[논문] 태양에너지
Solar Energy
Vol. 12. No. 1. 1992

태양열 이용 흡수식 냉난방시스템의 스프레드쉬트 모델

최 홍 규* · Rocco A. Fazzolari**
*한국 태양열 창출 주식회사
**Nuclear and Energy Engineering Dept. University of Arizona

Solar Absorption System Analysis with Spreadsheet Models

Choi, Hong Kyu* · Rocco A. Fazzolari**
* *Korea Solar Creative Cooperation*
** *Nuclear and Energy Engineering Dept. University of Arizona*

요 약

태양열이용 LiBr-water 흡수식 냉난방시스템의 시간단위의 simulation model을 SuperCalc spreadsheet를 사용해서 제시하였고 전체시스템의 열성능을 분석함으로써 spreadsheet simulation 방식의 가치를 입증하였다. Arizona, Tucson에 위치한 사무실형 건물의 시간단위의 냉난방 부하를 ASHRAE방식에 의하여 계산하였으며 전체시스템의 구성요소 및 제어방식에 대한 알고리즘을 제시하였다. 두 종류의 흡수식시스템의 열성능을 시스템매개변수를 이용해서 비교하였다.

ABSTRACT

An hourly simulation model of a solar LiBr-water absorption cooling and heating system (for brevity, solar absorption system) is presented, based on SuperCalc spreadsheet computational procedures. This paper demonstrates the value of using spreadsheet simulation techniques by examining the thermal performances of a solar absorption system. The hourly heating and cooling coil loads for a typical office building in Tucson, Arizona are modeled and calculated using ASHRAE methods. The details of the algorithms for the components and control schemes are presented. Two case studies are also presented using real system parameters.

1. INTRODUCTION

The primary objective of the project presented here was to develop a relatively simple spreadsheet computational approach to simulating a complex model of a solar absorption cooling and heating system. A microcomputer based spreadsheet program like SuperCalc is a convenient tool for the simulation programing of complex engineering models. A spreadsheet program is easy-to-use, has full-function computational capabilities, is simple to modify, and provides easy input/output presentations including graphics.

The overall model presented here is generic in the sense that the input parameters and, most importantly, the mathematical components are easily modified to describe variations in the design and operation of the system. A spreadsheet permits the rapid introduction of mathematical algorithms and computational sequences. Consequently, the modeling process is very flexible once the overall structure of the system model is placed in the spreadsheet. In the application described herein, specific real components are used, but the spreadsheet submodels can be conveniently changed.

An hourly prediction for a typical clear day in a month is used as a basis for the simulation period. The heating and cooling requirements for the months of January, April, July, and October were used to predict the annual performance of the system. Constant flow rates are assumed, consistent with manufacturers' performance data. In addition, a well mixed uniform temperature storage tank model is an important, possibly restrictive, assumption.

2. SYSTEM SCHEMATIC AND CONTROL

The entire system schematic is shown in Fig. 1. The solar portion of the model is patterned after an existing operating solar process heat plant in Coolidge, Arizona[1]. Absorption chillers have been added here to complete the model formulation. Four control modes and resulting heat transport and flow conditions are possible :

(1) collector to load and storage tank (partial tank charge)
(2) storage tank to load (discharge mode)
(3) collector to storage tank (charge mode)
(4) collector and storage tank to load (partial tank discharge)

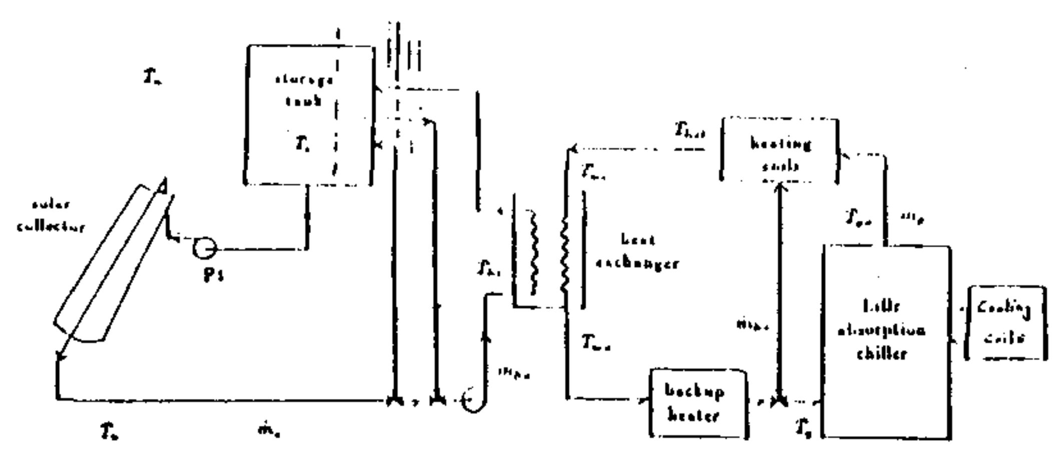

Fig. 1 Schematic Diagram of Solar Absorption System

The collector fluid flow rate $\dot{m}_c$ is assumed to be constant. The master controller of the system continuously monitors solar conditions at the collector field. The control function turns on the collector loop pump (P1) whenever the collector fluid outlet temperature exceeds the collector inlet temperature by 10°F or more, and turns it off otherwise. Therefore, if $(T_O - T_{f,in}) < 10°F$, $\dot{m}_c = 0$

At startup, the temperature of the heat transfer fluid (Caloria HT43TM) in the storage tank is assumed to be 200°F, the threshold temperature level required in the heat exchanger. When stored thermal energy is depleted to the point that less than 5°F differential temperature is sensed in the heat exchanger, the heat transfer fluid flow is stopped (i.e., $\dot{m}_{hx}=0$). The backup heater then takes over.

The absorption generator entering hot water temperature is fixed to a limited range of 170－190°F. The condenser entering water temperature is set at 85°F. The control schemes are designed to provide a preset hot water temperature to the generator as a function of the solar energy available and storage tank temperature.

The whole system consists of six submodels : building HVAC, solar collector, solar radiation, thermal storage tank, absorption chiller, and heat exchanger.

3. BUILDING MODEL

The building model is based on the ASHRAE cooling load temperature difference (CLTD) method including wall, roof and fenestration thermal characteristics, and the internal loads.[2] The air distribution system designs are specifically and mathematically modeled.[3] In this case, a dual-duct constant volume system is incorporated in the spreadsheet model. A typical University of Arizona building (45,000ft^2) is used to develop the input parameters.[1]

The CLTD and the cooling load factor(CLF) are used to estimate the hourly coil loads. The CLTD and CLF change with time and are a function of environmental conditions and building parameters.

The operation schedule for the air conditioning system is from 7 : 00 a.m. to 7 : 00 p.m., with the weekend off. Infiltration is assumed to be negligible., Fig. 2 and Fig. 3 show the spreadsheet computed hourly cooling and heating coil loads on the 21st days in January, April, July, and October.

4. SOLAR RADIATION MODEL

The single axis tracking parabolic trough collectors in the model utilize direct radiation only. On a clear day, the instantaneous direct solar radiation per unit area I_D can be predicted by[2]

$$I_D = A \exp\left[\frac{-B}{\sin\alpha}\right] \cos\theta \quad \cdots\cdots\cdots\cdots\cdots\cdots (1)$$

where A is the apparent solar radiation at zero air mass ; B is the atmospheric extinction coefficient ; α is the solar altitude angle ; θ is the solar incident angle.

For a plane tracking the sun about a horizontal axis, the angle of incidence can be expressed in terms of solar angles and geometric angles.[5] At the mean hour angle during a time step, I_D in eqn(1) is approximately the average value for the period.

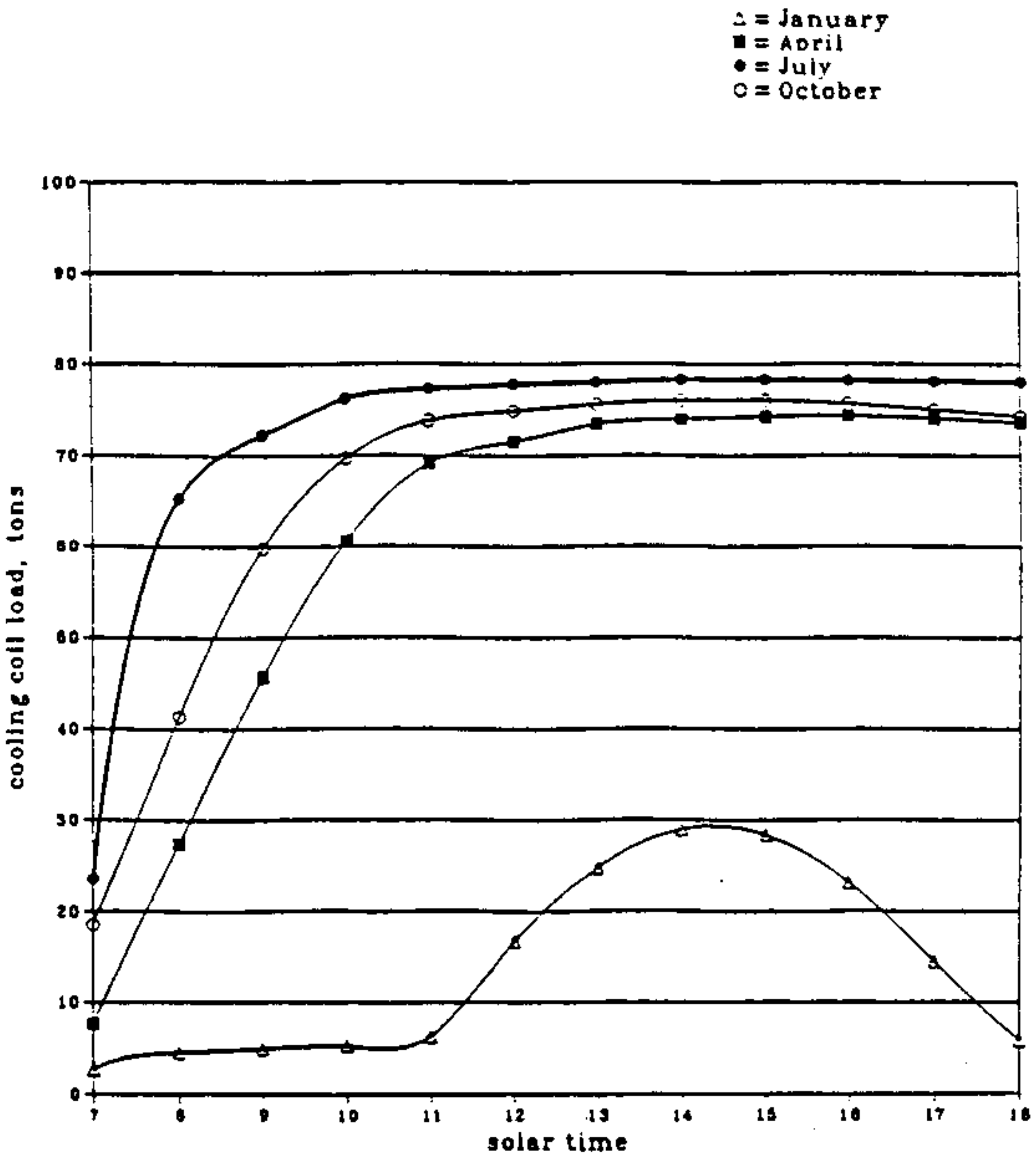

Fig.2 Hourly Cooling Coil Loads

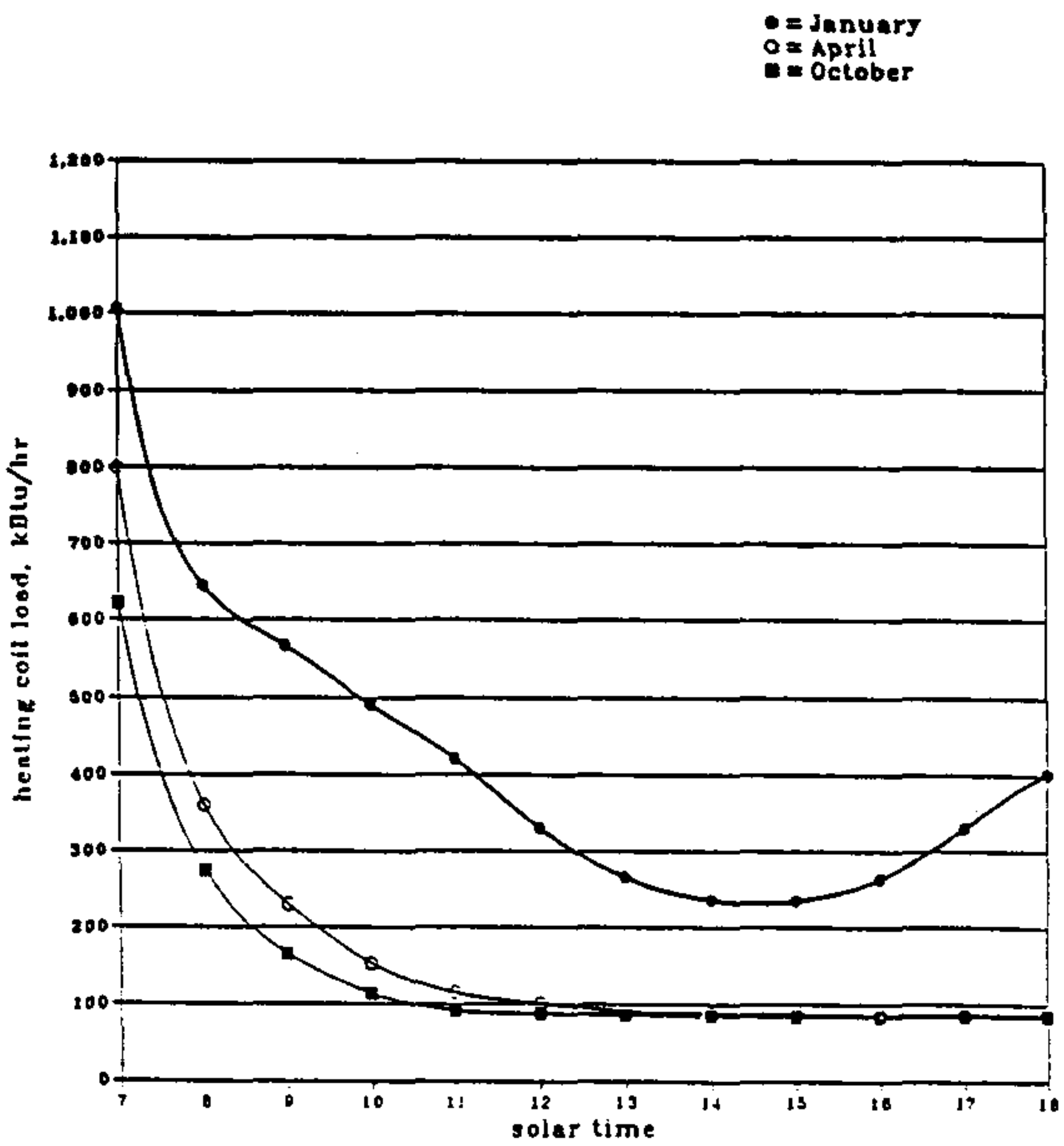

Fig.3 Hourly Heating Coil Loads

The overall fraction of array area that is shaded at any time is a function of the shaded fraction for a single row of collectors and the number of rows N_R as[6]

$$f_{bs} = \frac{(N_R - 1)\, f_{bsf}}{N_R} \quad \cdots\cdots\cdots\cdots\cdots\cdots \quad (2)$$

where f_{bsf} is the fraction of the aperture area shaded by an adjacent row of collectors, neglecting collector edge losses.

The effects of the incident angle θ on collector edge losses are combined into a function F(θ) as follows[7] :

$$F(\theta) = (1 - A \tan\theta) \quad \cdots\cdots\cdots\cdots\cdots\cdots \quad (3)$$

where the term A tan θ expresses the apparent reduction in aperture area due to blockage and end losses. The coefficient A is a geometric factor for the specific collector design.

The incident angle modifier is negligible when the incident angle is less than 60° as will be the case most often with tracking trough collectors. Considering shading and end losses, the incident averaged solar direct radiation per unit of aperture area for a time step is computed in the spreadsheet by

$$\bar{I} = \bar{I}_b (1 - f_{bs}) F(\theta) \quad \cdots\cdots\cdots\cdots\cdots\cdots \quad (4)$$

Fig. 4 shows the calculated incident solar radiation falling on the collector for typical clear days in the four months. The north-south axis delivers the maximum yield in July but the lowest yield in January. The east-west axis results in more even thermal output throughout the year.

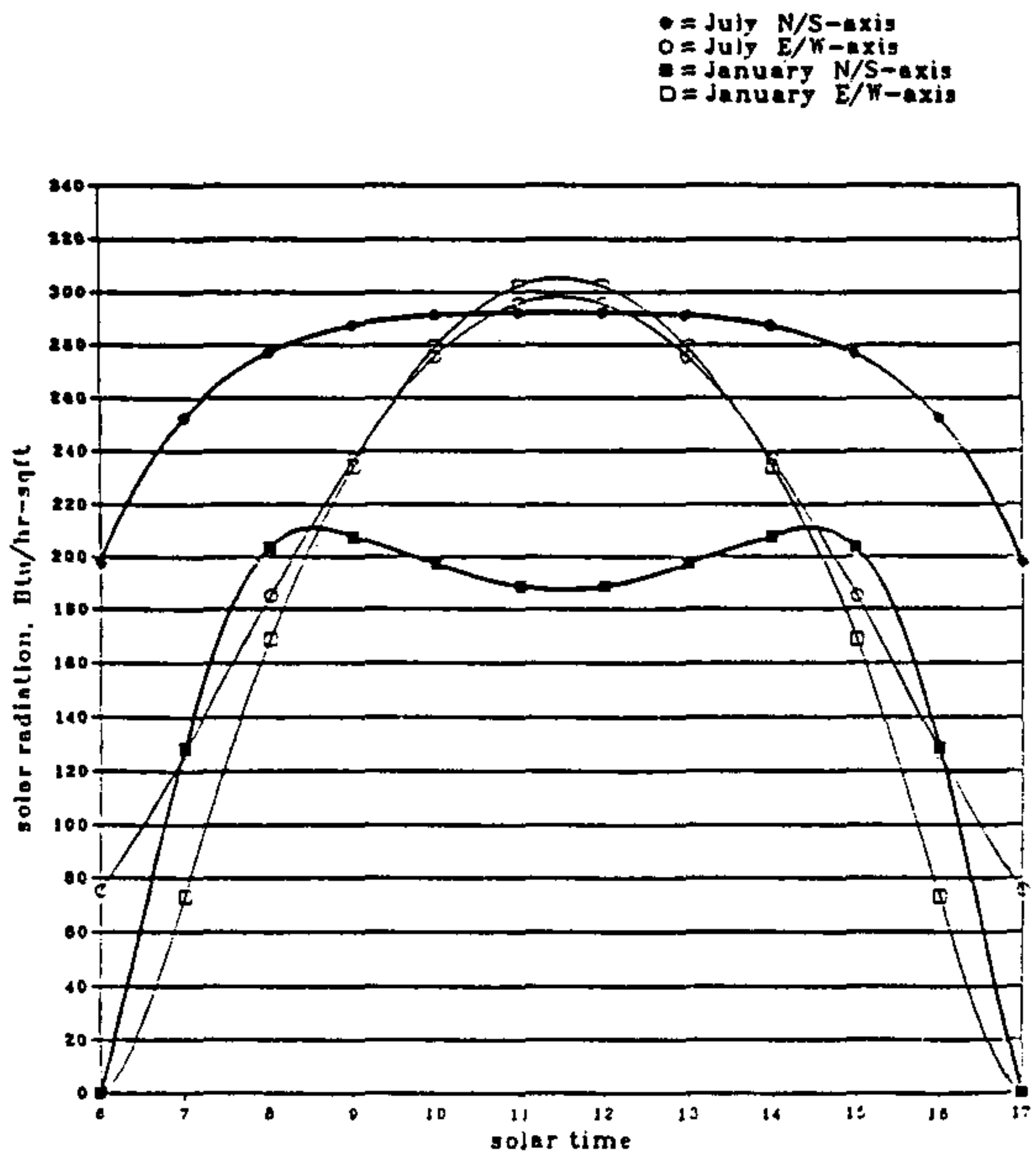

Fig.4 Incident direct Solar Radiation in Tucson, Arizons

5. SOLAR COLLECTOR MODEL

The Solar Kinetics T−700A collector was used for this model. Based on performance tests of the T−700A,[8] the collector instantaneous efficiency η_c can be expressed by the following equation over a wide range of temperatures and solar radiation I :

$$\eta_c \doteq .7358-.000154\triangle T-.0332\frac{\triangle T}{I} -.000121\frac{\triangle T^2}{I} \quad\cdots\cdots (5)$$

where

$$\triangle T = T_f - T_a \approx \frac{T_{f,in}+T_{f,out}}{2} - T_a, \ {}^\circ F$$

The corresponding hourly averaged form of the above equation can be expressed as

$$\bar{\eta}_c = .7358-.000154\triangle \bar{T}-.0332\frac{\triangle \bar{T}}{\bar{I}} -.000121\frac{\triangle \bar{T}^2}{\bar{I}} \quad\cdots\cdots (6)$$

where

$$\triangle \bar{T} \approx \frac{\bar{T}_s+\bar{T}_0}{2} - \bar{T}_a$$

Also, $\bar{T}_s$ is the average tank temperature ; $\bar{T}_0$ is the average collector outlet temperature ; $\bar{T}_a$ is the average ambient air dry bulb temperature ; $\bar{I}$ is the average solar radiation.

The available energy input $q_c\triangle t$ from the solar collector for the time step $\triangle t$ is

$$q_c\triangle t = \bar{I}\eta_c\triangle t = (\dot{m}c_p)_c(\bar{T}_0-\bar{T}_s)\triangle t \quad\cdots\cdots (7)$$

Combining eqn(6) and eqn(7) results in a quadratic equation for $\bar{T}_0$ which is solved by the spreadsheet program. The collector outlet temperature $\bar{T}_0$ is solved with an initial guess of $\bar{T}_s$. q_c and $\bar{T}_0$ in eqn(7) are adjusted to reflect the thermal losses in the piping.

6. STORAGE TANK MODEL

The storage tank is assumed to be completely mixed at uniform temperature $\bar{T}_S$. $\bar{T}_S$ is approximated by averaging the initial storage temperature T_{Si} and the final storage temperature T_{Sf} over the simulation time step of $\triangle t$.[9]

$$\bar{T}_s \approx \frac{(T_{si}+T_{sf})}{2} \quad\cdots\cdots (8)$$

$\bar{T}_s$ is a dependent variable, which depends on the initial storage tank temperature, the solar energy collection, the thermal energy removal for the load, and the ambient temperature.

An energy balance on the storage tank requires the energy input from the collector, the energy transfer as heat in the heat exchanger, the energy losses to the environment, and the internal energy change of the stored oil as follows :

$$q_c\triangle t = q_{hx}\triangle t + (UA)_s(\bar{T}_s-\bar{T}_a)\triangle t + (Mc_p)_s(T_{sf}-T_{si}) \quad\cdots\cdots (9)$$

where q_{hx} is the heat transferred in the heat exchanger ; $(Mc_p)_s$ is the thermal mass of stored oil in the storage tank.

Combining eqn (8) and eqn (9), the storage tank average temperature $\bar{T}_s$ over the simulation period is calculated by

$$\bar{T}_s = \frac{(q_c-q_{hx})\triangle t + (U\ A)_s\bar{T}_a\triangle t + 2(Mc_p)_sT_{si}}{(U\ A)_s\triangle t + 2(Mc_p)_s} \quad\cdots\cdots (10)$$

The accuracy of eqn (10) depends on the validity of eqn (8) and the duration of the time step. $\bar{T}_s$ in eqn (10) is used as a next guess for solving the quadratic equation of $\bar{T}_0$ until the two $\bar{T}_s$ became sufficiently close with each other. The net energy change in the storage tank q_{net} over the simulation time step of $\triangle t$ is given by

$$q_{net} = [q_c - q_{hx} - (UA)_s(\bar{T}_s-\bar{T}_a)]\triangle t = (Mc_p)_s(T_{sf}-T_{si}) \quad\cdots\cdots (11)$$

The final temperature of the storage tank T_{sf} is found from eqn (11). T_{sf} is used as the initial tank temperature T_{si} in the next time step of the simulation.

7. ABSORPTION CHILLER MODEL

A Yazaki LiBr-water absorption chiller is used as the simulation model.[10] Absorption chillers can be classified as single-effect and double-effect depending upon the stages of generation. A single/double-effect chiller combines a single-effect and a double-effect system so as to permit the direct use of hot water and gas heat in the chiller. Backup heat can be supplied without the need of an additional boiler. With this approach, solar heat and/or gas can be used conveniently to meet the space heating and cooling loads.

The coefficient of performance(COP) and the capacity factor(CF) are functions of the condenser entering water temperature and the generator entering water temperature ; they are defined by

$$COP = \frac{\text{hourly cooling load}}{\text{energy input to generator}} \cdots \quad (12)$$

$$CF = \frac{\text{hourly cooling load}}{\text{nominal cooling capacity}} \cdots \quad (13)$$

The cooling requirements are supplied by adjusting the number of operational absorption units and their operation time. The number of the operational units N and the operation time OT are calculated by

$$N = \text{integer}\left[\frac{\text{hourly cooling load}}{(CF)\ (NCAP)}\right] + 1 \quad \cdots\cdots \quad (14)$$

$$OT = \frac{\text{hourly cooling load}}{(N)\ (CF)\ (NCAP)} \quad \cdots\cdots \quad (15)$$

where NCAP is the nominal cooling capacity per unit.

The hot water flow rate to be supplied to the generator $\dot{m}_g$ and the backup heater gas consumption q_{gas} are given by following :

$$\dot{m}_g = (N)\ (\text{water flow rate specified per unit}) \quad \cdots\cdots \quad (16)$$

$$q_{gas} = \max\left[0, \frac{(\dot{m}c_p)_g(\bar{T}_g - \bar{T}_{wo})}{\eta}\right](OT) \quad \cdots\cdots \quad (17)$$

where η is the heater efficiency ; $\bar{T}_{wo}$ is the heat exchanger leaving water temperature.

For the single-effect chiller using hot water only, the entering temperature is fixed at 190°F to obtain reasonable cooling efficiencies. In the case of the single/double-effect, the system can be energized by hot water in the range of 170－190°F because of the added input of gas. The CF and COP depend on the generator entering water temperatures ; they are determined using empirical performance data. At 190°F or above, the hot water driven mode only is used. Lower than 170°F, the gas-fired mode takes over. Otherwise, both hot water and gas are used. The hot water contribution to the generator is calculated by

$$q_{water} = \frac{\text{hourly cooling load}}{COP} - q_{gas}\eta \cdots \quad (18)$$

The generator leaving water temperature $\bar{T}_{go}$ is found by an energy balance on the generator.

$$\bar{T}_{go} = \bar{T}_g - \frac{q_{water}}{(\dot{m}c_p)_g} \quad \cdots\cdots \quad (19)$$

If there is a space heating load requirement, the generator leaving water is allowed to flow through a heating coil to supply warm air to the space. The heating coil leaving water temperature $\bar{T}_{hcl}$ is calculated by

$$\bar{T}_{hcl} = \bar{T}_{go} - \frac{\text{hourly heating load}}{(\dot{m}c_p)_g} \quad \cdots\cdots \quad (20)$$

When the hot oil temperature in the storage tank is not sufficient to supply the heating demand, a backup heater is used with a single-effect chiller. The single/double-effect chillers, however, can also be operated directly in heating mode.

Considering the thermal losses in the piping, the heat exchanger entering water temperature $\bar{T}_{wi}$ is

$$\bar{T}_{wi} = \bar{T}_{hcl} - \triangle T \quad \cdots\cdots (21)$$

where $\triangle T$ is the temperature drop in the piping.

The water pumping power and the tower fan power were calculated using the manufacturer's design performance data and the part load ratio.

8. HEAT EXCHANGER MODEL

A shell and tube type heat exchanger is used in the model to transfer thermal energy from the hot Caloria fluid to the water. The U-value of the heat exchanger is assumed to be constant ; this is a reasonable assumption for compact single-phase heat exchangers.[11] The NTU(number of transfer units) method[12] is used to predict the performance of the heat exchanger. Due to the cooling load variations, the fluid flow rates of $\dot{m}_{hx}$ and $\dot{m}_g$ are regulated to supply adequate thermal energy to the absorption generator. The heat exchanger effectiveness ε is, therefor, varying. The heat available at the heat exchanger q_{hx} and the required amount of energy q_{req} are calculated by eqn (22) and eqn (23), respectively.

$$q_{hx} = \varepsilon C_{min}(\bar{T}_{hi} - \bar{T}_{wi}) \ (OT)$$
$$= (\dot{m}c_p)_g(\bar{T}_{wo} - \bar{T}_{wi}) \ (OT) \quad \cdots\cdots (22)$$
$$q_{req} = (\dot{m}c_p)_g(\bar{T}_g - \bar{T}_{wi}) \ (OT) \quad \cdots\cdots (23)$$

where

$$C_{min} = \frac{\min[(\dot{m}c_p)_{hx}, \ (\dot{m}c_p)_g]}{\max[(\dot{m}c_p)_{hx}, \ (\dot{m}c_p)_g]}$$

The hot oil flow rate $\dot{m}_{hx}$ is adjusted to meet the required heat transfer according to

$$\dot{m}_{hx} = \max\left[\dot{m}_c, \ \frac{q_{req}}{\varepsilon \ c_{p,hx}(\bar{T}_{hi} - \bar{T}_{wi})}\right] \cdots (24)$$

However, the flow must be limited to a realistic value because of the pumping requirements. In this case, the upper limit was set to $2\dot{m}_c$. $\bar{T}_{hi}$ is then determined by

$$\bar{T}_{hi} = \frac{\dot{m}_c[\bar{T}_o(CS) + (1-CS)\bar{T}_s] + (\dot{m}_{hx} - \dot{m}_c)\bar{T}_s}{\dot{m}_{hx}} \quad \cdots\cdots (25)$$

where CS=control function of solar loop (if $\bar{T}_o - \bar{T}_s \geq 10°F$, 1, otherwise, 0).

Iteration commands are conveniently available in SuperCalc and other spreadsheets. The three parameters ($\dot{m}_{hx}$, $\bar{T}_{hi}$, and ε) are iterated on within the specified range of $\dot{m}_{hx}$ until they were converged into the final values. The value of $\bar{T}_{wo}$ is found from eqn (22) when the iteration is terminated. If $\bar{T}_{wo}$ is higher than 190°F, the flow rate $\dot{m}_{hx}$ should be reduced to obtain $\bar{T}_g$ at 190°F. The following algorithm is used to limit $\bar{T}_{wo}$.

$$\bar{T}_{wo} = \min(190°F, \ \ \bar{T}_{wo}) \quad \cdots\cdots (26)$$

9. SPREADSHEET PROCESS

The SuperCalc[13] spreadsheet provides a matrix of 2,549,745 cells in which mathematical formulas, numerical data, and text can be entered, manipulated, and modified. The matrix layout, with hundreds of columns and thousands of rows, is ideal for a wide variety of spreadsheet applications in which large sets of numbers and formulas need to be computed and organized. Each cell has its own address on the spreadsheet ; the address is defined by the intersection of a column and a row.

In this application, the solar time is arranged in the first column ; therefore, each row reflects the corresponding solar time in sequence. Each column contains a dependent mathematical formula which essentially follows the sequence of information beginning with the solar radiation calculation and ending with the solar fraction. Fig. 5 shows a small portion of the spreadsheet and the arrangement of mathematical formulas. Each cell on the screen normally displays the values associated with the inherent formulas, however, it is possible to print in hardcopy either the values or the formulas. Fig. 6 represents mathematical and logical formulas for a few typical columns used in the simulation.

The abilities of SuperCalc to replicate formulas make a spreadsheet easy to program a complex model. The iteration capabilities make this simulation possible.

The system model developed here involves three major subsystems (solar, absorption, and HVAC) ; the subsystems are integrated by the use of logical control schemes. Complex control schemes are easier to introduce and verify in a spreadsheet than they are in a conventional computer program.

Spreadsheet permits the easy observation of detail numerical values for every algorithm used without tedious input/output programing and formating. The spreadsheet itself serves the function of a well organized output sheet on the monitor or in printed form. Errors can be detected and corrected at each step without difficulty. Moreover, the computational procedures can be automated with macros which allow complex interactions within the spreadsheet or between files.

10. CASE STUDIES

Two case studies are presented :

(1) single-effect chiller with T-700A collector

	AJ	AK	AL	AM
1 2 3	solar time (hr)	hour angle (degree)	sin(altitude angle)	control function daytime
4	6	15*(AJ5-12+.5)	C15*COS(AK5*C12)+C16	IF(AL5).1,1,0)
5	7	15*(AJ6-12+.5)	C15*COS(AK6*C12)+C16	IF(AL6).1,1,0)
6 7	8	15*(AJ7-12+.5)	C15*COS(AK7*C12)+C16	IF(AL7).1,1,0)
8	9	15*(AJ8-12+.5)	C15*COS(AK8*C12)+C16	IF(AL8).1,1,0)
9	10	15*(AJ9-12+.5)	C15*COS(AK9*C12)+C16	IF(AL9).1,1,0)

	AJ	AK	AL	AM
1 2 3	solar time (hr)	hour angle (degree)	sin(altitude angle)	control function daytime
4	6	-82.50	.29	1
5	7	-67.50	.49	1
6 7	8	-52.50	.67	1
8	9	-37.50	.82	1
9	10	-22.50	.92	1

Fig. 5 Spreadsheet with Occupied Cells

COLUMN	LEGENDS	FORMULAS
BL	# of units for gas heating	IF(BK7="heating", INT(BG7/H19)+1, 0)
BM	op-time for gas heating	IF(BK7="heating", BG7/(H19*BL7), 0)
BN	gas use for gas heating	BL7*BM7*H19*1000/(H33*C19*C20)
BO	# of units for water cooling	MIN(H21−BL7, BF7*(INT(BE7/(H25*H20))+1))
BP	of-time for water cooling	IF(BO7=0, 0, BF7*BE7/(H20*H25*BO7))
BQ	heat exchanger entering water temperature	CX7
BR	heat exchanger water flow rate	BF7*BO7*H20*H16/(C19*C20)
BS	heat exchanger input required	BP7*BR7*(H24−BQ7)
BT	heat exchanger oil flow rate	IF(BP7=0, 0, BF7*MIN(2*C26, MAX(C26, BS7/((CA7+.0001)*C23*(BV7−BQ7)))))
BU	control function	IF(BT7 > C26, 1, 0)

Fig. 6 Mathematical and Logical Formulas

(2) single/double-effect chiller with T−700A collector.

A north-south axis tracking was used to obtain the most solar energy during the summer months. The overall solar system performance is stated in terms of the solar fraction. The four month energy requirements for the two cases are summarized as follows :

Collector, ft^2	4,200	5,040	5,880	6,720	7,560
(Case-1)					
Electricity, kWhr	9.52	9.56	9.59	9.63	9.65
Natural gas, MBtu	745	658	566	471	376
Solar fraction	0.36	0.39	0.43	0.47	0.52
(Case-2)					
Electricity, kWhr	9.31	9.70	9.90	10.26	10.54
Natural gas, MBtu	526	445	381	321	261
Solar fraction	0.48	0.57	0.63	0.69	0.75

From the performance comparison, Case-2 shows a better solar fraction than Case-1 due to the improved COP and efficient accommodation to the heating or cooling needs. It is easy to extend the above results to obtain economic performance indicators for the systems under various operating conditions and design parameters.[14]

11. CONCLUSIONS

In this paper, a comprehensive and complex solar system model for heating and cooling a commercial building is presented which includes the building loads, the HVAC systems, the solar system, and absorption chillers. A complex logical control scheme is required to integrate the various components into an effective mathematical model of a real system.

In order to compare the more conventional programing approach to the use of personal computer (PC) based spreadsheets, the model described herein was also programed in FORTRAN. The primary benefit of using the mainframe version was the computational speed. A one day run on a 6MHz, 80286 PC took~5 minutes while on a VAX−11/780 a similar run took~20 seconds. However, the newer 80386 based personal computers operating at 25 MHz will close the computational time gap.

This effort also showed that spreadsheets are convenient simulation tools for complex systems. They are easily applied to solar system performance calculations. Spreadsheet models are convenient and appropriate for doing parametric studies and examining design alternatives. The formating of input-output information required in conventional computer programs is eliminated with spreadsheets ; graphic output presentations are also easy to obtain. The spreadsheet modeling approach provides a very transparent exposition of mathematics and information flow to both the programer and the user. All the algorithms and logic used can be conveniently displayed on a monitor or with a printer showing with details of each calculation.

REFERENCES

1. Fazzolari, R. A., et al., "Solar Process Heat System (SPHS) Going Strong," Strategic Planning for Cogeneration and Energy Management pp.451-453, Fairmont Press, Inc., Atlanta, GA. 1986.
2. ASHRAE Handbook, Fundamentals Volume, American Society of Heating, Refrigerating, and Air Conditioning Engineers, Atlanta, GA, 1981.
3. Knebel, D. E., Simplified Energy Analysis Using the Modified Bin Method, American Society of Heating, Refrigerating, and Air Conditioning Engineers, Atlanta, GA, 1983.
4. Fazzolari, R. A., et al., ESP Default Inputs and The Generic Building, Prepared for The University Utility Management and Planning Project, Tucson, AZ, December 1984.
5. Duffie, J. A. and Beckman, W. A., Solar Engineering of Thermal Processes, Wiley, New York, 1980.
6. Klein, S. A., et al., TRANSYS, A Transient Simulation Program, Solar Energy Laboratory, University of Wisconsin, Madison, Wisconsin, December 1983.
7. Ramsey, J. W., et al., "Experimental Evaluation of a Cylindrical Parabolic Solar Collector," *Journal of Heat Transfer*, Vol. 99, pp.163-168, May 1977.
8. Dudley, V. E. and Workhoven, R. M., Performance Testing of the Solar Kinetics T-700A Solar collector, Prepared by Sandia National Laboratory (SAND 81-0984), Albuquerque, NM, November 1982.
9. Lunde, P. J., Solar Thermal Engineering, Wiley, New York, 1980.
10. Technical Information Bulletin : Gas and Solar Air Conditioning Equipment, American Yazaki Corp. 13740 Omega Road, Dallas, Texas.
11. Lienhard, J. H., A Heat Transfer Textbook, Prentice Hall, Englewood Cliffs, NJ, 1981.
12. Kays, W. M. and London, A. L., Compact Heat Exchangers, McGraw-Hill, New York, 1964.
13. SuperCalc4 Spreadsheet Graphics and Data Management, Computer Associates International, Inc., San Jose, CA, 1986.
14. Fazzolari, R. A., "The Economics of Commercial Solar Heating and Cooling", *Proceedings of the ASME Solar Energy Conference*, Honolulu, Hawaii, March, 1987.

ABSTRACTS

SOLAR ENERGY VOL. 12, NO.1, 1992

An Experimental Study on the Interior Visual Environment of Classrooms in the Existing School Buildings

Lim, Sang-Hoon · Chun, Won-gee · Auh, Chung-moo · Lee, Nam-Ho
Korea Institute of Energy Research

The conventional school building design, which locates the hallway on the north side of the classroom, has proven to be very ineffective for indoor light control. Especially, glares and dark spots on the work plane as well as nonuniform illuminance on the chalkboard were causing undesirable visual effects for the students. The present study has investigated various problems with the classroom in the existing elementary and(junior) high school buildings from the point of indoor visual environment, and made some suggesions for possible solutions.

Solar Absorption System Analysis with Spreadsheet Models

Choi, Hong Kyu · Rocco A. Fazzolari
*Korea Solar Creative Cooperation
**Nuclear and Energy Engineering Dept. University of Arizona*

An hourly simulation model of a solar LiBr-water absorption cooling and heating system (for bervity, solar absorption system) is presented, based on SuperCalc spreadsheet computational procedures. This paper demonstrates the value of using spreadsheet simulation techniques by examining the thermal performances of a solar absorption system. The hourly heating and cooling coil loads for a typical office building in Tucson, Arizona are modeled and calculated using ASHRAE methods. The details of the algorithms for the components and control schemes are presented. Two case studies are also presented using real system parameters.C

Natural Convection Heat Transfer from a Hot Body in an Inclined Square Enclosure

Kwon, Sun-Sok* · Chung, Tae-Hyun**