed entirely of rot resistant wood. It consists of two ledgers, one at the North and one at the South of the pool area. The ledgers are fixed in place. The 1 x 4 pairs with the sheet metal sandwiched between them are set on and perpendicular to the ledgers; these louver supports are not fastened in place. They are placed on 16" centers to accept the louver panels. The whole setup is modular (16" centers) and goes together or comes apart very rapidly for maintenance purposes. It takes about 5-10 minutes to completely remove or reinstall all louvers, vertical shades and supports on the square foot test module.

The shade was evaluated from data taken during two thermosiphoning events during late summer, 1980. There were several areas of concern over the new

Vertical Shades to Block East/West Sun

shade. First, the profile was lowered resulting in no direct air flow path under the shade. Second, we didn't have time to paint any of the sheet metal. For the time being, the lack of white paint on the 45° louvers didn't pose a problem, because the new galvanized finish was extremely shiny and bright. However, the vertical shades under the louvers would reflect some light into the water instead of blocking it. Third, the louvers came with a 1/8" mesh screen which might have slowed air flow.

Overall, however, the shade system appeared to perform as well as the previous design. Furthermore, the details of construction in a modularized form have been worked out in sufficient detail to allow design and specification for contractors to work by. It appears that possible reduction in cooling performance due to reduction in air flow caused by lowering the shade has been compensated for by improvement in total shading of the pool.

Air Flow

Development of a Cool Pool Heat Exchanger

Despite the fact that algae and debris did not cause corrosion or stop thermosiphoning during the 1978-9 testing, there was concern over long term build up. Also, it is desirable to keep the water column fluid separate from open air to prevent corrosion in the steel water column tubes.

Various heat exchangers to separate evaporating and thermosiphoning water were studied. A discussion of the pros and cons of each method follows:

The thermosiphoning Cool Pool operates on a pressure difference of as little as 0.05 psi. This rules out the use of conventional piped heat exchangers since they introduce substantial frictional resistance and would stop the thermosiphoning. The designs shown present negligible frictional resistance to the water flow.

It is important that the surface of the pool be at an equal or higher level than the top of the water cylinder so that the thermosiphoning will work properly. This presented a design problem for the rigid sheet metal heat exchanger. The solution is shown in the adjacent figure.

The sheet metal exchanger (A) is tilted slightly from horizontal so that air will not be trapped under the metal and obstruct the heat transfer from the isolated lower water to the top, evaporating water. The highest point of the system (B) contains a float valve with make up water to insure the water level

Metal Heat Exchanger

remains high enough. If the water level were to drop so that there was an air gap between the lower water and the heat exchanger, cooling would be drastically reduced. Although the water under the heat exchanger is enclosed and should not need replenishing, experience has shown that a back-up system is wise. A second float valve located above the heat exchanger is necessary to replenish the evaporating water. An integrated sheet metal pool and heat exchanger was rejected because of high cost and difficulty of integration.

The adjacent figure shows the vinyl waterbag heat exchanger. Since vinyl has about the same density as water the water bag completely loses its structural shape when immersed in water. This presents a problem since the vinyl may drape over the connecting pipes and obstruct the thermosiphoning. Or it may float to the pool surface, become dry and reduce the water area available for evaporation. Therefore a restraining framework must be designed for the vinyl heat exchanger.

Since flexible vinyl film does not affect the pressure gradient within the roof pond only one float valve (located at the free surface) is necessary. Care must be taken that the water level in the interior column remains at or below the pool's free surface. This will not be a problem unless the cylinder is over-filled initially. The vinyl water bag heat exchanger was rejected because of difficulty in handling and special manufacture required.

The final heat exchanger design calls for a sheet of 30 mil fiber reinforced Hypalon^R membrane. The membrane is suspended horizontally, four inches below the top edge of the pool, allowing enough water depth for the mosquito fish above the membrane.

Vinyl Bag Heat Exchanger

Hypalon Heat Exchanger

Although only one manufacturer produces this material, the material is in widespread use as a commercial waste pond lining and cover. Consequently, it is not likely to go out of production. The Hypalon^R can be ordered in sheets made to size, and is then relatively easy to install on site.

A grid made of SCH 125 PVC sprinkler pipe on about 2 foot centers and filled with sand holds the membrane fairly flat in the water.

Plastic plumbing fittings made for hot tubs, called "through wall" or "inlet" and "outlet" fittings, are currently being used to detail plumbing into the pools. These fittings are neat and easy to install, using a nut for compression with rubber gaskets for the seal. They can be installed in tank walls up to about two inches thick.

Mosquito fish, *Gambusia Affinis*, have been grown in the evaporating pond above the heat exchanger to control mosquito larvae. Next cooling season we may also try fully screening the pond, although at present this does not appear to be cost effective, and could possibly impede evaporation air flow. In any event, the last cooling season ended before we had a chance to try the screen on the module shade.

Heat transfer through the hypalon membrane was evaluated as follows:

Thermal conductivity in Btu/h/sq ft/unit temp gradient

k = water	.343
rubber	.08
air	.0140
steel	26.2

The conductive and convective heat transfer from the room to the water columns was considered to be an air to steel to water transfer; in this case the limiting factor was the air $k = .0140$. The hypalon heat exchanger transfer was water to hypalon to water; in this case, the limiting factor was the hypalon $k = .08$. Even though there is less pond surface area than water column surface area (about 1/2), the hypalon heat transfer is about 5.7 times as fast as the air heat transfer. It was assumed that the room air velocities and the pool thermosiphoning velocities would be low, and that relatively stagnant films of air and water would be present at the respective membrane surfaces.

THIS PAGE
WAS INTENTIONALLY
LEFT BLANK

THIS PAGE
WAS INTENTIONALLY
LEFT BLANK

Appendix I

Math Models from 1978 Test Results

The three main heat transfer paths in the Cool Pool system are: building interior to water column, water column to roof pool (thermosiphon), and pool to atmosphere. This section shows analytic expressions for these heat transfer rates, discusses key variables, and presents some sizing rules of thumb.

It must be emphasized that the expressions for this system are coupled partial differential equations which are not amenable to analytic solution. The rules of thumb are presented for use until a more accurate computer solution becomes available.

Interior to Water Column

Heat is transferred from the building interior to the column by radiation and convection, through the column wall by conduction, and to the column water by convection. The convection and conduction terms can be combined into one resistance factor so that:

$$\text{(Eq. 1) } Q/A = \text{net radiation} + \text{convection and conduction}$$

The net radiation absorbed can be expressed as the amount of absorbed incident radiation less the radiation emitted from the water column.

The amount of incident radiation is determined by building geometry and temperatures. For a single column this can be approximated by:

(Eq. 2) Incident radiation absorbed = $\alpha_c \epsilon_b F_{c-b} \sigma T_b^4$

The symbols are defined on page . Radiation is used instead of radiosity throughout this treatment since absorptivities are close to 1 and this results in simplified equation.

The radiation emitted from the water column can be expressed as:

(Eq. 3) Emitted radiation = $\epsilon_c F_{c-b} \sigma T_c^4$

The preceding calculations establish that the net absorbed radiation can be approximated as:

(Eq. 4) $Q/A = \alpha_c \epsilon_b F_{c-b} T_b^4 - \epsilon_c F_{c-b} T_c^4$

Convection and conduction into the cylinder is given by the equation:

(Eq. 5) $Q/A = (T_a - T_w)/R$

The columns used in most Living Systems designs are made from corrugated culvert pipe. A first approximation for h_a could be calculated from the free convection vertical cylinder relationship:

(Eq. 5) $h_a = 0.19(T_a - T_c)^{1/3} [GrPr > 10^9]$

Notice that h_a is a function of column and air temperature.

There are few relationships available for h_w at the negligible water velocities encountered in the columns. A value of 45 BTU/°F ft² could be assumed.

The value of L/k for the cylinders used by Living Systems is 6×10^{-4} ft²-F-hr/BTU.

The resistance provided by h_a (air to column) is 2 orders of magnitude greater than the convective resistance from the column wall to the water and 4 orders of magnitude greater than the conductive resistance in the wall. Thus, these terms can be neglected and the convective and conductive heat transfer into the water columns can be approximated as:

(Eq. 6) $Q/A = 0.19 (T_a - T_w)^{4/3}$

The expressions for convective and radiative heat transfer can be combined to calculate total heat flowing into the column. However, neither the air nor the cylinder is at a constant temperature. Neglecting any circumferential differences, the temperatures must still be expressed as an unknown function of height. Therefore, these expressions must be integrated over the column height to yield a complete (but approximate) equation for the heat transfer into the water column:

(Eq. 6) $Q = \pi D \int [\alpha_c \epsilon_b F_{c-b} \sigma T_b^4 - \epsilon_c \sigma F_{c-b} T_c^4 + (T_a - T_w)/R] dh$

The temperature profiles are determined by the rate of thermosiphoning and the pool temperature. Since these profiles cannot be expressed as simple analytical expressions, these equations must be solved on a computer.

Column-to-Pool Heat Transfer

Interior heat is transferred to the roof pond by thermosiphoning. Warm water flows out of the columns and cool water flows into them. Net cooling is given by the expression:

$$(Eq. 7) \dot{Q}_t = \dot{m}_p (T_{out} - T_{in})$$

The mass flow rate ($\dot{m}$) is caused by a pressure differential between dense, cold downcomer water and less dense, warmer column water. This pressure differential causes water to flow. The velocity of the water can be determined by equating the pressure drop due to pipe friction (a function of velocity) with the pressure difference due to density variations (a function of temperature).

$$(Eq. 8) V^2/2g [(\ell/d + 25)f + K] \gamma_d = (\gamma_h)_d - (\gamma_h)_c$$

where 25 is the ℓ/d equivalent for a 90° elbow.

This equation neglects the friction generated by the flow in the column since this is several orders of magnitude smaller than the downcomer pipe friction.

Since the mass flow rate ($\dot{m}$) is a function of velocity Eq. 6 must be solved for velocity. In the case of laminar flow ($Re < 2300$) this yields a quadratic solution:

$$(Eq. 7) V = \sqrt{\left[\frac{(\ell/d + 25)16\nu}{K\ell} \right]^2 + \frac{2g}{K\gamma_d} [(\gamma_h)_d - (\gamma_h)_c]} - \frac{(\ell/d + 25)32\nu}{K\ell}$$

For turbulent flow the friction factor (f) is not a simple relation; as a result Eq. 6 must be solved for velocity by iteration. Once the velocity is solved the mass flow rate ($\dot{m}$) can be substituted into Eq. 5.

$$(Eq. 10) \dot{m} = Va$$

Notice that ingoing and outgoing water temperatures as well as the temperature profiles in the column and downcomer must be known to solve this equation. Again we find the equation is coupled and cannot be solved without a computer.

Pool to Ambient

The roof pond rejects heat via radiation, convection and evaporation.

Radiation:

Net radiation from the pool can be expressed as the radiation from the pool less the radiation from the shade and sky back to the pool. The inside of the shade was painted black so its absorbence is close to 1.

(Eq. 11) Net radiation =

$$\alpha_p T_p^4 - F_1 \alpha_p \epsilon_1 T_1^4 - F_2 \alpha_p \sigma T_2^4 - I$$

The view factors F_1 and F_2 depend on the geometry of the shade. For the test module $F_1 \approx 0.7$ and $F_2 \approx 0.3$.

T_1 can be assumed to be the Sol Air temperature. T_2 , the effective sky temperature, can be found by empirical correlations relating ambient air temperature and amount of cloud cover to the sky radiation.

The radiative cooling of the pool depends on pool, sky and shade temperatures, as well as shade geometry.

Convection:

Pool convection to the atmosphere can be approximated by the forced convection correlation for air over a horizontal isothermal surface.

$$(Eq. 12) h_a = 0.664 Re_L^{1/2} Pr^{1/3} \frac{k_a}{L_p}$$

$$(Laminar Re < 10^6)$$

or:

$$(Eq. 13) h_a = 0.036 Re_L^{4/5} Pr^{1/3} \frac{k_a}{L_p}$$

$$(Turbulent Re > 10^6)$$

The convection depends on pool and air temperatures and wind velocity under the shade.

We found that the shade consistently reduced wind velocity to an average of 1.7 ft/sec over the pool surface. This means that the Reynolds number is 62,000 and h is 0.4 BTU/°F-ft²-hr.

The convective cooling can be expressed as:

$$(Eq. 14) Q/A = h(T_{pT} - T_a)$$

During the night when the air temperature may drop below pool temperature the pool can reject heat by convection. When the air temperature is higher convection causes heat gains in the pool.

Evaporation:

Evaporation depends on wet bulb temperature, pool surface temperature and wind velocity over the pool surface

Experimental mass transfer data for evaporation from shaded pools is sparse. The usual method of calculating heat loss due to evaporation is to determine a mass transfer coefficient (G) analogous to the heat transfer coefficient. The mass transfer due to evaporation is then:

$$(Eq. 15) \dot{m}/A = G(W_s - W_\infty)$$

The heat flow can then be calculated by multiplying the amount of evaporating mass by the heat of vaporization per pound of water.

$$(Eq. 16) Q/A = i_{fg} G(W_s - W_\infty)$$

The mass transfer coefficient (G) can be found by equating the Chilton-Colburn j-factors.

$$(Eq. 17) \frac{G_{am}}{V \frac{P}{m} t} \left(\frac{v}{D_v} \right)^{2/3} = \frac{h}{\rho C V_m} \left(\frac{C_u}{k_a} \right)^{2/3}$$

Heat transfer analogies have been verified by many experiments but mass transfer analogies are not so well verified and may have up to 100% error associated with them.

Making the simplifying assumptions that $P_{am}/P_t = 1$ and assuming constant conductivity, density, viscosity, diffusivity, and heat of vaporization:

$$(Eq. 18) G \sim 4.65h$$

The heat transferred by evaporation from a unit area of pool surface can be approximated as:

$$(Eq. 19) Q/A = 4595h(W_x - W_\infty) \text{ BTU/hr/ft}^2$$

Total Heat from Pool:

The radiative, convective and mass transfer heat flows can be combined into one expression:

$$(Eq. 20) Q/A = 4595(W_s - W_\infty) + h(T_p - T_a) + \sigma \epsilon_p T_p^4 - F_{1p} \alpha_p \sigma \epsilon_1 T_1^4 - F_{2p} \alpha_p \sigma T_2^4$$

where all temperatures must be in absolute degrees and English units are used.

The heat rejected by the pool will be a function of pool temperature, air temperature, partial pressure of water vapor in air (determined from wet bulb temperatures or relative humidity), sky temperature and wind velocity. The pool temperature is determined by the amount of water present as well as the temperature of the incoming water. Thus, this aspect of heat transfer is also coupled to the rest of the system and cannot be solved without knowing what is happening in the water columns.

Rules of Thumb

Until adequate sizing information is available from computer simulations of the Cool Pool system, these rules of thumb are presented.

The three heat transfer modes discussed previously are uncoupled by assuming various temperatures and flows. It must be emphasized that these assumptions are only justified to the extent that they provide interim, approximate figures for designers.

Sizing Columns:

Assumptions:

- 1) Inside air temperatures (T_a) vary linearly from 72°F at the floor to 82°F at the 10' level.
- 2) Tube water temperatures (T_w) vary linearly from 67°F at the bottom to 77°F at the top.
- 3) Water leaves the column at 77°F and enters the column at 67°F.

Integrating Eq. 6 using the assumed temperature profiles reveals that 1 ft² of column surface is capable of absorbing 9 BTU/hr under the assumed conditions.

To choose the number of cylinders required to cool a certain building, the peak hourly cooling load must be calculated. The number of columns required is found from:

$$(Eq. 21) \# \text{ Tubes} = \frac{\text{cooling load}}{9\pi DH}$$

where: D = average tube diameter
H = tube height.

Experience at Living Systems shows that this rule gives good values for solar-tempered houses.

Sizing Pool Surface:

Assumptions:

- 1) $T_p = 72^\circ\text{F}$ (pool temperature)
- 2) Pool is 1' deep
- 3) Average wind speed under shade is 1.5 ft/sec
- 4) $h = 0.6 \text{ BTU/ft}^2\text{F/hr}$
- 5) Net daily radiation into the pool is assumed from experimental results to be 180 BTU/ft²/day.

$$(Eq. 22) Q = h(T_o - T_p) + \frac{1_{fg} h}{.897cp}(W - W_{sp})$$

which reduces to:

$$(Eq. 23) Q = 89.8 - 0.6T_o - 2757W \text{ (BTU/ft}^2\text{-hr)}$$

where:

$$W = 0.622 \frac{100}{RH} \left[1 + \left(\frac{0.622}{W_s} \right) \right] - 1$$

$$-1 \left(\frac{\text{lb}_m \text{ water}}{\text{lb}_m \text{ dry air}} \right)$$

RH = relative humidity outside

$$W_s = 0.622 P_{vs} / (P - P_{vs}) \text{ (lb}_m \text{ water/}$$

$$\text{lb}_m \text{ dry air)}$$

$$P = 29.921 \text{ (inches Hg)}$$

$$P_{vs} = \exp[-3.006 + 4.310 \times 10^{-2}T_a -$$

$$6.506 \times 10^{-5}T_a^2] \text{ (inches Hg for}$$

$$74.97 < T_a \leq 110^\circ\text{F)}$$

$$\text{or: } P_{vs} = \exp[-3.078 + 4.527 \times 10^{-2}T -$$

$$8.0101 \times 10^{-5}T_a^2] \text{ (inches Hg for}$$

$$74.97 < T_a \leq 110^\circ\text{F)}$$

To size the pool, average hour by hour outside temperature and relative humidity for July or August should be used in Eq. 23. The 24 hours of heat transferred from the pool should be added up to equal Q_{daily} . Then the daily cooling load should be calculated:

$$\text{(Eq. 25) \# sq ft pool required} = 1.5 \frac{\text{daily cooling load}}{Q_{\text{daily}}}$$

A safety factor of 1.5 has been included since the sizing formulas do not consider peak loading conditions.

Neither of these sizing methodologies considers the effect of thermal mass and time lag between cooling load and cooling delivered.

DEFINITION OF TERMS

A = surface area (ft ²)	g = gravity acceleration (32.2 ft/sec ²)
a = cross-sectional area of downcomer (ft ²)	Gr = Grashoff number (a function of temperature)
C = specific heat of air (BTU/°F-lb _m)	H = column height (ft)
C _p = specific heat of water (1 BTU/°F-lb _m)	h _a = heat transfer coefficient from air to column wall (BTU/°F-ft ² -hr)
d = pipe diameter	h _w = heat transfer coefficient from column wall to water (BTU/°F-ft ² -hr)
D = mean water column diameter (ft)	I = diffuse and reflected insulation incident on pool
D _v = diffusivity of water in air. $D_v \sim .000146 T^{2.5} / (T + 441) \frac{\text{ft}^2}{\text{hr}}$	K = friction factor for expansions
dh = incremental column height	T _c = column surface temperature
F ₁ = view factor from shade to pool	α _c = column absorptency estimated at 0.93 for a dark color
F ₂ = view factor from sky to pool (small since shade obscures much of the skyview)	α _p = absorptency of pool
F _{c-b} = view factor from the building to the column. For a single column this can be approximated as 1. For two columns it is about 0.86 and for a middle column it could be taken as 0.72	μ = viscosity of air
f = friction factor which is velocity dependent for laminar flow. $f = 64/R_e$	ν = kinematic viscosity of water
G = mass transfer coefficient (sec/ft)	ρ = density of air
	ρ _∞ = partial mass density of water vapor in distant air

THIS PAGE
WAS INTENTIONALLY
LEFT BLANK

THIS PAGE
WAS INTENTIONALLY
LEFT BLANK

Appendix II

Math Models from 1979 Test Results

On May 24, and May 25, 1979 the first thermosiphoning test was conducted. The thermosiphoning rate was measured by injecting vegetable dye through a rubber sleeve connecting a 4 ft. long 1½" ID transparent acrylic tube to the riser outlet. The length of time that the fastest central streamline took to traverse the 4' tube was recorded. The flow was obviously laminar. Therefore the average velocity was determined by dividing the centerline velocity in half.

Average measured velocities were compared to theoretical velocities calculated from the relationship:

$$\rho \frac{v^2}{2} \left(f \frac{L}{d} + K \right) = \Sigma \gamma h_d - \Sigma \gamma h_c$$

where ρ = density of water

v = average velocity

f = friction factor = $64/Re$

Re = Reynolds number

L/d = length + diameter ratio ~ 120

K = expansion factor (4.5 for 5 expansions)

$\Sigma \gamma h_d$ = height $\times$ density for downcomer

$\Sigma \gamma h_c$ = height $\times$ density for column (from measured temperatures)

Table I shows the experimental velocities and the calculated velocities. The maximum discrepancy was 12.6% which shows excellent agreement considering the error analysis indicates the maximum possible error in calculated velocity resulting from cumulative measurement errors could reach 20%.

TABLE I

READING	UGM	V _{CALC}	V _{EXP}	DIFF	RATIO
1	0.091	0.0598	0.0535	11.8	0.895
2	0.067	0.0460	0.0472	-2.6	1.026
3	0.074	0.0519	0.0471	10.2	0.908
4	0.098	0.0639	0.0654	-2.2	1.023
5	0.109	0.0699	0.0702	-0.4	1.004
6	0.110	0.0710	0.0673	5.5	0.948
7	0.109	0.0702	0.0678	3.5	0.966
8	0.105	0.0684	0.0601	13.7	0.879
9	0.091	0.0608	0.0570	6.7	0.937
10	0.075	0.0514	0.0522	-1.6	1.016
11	0.077	0.0529	0.0487	8.6	0.921
12	0.085	0.0572	0.0585	-2.2	1.022
13	0.095	0.0627	0.0625	0.4	0.996
14	0.100	0.0650	0.0637	2.1	0.980
15	0.102	0.0659	0.0645	2.2	0.978
16	0.100	0.0648	0.0631	2.7	0.974
17	0.097	0.0630	0.0625	0.8	0.992
18	0.100	0.0639	0.0617	3.5	0.966
19	0.097	0.0622	0.0617	0.8	0.993
20	0.090	0.0582	0.0588	-0.9	1.010
21	0.086	0.0560	0.0560	0.1	0.999
22	0.078	0.0512	0.0536	-4.5	1.047
23	0.072	0.0480	0.0539	-11.0	1.124
24	0.065	0.0438	0.0447	-2.0	1.021
25	0.065	0.0440	0.0449	-1.9	1.019
26	0.063	0.0431	0.0484	-11.0	1.123
27	0.070	0.0475	0.0455	4.5	0.957
28	0.071	0.0482	0.0482	0.1	0.999
29	0.070	0.0475	0.0479	-0.7	1.007
30	0.074	0.0467	0.0521	-10.4	1.116
31	0.071	0.0449	0.0534	-15.9	1.189

Thermosiphoning

Thermosiphoning hot water systems have been studied by Baughn, Dougherty and Crowther as well as others.

The pressure difference caused by the density variation between two columns of water can be related to the flow velocity by the following equation:

$$(eq. 2) \quad \int \gamma_{dh} \text{cold} - \int \gamma_{dh} \text{warm} = \gamma \frac{v^2}{2g} \left(\bar{f} \frac{L}{d} + K \right)$$

where

$\int \gamma_{dh} \text{cold}$ = the water density times differential height integrated over the height of the column of cooler water

$\int \gamma_{dh} \text{warm}$ = the density-height product integrated over the height of the column of warm water

γ = water density

h = height

v = velocity

g = gravity

f = friction factor for pipe losses

L/d = length to diameter ratio for pipe

K = expansion factor used in figuring pipe frictional losses

The right side of this equation is the conventional expression for the pressure drop due to flow within a pipe.

For laminar flow the friction factor, f , can be found from the following relationship:

$$(eq. 3) \quad f = 64/Re$$

where Re is the Reynolds' number based on the pipe diameter.

Flow through the 1 1/2" (3.8 cm) diameter column was found to create very little friction compared to the flow through the downcomer and riser. The L/d ratio was 120 and the expansion factor, K , was 4.0. A second order curve fit relates the viscosity and density of water to its temperature.

$$(eq. 4) \quad \gamma(\#/ft^3) = 62.3302 + 6.706 \times 10^{-3}T(^{\circ}F) - 1.016 \times 10^{-4}T^2 \quad [60^{\circ}F < T < 79^{\circ}F]$$

$$(eq. 5) \quad v(ft^2/sec) = 2.7897 \times 10^{-5} - 3.491 \times 10^{-7}T(^{\circ}F) + 1.458 \times 10^{-9}T^2 \quad [60^{\circ}F < T < 80^{\circ}F]$$

The expected flow rate was calculated from eq. 9 and the measured column, downcomer and pool temperatures. The curve fit relation was used for viscosity and density. This flow rate was compared to that actually measured. Table I shows that the calculated velocity agrees very well with the measured velocity.

The thermosiphoning mass flow rate, and incoming and outgoing water temperatures were measured to calculate the heat leaving:

$$(eq. 6) \quad \dot{Q} = \dot{m}c_p(T_{out} - T_{in})$$

Equations 2 and 6 were used to calculate the thermosiphoning rate in the computer model.

Radiation into the Water Column

Net radiation into the water column can be calculated from the set of equations:

$$(eq. 7) \quad \sum_{j=1}^n \left(\frac{\delta_{Kj}}{\epsilon_j} - F_{K-j} \frac{1-\epsilon_j}{\epsilon_j} \right) \frac{Q_j}{A_j} = \sum_{j=1}^n (\delta_{Kj} = F_{K-j}) \sigma T_j^4$$

where F_{i-j} = view factor from surface i to surface j

A_i = area of surface i

ϵ_i = emmissivity of surface i

Q_i = net heat flux from surface i

n = number of surfaces

If the surface absorbtivity is equal to its emmissivity these equations give the exact radiative heat transfer.

Difficulties creep in when these equations are applied to a real enclosure. View factors are often difficult to determine. Emmissivities of wall paints at about 78°F (25°C) have not been exactly determined. Dark paint can range between .85 and .96. White paint ranges between .80 and .92.

ASHRAE suggests approximating wall temperatures at the air temperature so that radiative transfer into the water column can be expressed as:

$$(eq. 8) \quad \dot{Q} = \alpha A \sigma (\bar{T}_a - \bar{T}_c)$$

where α = column absorbtivity

A = column surface area

σ = Stefan-Boltzmann constant

$\bar{T}$ = average air temperature

$\bar{T}_c$ = column temperature

This works well for some conditions. However, wall capacitance, wall thermal conductance, infiltration, insolation, the outside temperature history, and the presence of thermal storage mass can all effect the relationship between wall temperature and air temperature. Since radiative transfer is dependent on the fourth power of temperature the use of an average temperature for its calculation will not always yield an accurate answer.

The following figure shows this same building at 3:00 p.m. in the afternoon. The wall temperatures are very different from the average air temperature. The radiative heat transfer to the column calculated from the average air temperature will be greater than the enclosure analysis indicates.

Afternoon Temperatures

The accuracy of using air temperature as an approximation for radiation was tested by comparing the transfer rate to that obtained by using an enclosure analysis.

Since the column temperature varied by less than 3°F (2°C) it was modeled at one average temperature. Each of the 4 walls were assumed to be at a constant temperature equal to that measured 4 ft. from the floor. The ceiling was divided into two isothermal areas, one exposed to the sky and one under the roof pond. The floor was modeled at one temperature. Since, presumably, the surfaces were not isothermal some error will be introduced by these assumptions.

Here are shown hour-by-hour calculations of radiative heat transfer. The wall and column emissivities were .91 and .93.

Radiation: Enclosure and Air Temperature Calculations

This graph shows that using the average air temperature for radiation calculation overestimates the heat transfer into the thermal mass for this particular building. This error is less during the night when wall temperatures are more uniform.

Heat is transferred from the indoor air to the water column by convection, through the corrugated metal column wall by conduction and to the water by convection. The free convection heat transfer coefficient from air to a vertical cylinder can be found from:

$$(eq. 9) \quad Nu = 0.13 (GrPr)^{.33} \quad (10^8 < GrPr < 10^{12})$$

Since the water column is 8 feet (2.4 m) tall, $GrPr \ 10^9$ and the free convection is in the turbulent region. The corrugations may increase the heat transfer slightly. This relation has been simplified for air at 70°F (21°C):

$$(eq. 10) \quad h_a = 0.19 (T_a - T_c)^{.33} \\ \text{Btu/ft}^2\text{-hr-}^\circ\text{F} \ (GrPr \ 10^9)$$

where:

h_a = Heat transfer coefficient (Btu/ft²-hr-°F)

T_a = air temperature

T_c = column temperature

Equation 9 is often used in place of equation 8. Notice there is no dependence on the surface geometry in equation 9. Since conditions are evaluated at 70°F, equation 9 yields a slightly lower heat transfer coefficient than equation 8.

Since the water column is within an enclosure, convective air loops may be set up between the cool column and the warmer walls. This will increase the velocity of the air moving past the column and thus increase the convective heat transfer rate. Thus, equation 9 probably underestimates convective heat transfer. This hypothesis will be discussed further.

The thermal resistance per unit area between the air and the column surface is the inverse of the heat transfer coefficient:

$$(eq. 11) \quad R_a = 1/h_a$$

where R_a is the thermal resistance between the interior air and the column surface. Since the temperature difference is usually less than 10°F (6°C) this resistance is about:

$$(eq. 12) \quad R_a \sim 2.4 \text{ (ft}^2\text{·hr·}^\circ\text{F)/BTU}$$

Conduction through the .064" (.16 cm) thick metal column skin is 4 orders of magnitude faster than convection from the air to the surface of the column. The conductivity of steel is 26.2 BTU/°F ft hr so that the resistance:

$$(eq. 13) \quad R_c \ L/K = 2.0 \times 10^{-4} \text{ (ft}^2\text{·hr·}^\circ\text{F)/BTU}$$

where R_c is the thermal resistance of the column wall.

The heat transfer coefficient from the column skin to the water can be found from the free convection relationship of equation 9. At a 10° difference between column and water temperature and a water temperature of 73°F (23°C) the resistance, R_2 is 1.9×10^{-4} ($\text{ft}^2 \cdot \text{hr} \cdot ^\circ\text{F}$)/BTU. Again, this resistance is four orders of magnitude less than the air to column surface resistance.

Because the air to column resistance dominates convective heat transfer into the water column, conduction and convection to the water can be neglected.

The water column was placed on $1/2''$ thick plywood on hollow cinder blocks. Thus, conduction into the base of the column is negligible. Heat transfer into the cylinder top was neglected since the 1 inch air gap between the plastic cap and the water surface was assumed to act as an insulating space.

If a heat balance is done on the water column it is found that the heat entering the column from radiation and convection must equal the change in the column heat content plus the heat leaving by thermosiphoning.

The change in heat content was determined from the hourly change in temperatures of the column.

$$\text{(eq. 14)} \quad \Delta Q = m_c c_p (\bar{T}_2 - \bar{T}_1)$$

where: m_c = mass of water in column

c_p = specific heat of water

$\bar{T}_2$ = new average water temperature

$\bar{T}_1$ = old average water temperature.

From this, the hourly heat transfer rate into the water column can be calculated.

Calculated heat transfer rates (based on measured air, wall and column temperatures) and the experimentally determined heat transfer rates are presented in graph form. The lowest curve results from adding the convective transfer ($h - .19 (\Delta T)^{1/3}$) to the enclosure analysis radiation. The other curve results from adding this convective transfer to radiation calculated from the average air temperature. The solid line represents the experimental heat transfer rate.

Total Heat Into Column

The accurate enclosure analysis radiation added to the conventional convective heat transfer underestimates the total heat transfer. When the (incorrect) air temperature radiation is added to the conventional convective transfer, the calculated result agrees very well with the experiment. This is because radiation is overestimated to compensate for the underestimation of convection. Thus, this method gives a correct answer for the wrong reasons.

This discrepancy is greatest when the wall temperature is much higher than the mass temperature. This lends support to the hypothesis that convection loops within the enclosure are increasing the heat transfer rate.

This analysis implies that the heat transfer coefficient is consistently higher than that predicted by equation 4. However, this is only an indication that more work should be done on this question since there are many uncertainties involved in this analysis. The water column contains so much mass that a 0.1°F change in temperature can mean a change of 86 BTU's. Since temperatures were read to the nearest 0.1°F and the total hourly change in column energy content averaged 130 BTU/hr this introduces a possible 66% error. The thermosiphoning rate was much higher (300 - 600 BTU/hr) so that the impact of this error on the calculated convection rate is about 20%. The fact that the trend is consistent argues that the actual error was less than this.

Calculated enclosure radiation could add additional error to the convection. Emissivities were unverified and the view factors were approximate. Modeling the columns and enclosure as 10 isothermal surfaces introduces an unknown error. In any case, it is clear that using air temperature instead of surface temperatures for radiation calculations will overestimate radiative transfer in this situation. The computer model uses average air temperature for the calculation of radiation and convection into the water column.

Evaporation

The Reynolds analogy for heat transfer can be extended to convective mass transfer:

$$(eq. 14) \quad \frac{Sh}{Re \cdot Sc} = \frac{Nu}{Re \cdot Pr} = \frac{f}{2}$$

where: Sh = Sherwood number

Sc = Scranton number

Re = Reynolds number

Pr = Prandtl number

Nu = Nusselt number

f = friction factor

This is adequate for $Pr \approx 1$, and small temperature gradients. Chilton and Colburn modified the above relationship:

$$(eq. 15) \quad \frac{Sh}{Re} (Sc)^{-1/3} = \frac{Nu}{Re} (Pr)^{-1/3} = f/2$$

For water vapor in air this reduces to:

$$(eq. 16) \quad \frac{h}{g} = 0.21$$

where h = convective heat transfer coefficient

g = mass transfer coefficient

Although the heat transfer correlation to friction is not accurate for Reynolds numbers less than 10,000 it is felt that equation 14 will give a good estimate at lower Reynolds numbers if the heat transfer coefficient can be determined from some other relationship. Equation 16 is used in the computer simulation. The energy transfer rate from evaporation is

$$(eq. 17) \quad Q = g i_{fg} (w_o - w_p)$$

where: g = mass transfer coefficient

i_{fg} = heat of vaporization of water

w_o = outside humidity ratio

w_p = humidity ratio of saturated air at pool temperature

A curve fit for i_{fg} was used.

$$(eq. 18) \quad i_{fg} = 1091.40 - .495T - 5.00 \times 10^{-4} T^2 \text{ BTU/lb}_m \\ [60^\circ\text{F} < T < 80^\circ\text{F}]$$

The humidity ratio (w) was found from the relationships:

$$(eq. 19) \quad P_{vs} = \exp[-3.078 + 4.527 \times 10^{-2} T \\ - 8.0101 \times 10^{-5} T^2]$$

where P_{vs} is the saturated vapor pressure in inches of H_g at $T(^{\circ}\text{F})$ and $35^{\circ}\text{F} \leq T \leq 74.97^{\circ}\text{F}$

$$(eq. 20) \quad P_{vs} = \exp [-3.0006 + 4.310 \times 10^{-2} T - 6.5016 \times 10^{-5} T^2] \quad 74.97 < T \leq 110^\circ F$$

$$(eq. 21) \quad w_s = .622 P_{vs} / (P - P_{vs}) \text{ lb}_m / \text{cu. ft dry air}$$

where P = total pressure

$$(eq. 22) \quad w = .622 \left[\left(\frac{100}{RH} \right) \left\{ 1 + \frac{.622}{w_s} - 1 \right\} \right]^{-1} \text{ lb}_m / \text{cu. ft}$$

where RH = Relative Humidity.

Equation 21 was used to find the saturated humidity ratio over the pool. Equation 22 could be used to calculate the outside humidity ratio from the relative humidity. In this program the outside humidity ratio is calculated from the wet and dry bulb temperatures and used as an input.

Convection into the Roof Pond

The roof pond can be modeled as an isothermal flat plate. It is difficult to decide whether the convection is free or forced since wind speed varies and the shades may slow the air movement over the pool. It is clear that the heat transfer will be equal to or greater than that which the free convection relationship yields.

During the day when the pool is warmer than the outside air the following relationship would apply for turbulent free convection heat transfer.

$$(eq. 24) \quad h = 0.19 (\Delta T)^{0.33}$$

where: h = the heat transfer coefficient
BTU/°F-ft²-hr

ΔT = the difference between air temperature and pool temperature

During the night when air temperatures drop below the pool temperatures the following turbulent free convection relation would apply:

$$(eq. 25) \quad h = 0.22 (\Delta T)^{0.33}$$

The average summer wind speed in the Sacramento area is 12.5 ft/sec (3.8 m/s). Measurements taken by previous researchers indicate that the shade used on the test building roof pond reduces the wind velocity over the water to an average of about 2 ft/sec.

The forced convection correlation for air over a horizontal isothermal surface is: (11)

$$(eq. 26) \quad h_a = 0.664 Re_L^{1/2} Pr^{1/3} \frac{k_a}{L_p}$$

(Laminar $Re < 10^6$)

or:

$$(eq. 27) \quad h_a = 0.036 Re_L^{4/5} Pr^{1/3} \frac{k_a}{L}$$

(Turbulent $Re > 10^6$)

where k = thermal conductivity of air

L = plate length

If the laminar equation is evaluated for a 68°F (20°C) and 80°F (27°C) air this yields

$$(eq. 28) \quad h = .69 \sqrt{v/L} = .28 \sqrt{v} \frac{BTU}{^\circ F \text{ hr ft}}$$

$[v < 28 \text{ ft/sec}]$

where v is the velocity in feet per second. For a velocity of 2 ft/sec, $h = .4 \text{ BTU}/^\circ F \text{ hr ft}^2$. If the outside temperature is more than 9°F (5°C) different from the pool temperature the free convection will be greater than the forced convection. In this case the computer program chooses the greater of the two heat transfer coefficients.

Conduction

Into Pool:

The pool edges are insulated with R19 fiberglass batts and will gain (or lose) heat from the outside air.

$$(eq. 29) \quad \dot{Q} = (1/R) A (T_a - T_b)$$

where $R = 20 (^\circ F \cdot \text{hr} \cdot \text{ft}^2)/\text{BTU}$

$A = 32 \text{ ft}^2$ area

T_a = outside temperature

T_b = pool temperature

The pool bottom is insulated from the living space by R-19 fiberglass batts and will transfer heat according to the above relation. In this case: the area is 60 ft² (2.2m²) and T represents the inside temperature.

The top edges of the sheet metal pool container extend into the air and may act as fins by conducting heat into the pool. This effect was neglected.

Into Downcomer:

Since the ABS downcomer is immersed within the column heat will conduct across the plastic to heat it according to the equation:

$$(eq. 30) \quad \dot{Q} = \frac{1}{R} A (T_c - T_d)$$

where R = conductive and convective resistance of the ABS plastic and water

T_c = column water temperature

T_d = downcomer water temperature

A = surface area of downcomer

This relation was used in the computer simulation. It was assumed that convection loops set up within the column between the warm column and the cooler downcomer would result in the conducted "coolth" from the downcomer falling down to the lowest column node. Thus, the conduction was calculated node by node but the total heat was removed only from the bottom column node.

Net Radiation into the Pool

Although the shades block direct sun from the pool, diffuse radiation enters it during the day. The shades heat up and radiate into the pool. At night the pool can lose heat by radiation to the cooled shades and the small amount of visible sky. The net radiation into the pool was measured and this constituted one of the computer inputs.

Building Heat Load

The test building heat load was calculated according to ASHRAE standards to yield an effective $U \times A$ value of 74 BTU/°F-hr. The heat entering the test building air was calculated by

$$(eq. 31) \quad Q = U \times A (T_{out} - T_{in})$$

This treatment does not take into account the lag induced by wall capacitance nor the effect of insulation on the walls and roof. It was not the intent of this work to model the building itself.

Computer Simulation

The Cool Pool system was divided into ten nodes. The column was divided into 5 slices; the downcomer into 3 slices; the interior air was modeled as one node and the roof pond makes the tenth node. Each node was assumed to be at a uniform temperature. A heat balance was done for each node to generate a system of ten nonlinear first order differential equations. These equations were solved by a library program. A list of the nodes and their heat and mass transfer processes follows:

Computer Nodes

Node 1 The Roof Pond

- A) mass flow from volumn top (eq. 2,6)
- B) mass flow to downcomer top (eq. 2,6)
- C) evaporation to outside air (eq. 17)
- D) convection to outside air (eq. 24-26)
- E) net radiation into pool (experimental)
- F) conduction to living space (eq. 29)
- G) conduction to outside (eq. 29)

Node 2-4 The Downcomer

- A) mass flow in (eq. 2,6)
- B) mass flow out (eq. 2,6)
- C) conduction from water column (eq. 30)

Node 5-9 The Column

- A) mass flow in (eq. 2,6)
- B) mass flow out (eq. 2,6)
- C) radiation to building interior (eq. 8)
- D) convection to interior air (eq. 10)
- E) conduction to downcomer from lowest node only (eq. 30)

Node 10 The Interior Air

- A) radiation from column (eq. 8)
- B) convection from column (eq. 10)
- C) building load (eq. 31)

Hourly Inputs

- C(2) outside temperatures
- C(3) absolute humidity
- C(4) net radiation into pool
- C(5) building heat load

The specifiable parameters such as cool pool geometry are listed in Appendix I. These parameters can be read in so that Cool Pool systems of varying sizes can be simulated. Appendix I also lists the variable values such as viscosity and density which are calculated each hour or each time step depending on their importance and impact.

The model was verified by initializing the ten node temperatures to match the experimentally measured temperatures. The program was allowed to simulate 24 hrs. of performance using hourly inputs (outside temperature, humidity, net radiation into pool, and building load) which were the same as those measured during the experiment. The node temperatures and the thermosiphoning rate were then compared to the experimental values with very good agreement.

Heat transfer from the air to the column is calculated using average air and column temperatures. This gives good results and eliminates the necessity of modeling stratification in the air. From the air temperature profiles it can be seen that air stratification is dependent on the outside temperature history, building design, and capacitance (to name a few factors). The following graph shows the average of the experimental air temperatures compared to the computer simulation.

Average Air Temperatures

Due to instrument difficulties no data was recorded at 10 p.m. and 11 p.m. The air temperature was extremely sensitive to the building load estimate since it had little capacitance and factors such as infiltration could only be crudely estimated. However the greatest deviation in temperature was 2.7°F (1.5°C)

Conduction from the column to the downcomer was calculated for adjacent nodes but the net heat transferred into the three downcomer nodes was only subtracted from the lowest column node. This simulates convective loops within the column which enhance water stratification. Mass leaves the node at the node temperature and enters the node of the previous node's temperature. Then the new isothermal node temperature is determined. This method yields close agreement to the measured column temperatures as shown below.

Column Temperatures

The graph below shows experimental and pool temperatures. Again, there is excellent agreement. In this case the pool was probably larger than required for the heat load from thermosiphoning. Thus, the rest of the system had little impact on it.

Pool Temperature

The pool temperature was critical in determining the thermosiphoning rate since the colder downcomer water was greater than the pressure difference was. Figure 14 shows the calculated vs. experimental thermosiphoning rate. Near midnight, the calculated rate was 15% lower than the measured rate. The rest of the time there was much better agreement.

Thermosiphoning

Here we come to the most important calculation, heat absorbed by the column. This is the factor which determines sizing. As discussed earlier, the experimental calculation could be off by 20% or more. The agreement with simulated values is well within 20%.

Downcomer Temperatures

Total Heat Into Column Wall

Conclusion

This work has increased the experimental data available on the thermosiphoning roof pond. Although conventional methods for calculating the thermosiphoning rate, heat transfer and evaporation do a good job modeling this system, it is clear that convective heat transfer relationships for enclosed spaces need to be developed.

Now that a good computer simulation is available for this system it will be easy to input other climate values and system parameters in order to size the Cool Pool for different regions.

The computer program is presented as a subroutine that can be inserted as a part of any other climatic/thermal performance program in Fortran; it follows on p. 99.

Sizing for Different Climates

A standard 1400 square foot solar house with R19 walls, R30 ceiling, 200 square feet of shaded south glass, 90 square feet of north glass was used to estimate the hourly cooling load for Sacramento, Tuscon, and Albuquerque. Three-hourly data on outside temperatures and absolute humidity for July, 1978 was averaged and interpolated to provide hourly climate input to the computer program. Tables I & II show the load which one 10' tall, 1½' diameter water column (approximately 40 ft.² surface area) can provide with different amounts of roof pond.

The tables can be used to size a cool pool system for one of the three climates listed. First calculate the peak hourly cooling load for your building. Then subtract cooling from other mass (such as the slab floor, internal sheetrock, etc.) Then divide the remaining unmet cooling load listed in the appropriate table. This yields the number of water columns (or equivalent 40 square feet of surface) required. Then the pool area (10, 15, 20 ft) is multiplied by the number of columns. This indicates the surface area of one foot deep roof pond required.

The key parameters are: the number of 1½" ID or larger connecting pipes, the water mass surface area, and the pool size.

TABLE I - PEAK HOURLY LOAD (BTU)

POOL AREA (FT ²)	10	15	20
Albuquerque	435	441	445
Sacramento	455	501	508
Tuscon	226	229	319

TABLE II - 24 - HOUR LOAD (BTU)

POOL AREA (FT ²)	10	15	20
Albuquerque	4804	5115	5176
Sacramento	4089	4566	4624
Tuscon	2986	3048	4286

Fortran Program

```

SUBROUTINE COMPD(T,X,P,C)
  DIMENSION P(20),C(22),T(2,10)
  TAVD=(T(1,2)*2+T(1,3)+T(1,4))/4
  C(7)=2.7897E-05-3.491E-078TAVD+
-1.458E-098TAVD*TAUD
  C(8)=62.3302+6.704E-038TAVD-1.016E
-048TAVD*TAUD
  C(9)=-.240+.458C(3)
  CALCULATE HEAT OF VAPORIZATION
  C(4)=1091.4-.4958T(1,1)-.00058
-T(1,1)*T(1,1)
  VAPOR PRESSURE OVER POOL
  IF(C(2).LT.74.97) GO TO 100
  PUS=EXP(-3.004+.04318T(1,1)-6.5016
-E-058T(1,1)*T(1,1))
  GO TO 101
  PUS=EXP(-3.078+.045278T(1,1)-8.010
-1E-058T(1,1)*T(1,1))
  C(10)=-.4228PUS/(P(18)-PUS)
  DENSITY PRESSURE DIFFERENCE
  DDM=6.704E-038((P(7)-P(1))*T(1,1)+
-T(1,2)+T(1,3)+T(1,4))*848P(5)/15.-
-T(1,6)+T(1,7)+T(1,8)+T(1,9))*8P(
-5)/5)+1.016E-048((T(1,6)*882+T(1,7)
-882+T(1,8)*882+T(1,9)*882)*8P(5)/5-
-P(7)-P(1))*8T(1,1)*882-(T(1,2)*882+
-T(1,3)*882+T(1,4)*882)*848P(5)/15)
  CALCULATE VELOCITY
  IF(DDM.LT.0) GO TO 800
  A=1.
  B=64.8C(7)*P(2)/(P(1)*P(3))
  S=-2.832.17*DDM/(C(8)*P(3))
  C(11)=-B+(B*B2.-4888S*S)/(28A)
  GO TO 801
  C(11)=0.
  CALCULATE MASS FLOW RATE
  C(17)=3.1428C(11)*P(1)*P(1)*862.4
-83600./4.
  CALCULATE HEAT TO COLUMN WALL
  C(12)=T(1,3)+T(1,6)+T(1,7)+T(1,8)
-4T(1,9))/5.
  C(13)=P(20)*8.173E-088((T(1,10)+
-459.69)*884-(C(12)+459.69)*884)
  C(14)=-.198((T(1,10)-C(12))*88(4/3))
  Q=3.1428P(6)*P(5)*8(C(13)+C(14))/5
  C(16)=-580
  CALCULATE POOL EVAP & CONV
  P(11)=0.8008C(4)/(C(9)*C(9))
  CONV=P(8)*P(10)*8(C(2)-T(1,1))
  EVAP=P(8)*P(11)*8(C(3)-C(10))
  EQUATIONS
  T(2,1)=(C(17)*8(T(1,9)-T(1,1))+
-CONV+EVAP+C(4))/P(12)
  T(2,2)=(C(17)*8(T(1,1)-T(1,2))-P(4)
-8P(16)*8(28T(1,2)-T(1,9)-T(1,8))/2.)/
-P(13)
  T(2,3)=(C(17)*8(T(1,2)-T(1,3))+P(4)
-8(T(1,7)-T(1,3))*8P(16))/P(13)
  T(2,4)=(C(17)*8(T(1,3)-T(1,4))+P(4)
-8(T(1,6)-T(1,4))*8P(16))/P(13)
  T(2,5)=(C(17)*8(T(1,4)-T(1,5))+0
-8P(16)*8P(4)*8(T(1,2)+T(1,3)+T(1,4)
-T(1,6)-T(1,7)-.58(T(1,8)+T(1,9))))
-/P(14)
  T(2,6)=(C(17)*8(T(1,5)-T(1,6))+0)/P(14)
  T(2,7)=(C(17)*8(T(1,6)-T(1,7))+0)/P(14)
  T(2,8)=(C(17)*8(T(1,7)-T(1,8))+0)/P(14)
  T(2,9)=(C(17)*8(T(1,8)-T(1,9))+0)/P(14)
  T(2,10)=(C(5)-5.88)/P(15)
  RETURN

```

VARIABLE DEFINITIONS

```

C(2) Outside temp.
C(3) Outside abs. humidity
C(4) Net rad. into pool
C(5) Hourly building load
C(6) Heat of vapor. of water
C(7) Kinematic visc. of water
C(8) Water density
C(9) Spec. heat of moist air
C(10) Abs. humidity over pool
C(11) Thermosiphoning vel.
C(12) Average column temp.
C(13) Column radiation
C(14) Column convection
C(17) Mass flow rate
DDM Pressure caused by and density dif-
ference
P(1) Diam. of downcomer
P(2) L/d pipe friction
P(3) Pipe frict. expansion
P(4) Downcomer conduct.
P(5) Column height
P(7) Pool depth
P(8) Pool surface area
P(10) Conv. coef. to pool
P(11) Evap. coef. to pool
P(12) Pool heat capac.
P(13) Downcomer node capac.
P(14) Column node heat capac.
P(15) Air heat capacitance
P(16) Downcomer node area
P(17) Column node area
P(18) Total air pressure
P(20) Column emissivity
TAVD Ave. downcomer temp.
T(1,i) Node temperatures
T(2,i) Temp. derivatives

```

Summertime - 5% Design Wet-Bulb Temperature

In a normal summer, June through September, there would be approximately 150 hours (5%) at or above the temperature values plotted.

Cost Estimate

Cost estimates for construction of integrated components are hard to pin down for the following reasons:

Inflation during time delays between prediction and publication.

Several options for construction are presented. Costs may vary depending on techniques selected.

Prevailing usage rates vary considerably throughout the U.S. Material cost vary to a certain extent also.

Nevertheless, the following brief estimate is presented as of summer 1978 in northern California.

Shade, including shade support	- \$10/S.F. of shaded area
Pool(s)	- \$10-\$15/S.F. of pool surface area
Water Columns	- \$200 each
Plumbing	- \$70/tube

Published by Living Systems

Route 1, Box 170
Winters, California 95694
(916) 753-3033