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**DEVELOPING AND UPGRADING OF SOLAR SYSTEM THERMAL ENERGY
STORAGE SIMULATION MODELS**

Technical Progress Report for September 1, 1978—February 28, 1979

By
J. K. Kuhn
G. F. von Fuchs
A. P. Zob

May 1979

Work Performed Under Contract No. FG-77-C-02-4482

Boeing Computer Services Company
Seattle, Washington

U.S. Department of Energy

Solar Energy

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CONTENTS

	<u>Page</u>
ABSTRACT	v
1.0 INTRODUCTION	1
2.0 PROJECT OBJECTIVES	2
3.0 SUMMARY OF PRIOR ACTIVITIES	3
4.0 PROGRESS	7
4.1 ROCK BED COMPUTER MODEL PROGRESS	7
4.1.1 Model Development	7
4.1.2 Validation	12
4.1.3 Discussion of Discrepancies	21
4.1.4 Need For Effective Rock-Beds	25
4.1.5 Conclusions	28
4.2 LIQUID THERMAL ENERGY STORAGE MODELS	30
4.2.1 Introduction	30
4.2.2 Temperature Stratification in Storage Tanks	31
4.2.3 Computer Models for Storage Tanks	34
4.2.3.1 CSU Model	35
4.2.3.2 Extended SOLSYS	36
4.2.4 Simulation of Storage Tank Performance	39
4.2.4.1 Steady-State Charge and Discharge Tests	40
4.2.4.2 Comparison With Experimental Data	43
4.2.4.3 Transient Test	45
4.2.4.4 Storage Capacity Requirement	50
4.2.4.5 Performance Bounds	54
4.2.4.6 Simulation Accuracy Versus Computer Cost	57
4.2.5 Conclusions	57
5.0 PLANNED FUTURE ACTIVITY	60
6.0 REFERENCES	61

LIST OF FIGURES

	<u>Page</u>
3-1-a Rock-Bed Equations From Prior Six Months Report Dated September 1978	4
3-1-b Rock-Bed Equation Notation	5
4.1-1 Comparisons of Numerical Techniques	14
4.1-2 Internal Bed Temperatures, Charging Test from U.W.	16
4.1-3 Exit Air Temperatures, Charging Test from U.W.	18
4.1-4 Internal Bed Temperatures, Discharging Test from U.W.	19
4.1-5 Exit Air Temperature, Discharging Test from U.W.	20
4.1-6 Internal Bed Temperatures, Charging Data from CSU	22
4.1-7 Internal Bed Temperatures With Modified CSU Charging Data	26
4.1-8 Costs for Fan Power and Effective Rock-Beds	29
4.2-1 Schematic Diagram of Storage Tank in Solar Heating System	33
4.2-2 Development of the Extended SOLSYS Concept	38
4.2-3-a Evolution of Temperature Profiles During Charge Test - Simulation with CSU Model (N = 5)	41
4.2-3-b Evolution of Temperature Profiles During Charge Test - Simulation with CSU Model (N = 100)	41
4.2-3-c Evolution of Temperature Profiles During Discharge Test - Simulated with CSU Model (N = 100)	42
4.2-3-d Evolution of Temperature Profiles During Discharge Test - Simulated with Extended SOLSYS Model	42
4.2-4 Comparison of Temperature Profiles Generated by CSU Model with Test Results	44

LIST OF FIGURES (Continued)

	<u>Page</u>
4.2-5 Outlet Temperature vs. Time During Discharge Test	46
4.2-6 Transient Simulation Input Variables	46
4.2-7 Development of Temperature Profiles During Transient Charge Test	48
4.2-8 Variation in Outlet Temperature to Load During Transient Simulation	49
4.2-9-a Storage Efficiency of Perfectly Stratified Tank as a Function of Minimum Acceptable Outlet Temperature for Various Values of Storage Volume	51
4.2-9-b Storage Efficiency of Partially Stratified Tank as a Function of Minimum Acceptable Outlet Temperature for Various Values of Storage Volume	52
4.2-10 Comparison of Optimized Storage Efficiency Curves for Perfectly Stratified and Partially Stratified Tanks	53
4.2-11 Storage Efficiency of a Completely Mixed Tank as a Function of Minimum Acceptable Outlet Temperature for Various Values of Storage Tank Volume	55
4.2-12 Comparison of Optimized Storage Efficiency Curves for Perfectly Stratified and Completely Mixed Storage Tanks	56
4.2-13 The Trade-off Between Accuracy and Computer Cost	58

ABSTRACT

This progress report summarizes work performed in the third six month period of a three year project entitled "Developing and Upgrading of Solar Systems Thermal Energy Storage (TES) Simulation Models." The project objectives, progress, and plans are summarized below.

OBJECTIVES

- Collect, standardize, and link existing TES models from the literature and other STO contractors.
- Correlate TES models with available TES component data.
- Develop streamlined versions of validated TES component models.
- Provide DOE and industry with a competent solar TES simulation tool.

PROGRESS

Progress during this reporting period was primarily concerned with the first and second objectives.

- The rock bed model selected for validation was further explored and compared to test data. Correlation with data from different experiments is showing improvement over the correlations reported in the last six month progress report. The present model is considered acceptable at this time but more confirmation of its accuracy would be desirable.

- Two more water tank models have been obtained, one from the University of Alabama and one from Colorado State University (CSU). Work has continued with model testing, validation and evaluation. Additional laboratory test data on water tank performance has been obtained. Some significant improvements have been made in the SOLSYS algorithm resulting in a more powerful simulation tool referred to as the Extended SOLSYS Model. That and the CSU model have been selected for streamlining and for incorporation into TRNSYS.

PLANS

Plans for the immediate future are concerned with the first, second and third objectives.

- Validation of the upgraded rock-bed model will be continued and the model streamlining effort initiated.
- The current water tank models will be further validated and the streamlining effort initiated. Validation of the CSU model and the Extended SOLSYS model will continue. Both of these models will be incorporated into TRNSYS in order to test their suitability for systems simulations. Computational efficiencies of these algorithms will be improved, as part of the final streamlining effort.

1.0 INTRODUCTION

This progress report summarizes work performed in the third six months of a three year project, entitled "Developing and Upgrading of Solar System Thermal Energy Storage Simulation Models."

The purpose of this project is to select and upgrade solar energy storage models, incorporate these models into a computer simulation, validate the model with comparisons to measured data, and make these models available for use by engineering and architectural organizations.

The first six months of this program were concerned primarily with the gathering of information regarding available models and simulation programs. This work resulted in a technical progress report titled "Developing and Upgrading of Solar System Thermal Energy Storage Simulation Models" dated February, 1978. This report reviews and assesses the state-of-the-art in thermal energy storage.

The second six months of the program was concerned with the upgrading and validation of a rock bed model and a water tank model. Both of these models were compared with all important models in the literature and validated utilizing available test data. A technical progress report entitled "Developing and Upgrading of Solar System Thermal Energy Storage Simulation Models" dated September, 1978 reports this progress.

The present report is concerned with the progress which has occurred during the third six month period. It briefly describes the objectives of the project, a review of prior activities, and current progress including coordination activities. As a final discussion, the plans for the future are discussed, immediate plans in some detail and longer range plans in less detail.

2.0 PROJECT OBJECTIVES

The objectives of the project consist of four elements, each with an associated major work area:

- Objective 1: Collect, standardize and link existing TES models from the literature and other STO contractors.
- Objective 2: Correlate TES models with available TES component test data.
- Objective 3: Develop streamlined (analytic) versions of validated TES component models.
- Objective 4: Provide DOE and industry with highly competent user-driven solar TES simulation tool for energy modeling.

The project activities to date have been concerned with Objectives 1 and 2. The rock bed and water tank have been selected initially for analysis and detailed correlation with test data.

3.0 SUMMARY OF PRIOR ACTIVITIES

In the first year of the project, the rock bed math models, liquid tank thermal storage math models, and solar heating and cooling code status were reviewed. Following this review a rock bed model was selected, upgraded, and compared with data from existing instrumented rock beds. Figure 3-1 illustrates the rock bed model equations that have been incorporated into the model. The programming and model validity is illustrated from a mathematical point of view on Figure 4.1-1 which shows a comparison of the computer model with a closed form mathematical solution. The numerical accuracy is good. Figure 4.1-6 illustrates the performance that was obtained in the past from the rock bed at the University of Colorado. It should be noted that the agreement was not good causing a significant increase in the effort to find improved validation data.

Following the review, the liquid tank thermal storage models were compared to determine their differences and similarities. The most important physical phenomenon to be simulated is the thermocline, that is, the temperature profile in the water tank, which has an important relationship with extraction efficiency and outlet temperature/time characteristics. The TRNSYS model was found incapable of simulating the relatively high extraction efficiencies observed experimentally, even after the maximum number of segments in the model has been increased from 10 to 20.

Extensions to the basic Schumann model result in the following equations:

$$\frac{\partial T_f}{\partial x} = \frac{(NTU)_c}{L} \cdot (T_b - T_f) \quad \text{Equation 4.4}$$

$$\left[\begin{array}{l} \text{rate of change} \\ \text{of fluid temperature} \\ \text{with axial distance} \end{array} \right] \propto \left[\begin{array}{l} \text{rate of heat} \\ \text{transfer between} \\ \text{bed and fluid} \end{array} \right]$$

and,

$$\rho_b C_b (1-\epsilon) \cdot \frac{\partial T_b}{\partial t} = - \frac{\dot{m} C_f}{A} \cdot \frac{\partial T_f}{\partial x} - \frac{U \cdot P}{\lambda} \cdot (T_b - T_{env}) + k_{eff} \cdot \frac{\partial^2 T_b}{\partial x^2} \quad \text{Equation 4.5}$$

$$\left[\begin{array}{l} \text{transient energy} \\ \text{change in bed} \\ \text{material due to its} \\ \text{capacitance} \end{array} \right] = \left[\begin{array}{l} \text{energy exchanged} \\ \text{between the bed} \\ \text{and the fluid} \end{array} \right] - \left[\begin{array}{l} \text{energy lost} \\ \text{thru bed walls} \\ \text{to the environ-} \\ \text{ment} \end{array} \right] + \left[\begin{array}{l} \text{loss of bed temperature} \\ \text{stratification due to} \\ \text{"effective" axial con-} \\ \text{ductivity} \end{array} \right]$$

where Jeffreson's correction has the following form:

$$(NTU)_c = \left\{ \frac{D_p}{L} [P_e]^{-1} + \frac{(1+(Bi)_x/5)}{NTU} \cdot \beta^2 \right\}^{-1} \quad \text{Equation 4.6}$$

Figure 3-1a Rock-Bed Equations from Prior Six Months Report Dated September 1978

in which NTU may be defined as:

$$NTU = \frac{L}{(Re)(Pr)} \cdot \alpha_v \cdot (Nu); \text{ (from Equation 4.15)}$$

with the Nusselt (Nu) determined from a correlation such as that of Jury and Berbano:

$$(Nu) = \frac{h_A D_p}{k_f} = 0.64512 \cdot (Re)^{0.6716} \cdot (Pr)^{0.34685}$$

and where $(Re) = \frac{\dot{m} \cdot D_p}{A \cdot \mu_f}$ = superficial particle Reynolds number

$$(Pr) = \frac{C_f \mu_f}{k_f} = \text{fluid Prantl number (about 0.7 for air at } 50^\circ\text{C.)}$$

$$\alpha_v = \frac{6 \cdot (1-\epsilon)}{\psi \cdot D_p} = \text{bed area to volume ratio (Equation 4.14)}$$

L = bed length in axial direction

h_A = film heat transfer coefficient (on an area basis) between bed and fluid

D_p = median effective diameter of bed particles

k_f = thermal conductivity of fluid

$\dot{m}$ = fluid mass flow rate

A = bed cross sectional area

μ_f = Newtonian coefficient of viscosity for fluid

C_f = specific enthalpy of the fluid

ψ = bed particle shape factor denoting departures from spherical

Figure 3-1.b Rock-Bed Equation Notation

- ϵ = bed void fraction
 T_f = fluid temperature
 T_b = bed particle temperature
 ρ_b = bed particle density (not including voids)
 C_b = bed particle specific enthalpy
 t = time
 x = axial distance through the bed

 $\left. \vphantom{\begin{matrix} t \\ x \end{matrix}} \right\}$ independent variables

 U = overall heat transfer coefficient characterizing energy loss to the environment
 P = perimeter of rock bed such that $P \cdot L$ = area of sides of bed
 T_{env} = temperature of the environment to which bed loses energy
 k_{eff} = effective thermal conductivity of the bed in axial direction
 $(NTU)_c$ = the corrected value for NTU which accounts for dispersion and intraparticle conduction
 (Pe) = equivalent fluid phase axial dispersion Peclet number (dimensionless) with particle diameter as characteristic length parameter
 $(B_i)_x = \frac{h_A D_P}{2 \cdot k_b}$ = the interface Biot number at bed particle surface
 k_b = thermal conductivity of the bed particle

Figure 3-1.b (Continued) Rock-Bed Equation Notation

4.0 PROGRESS

4.1 ROCK BED COMPUTER MODEL PROGRESS

Several models of TES have been collected and the better ones are being upgraded. Some comparisons with test data have been performed and a few sensitivity studies have been made. We are still attempting to obtain additional detailed test data from installations, especially those with well instrumented rock-beds. We plan to have a streamlined rock-bed model developed within the next few months. A detailed but not streamlined model for a rock-bed component will be available shortly in a form compatible with TRNSYS (the transient system simulation computer program developed at the solar energy laboratory of the University of Wisconsin, see Reference (1)).

4.1.1 Model Development

Nearly a dozen models of rock-beds have been collected from the literature and other investigators. With one exception, all rock-bed models discovered thus far are based on the Schumann equations (2). That exception is due to Margolis (3) and deals not with storage or recovery of energy per se, but analyzes the gradual degradation of the stored energy due to internal redistribution (de-stratification) and losses to the surroundings. The analysis assumes a cylindrical geometry and no fluid motion (not even natural convection) and arrives at a multidimensional transient analytical solution. The Schumann equations by contrast are one dimensional and assume forced fluid motion implying either charging (adding to storage) or discharging. Those equations are coupled partial differential equations representing fluid and bed temperatures as functions of location and time. See References (4,5) for a discussion of several such models.

The principal physical objection to the Schumann equations is the assumption of plug flow, for which fluid velocity components are constant over any cross section of the bed. As will be discussed later, this assumption (very necessary for a simple one

dimensional model) is often violated. Other than using a more complex multi-dimensional model, there appears to be no method of accurately modeling beds which do not have essentially plug flow. Other objections are more easily overcome by simply adding terms for:

- heat loss to the surroundings (see References (6,1))
- effective axial conductivity (to account for gradual de-stratification (7,1))
- modified dimensionless heat transfer coefficient, NTU, to account for conductivity into the rocks and axial dispersion due to flow (8,9).

The principal numerical difficulty with the Schumann equations stems from the large discrepancy between the heat capacities of air and rock. Straightforward finite difference solutions require a very fine discretization and an inordinate amount of computer time. (See Reference (6) for an analysis of this point.) Four approaches for avoiding these numerical difficulties have been taken.

The simplest solution for avoiding the numerical difficulties alluded to is that of Hughes et al (7) which assumes air and rock at the same axial location in the bed are at the same temperature. One can then simply discard the fluid equation. This is the model incorporated in TRNSYS (1) and it is satisfactory when details of the rock-bed are not needed.

Another approach is that taken by Riaz (10) in which an equivalent axial conductivity of the bed is combined with the mass flow rate and heat transfer coefficient by a transformation of variables thereby also dropping the equation for fluid temperatures. For some boundary conditions, this allows a closed form solution involving the error function.

The third method is essentially that of Hughes (8). Several numerical schemes were employed by Hughes (8,7). However, the one of concern here and ultimately chosen for upgrading will be referred to as Hughes (8), by which is meant equation 2.3-14 on pages 43 and 44 of Reference (8). (This model may also be regarded as a generalization of Mumma and Marvin (6).) The model derivation assumes that the partial differential

equations may be decoupled. The rock temperature is assumed to be fixed for a short time span, since it is only slowly changing compared with the air temperature. In a quasi-Galerkin approach, the rock-bed is subdivided into segments and the temperature in any segment is represented by a polynomial of fixed degree (in our case, linear). Given these polynomial representations for rock-bed temperature, a closed form solution for the fluid equation can be obtained. This solution is then available for calculating the next set of rock-bed temperatures using a more traditional finite difference representation for the bed equation.

The fourth method is to utilize an implicit or stiffly stable integration scheme. Hughes (8) in equation 2.3-16 employed a fully implicit scheme in which tri-diagonal systems of equations had to be solved each time step. The so called "leap frog" method discussed in Jacob (11) was apparently employed by Balcomb (12) and Rogers (13). Although these schemes are attractive, they are not in keeping with the design philosophy of the typical simulation system such as TRNSYS (1) or SIMWEST (14) in which the details of the integration technique are left to the general solution package outside the component model. It is debatable as to whether one needs to cooperate with the simulation system especially as the internal bed temperatures are not really of interest to the rest of the system and thus are not strictly speaking "state variables". However, the model of Hughes (8) (equation 2.3-14) alluded to above is an attractive alternative and is simpler to implement than would otherwise be the case. That is, if integration schemes are contained within the model, then synchronization problems can arise between the component model and the rest of the system. Thus, for these reasons, this fourth method was discarded.

A Description of the Upgraded Rock-Bed Model

The model chosen here is essentially that of Hughes (8) described briefly above. Some minor improvements were made in the numerical procedure which are briefly outlined below:

- Heat losses were formerly only calculated for the sides of the bed. They are now also included from the plenum area at the top of the bed.

- In the quasi-Galerkin approach, the form of the polynomials representing the rock-bed temperatures have been modified by changing the intercepts. This improves stability and may slightly detract from convergence (implying the need for more space increments).
- A better correlation for heat transfer coefficients has been incorporated, viz. that of (15); however, there is still room for improvement.

More detail for the above is available in Reference (4). An outline of the calculation procedure for exit air temperature and bed temperature derivatives follows. (Integration of the derivatives occurs externally.) It is written in pseudo-code. An Algol-like notation is used with the compound symbol $:=$ representing the replacement operator. The range of For loops (similar to Fortran do loops) are terminated by endFor statements.

An Outline of the Algorithm for the Rock-Bed Model

Definitions:	T_{in}	= Temperature of air flowing into rock-bed.
	T_{env}	= Temperature of environs to which bed loses heat.
	T_j	= Current Rock-bed Temperature at each segment j .
	A	= Cross sectional area of the rock-bed.
	U	= Overall heat transfer coefficient for bed heat loss.
	N	= Number of segments into which bed is subdivided.
	NTU_c	= Dimensionless heat transfer coefficient, air to rock (7) (recalculated each time flow rate changes).

Algorithm:

. Re-store rock-bed temperatures T_j as Tr_i to aid calculation.

For $i := 1$ thru N with $j := i$ unless reversed flow exists (i.e. discharging)
in which case $j := N+1 - i$; (reversed index)

$Tr_i := T_j$

endFor

$$Tr_0 := Tin; \quad Tr_{N+1} := Tr_N$$

- Setup coefficients to be used in the calculation loop.

$$FC := (\text{fluid mass flow rate per area}) \times (\text{fluid specific heat})$$

$$E := \exp(-NTU_c / N); \quad (\text{where exp is Napierian exponential})$$

$$CAPN := (\text{rock-bed mass}) \times (\text{bed specific heat}) / N$$

$$UAN := (\text{overall bed heat loss coef.}) \times (\text{external area for sides of bed}) / N$$

$$EFCON := (\text{effective axial bed conductivity}) \times (\text{area}) / (\text{bed length} / N)$$

$$Tf_1 := Tin$$

- Compute exit air temperature and the rate of change in rock-bed temperatures (to be used externally, to predict bed temps. (T_j) at next time step).

For i := 1 thru N with j := i unless reversed flow exists (i.e. discharging)
in which case j := N+1 - i; (reversed index)

$$B := 0.125 \times (3.Tr_{i-1} + 6.Tr_i - Tr_{i+1}); \quad (\text{temp. intercept})$$

$$S := -0.5 \times (Tr_{i-1} - Tr_{i+1}); \quad (\text{temp. slope at i})$$

$$DEL := B - S / (NTU_c / N)$$

$$Tf_{i+1} := E \times (Tf_i - DEL) + S + DEL; \quad (\text{air temp. at each i})$$

$$Qenv_i := UAN \times (Tr_i - Tenv); \quad (\text{heat loss at each i})$$

$$dT_j/dTime := (1./CAPN) \times (FC \times (Tf_i - Tf_{i+1}) - Qenv_i \\ + EFCON \times (Tr_{i-1} - 2.Tr_i + Tr_{i+1}))$$

endFor

$$Tmplem := Tin$$

$$\text{If (reversed flow exists) then } Tmplem := Tf_{N+1}; \quad (\text{plenum temp.})$$

$$dT_1/dTime := dT_1/dTime - (2./CAPN) \times A \times U \times (Tmplem - Tenv); \quad (\text{plenum})$$

$$Tout := Tf_{N+1}; \quad (\text{exit air temperature})$$

end Algorithm.

The following parameters are required in the above model:

Rock-Bed description; cross sectional area, perimeter, height, median rock diameter, void ratio, effective bed conductivity, and heat loss coefficient.

Material properties (for air & rock); specific heat, conductivity and density.

Boundary conditions; initial temperature profile, and (versus time) the mass flow rate of air into the bed, its direction and temperature.

Heat Transfer Coefficient Correlations

Considerable effort has been expended in discovering the proper correlation to employ for the heat transfer correlation. Finally, the correlation of Jury & Berbano (15) was chosen as corrected by the method of Jeffreson (9). Again for more detail see References (4,5). Validation efforts thus far seem to indicate that better agreement would be obtained between measured data and predicted values if the heat transfer coefficients were even lower (by about 20%) than those obtained by the above mentioned procedure. Therefore, other than accepting the above correlation (included in the model), the user may, alternately, either provide a calculated value for heat transfer coefficient or provide another correlation of Nusselt number vs. particle Reynolds and Prantl numbers.

4.1.2 Validation

The value of a simulation model is in its use as an aid toward evaluation of system performance under many parametric combinations for which explicit data do not exist. In addition, the determination of optimal parameter combinations is much more easily performed with a simulation model than experimentally. However, the value of simulation is lost if confidence has not been established in the credibility of the predictions. Thus, some agreement with experimental data is necessary as well as established bounds on the domain of applicability.

Comparisons of data from several different sizes of rock-beds require some normalization procedure. Two parameters which are particularly useful in characterizing rock-beds are the dimensionless heat transfer coefficient (between air and rock) and

the theoretical charge or discharge time (which assumes perfect or infinite heat transfer). These parameters are defined as follows:

$$NTU = \frac{(\text{volumetric heat transfer coefficient}) \times (\text{bed volume})}{(\text{air mass flow rate}) \times (\text{air specific heat})}$$

The parameter NTU (sometimes called the number of transfer units) may be thought of as a measure of heat exchanger effectiveness. A value of several hundred implies nearly perfect heat transfer, while a value significantly less than 10 implies a poor rock-bed from the view point of thermal energy storage.

The bed time constant, TAU_c , is the time required to charge the bed if the heat transfer were infinite. It has units of time (e.g., hours) and is defined as follows:

$$TAU_c = \frac{(\text{bed storage mass}) \times (\text{bed specific heat})}{(\text{air mass flow rate}) \times (\text{air specific heat})}$$

Rather than present normalized results, the above two parameters will be given along with the results presented in hours and degrees centigrade. The one exception will be bed length which is normalized to one (with bed dimensions indicated on the plot) whenever temperature profiles versus bed length are presented. Normalized plots could then be obtained from those given, for example by dividing all times by the bed time constant.

Analytic Comparisons

Many models contain approximate solutions to differential equations involving a degree of discretization error. Often, closed form solutions exist for simple boundary conditions. Where these solutions are available, comparisons can be made to check that no gross representation errors have been made in the formulation of the more general numerical model. Additionally, an estimate of the necessary sizes of time and space increments for adequate representation can be obtained, at least for those boundary conditions available, and alternate numerical techniques can be compared for convergence. An example of such a comparison is shown in Figure 4.1-1. The

Figure 4.1-1 Comparisons of Numerical Techniques.

Departures from analytical solution /16/ at five hours are shown (after 10 time steps) using 15 bed segments and $\frac{1}{2}$ hour time step. Temperatures ranged over 30 centigrade degrees, thus when delta T above is 3.0, it represents an error of 10%.

analytical solution was taken from Carslaw and Jaeger (16) with the following boundary conditions:

- uniform initial temperature, both fluid and bed,
- step increase in fluid temperature entering the bed at time zero,
- constant flow rate and inlet temperature,
- no facility for discharging stored energy.

These comparisons were performed under the same conditions as those used in (6), (their Figures 4 through 6) except we used fifteen bed segments with a half hour time step. The bed was 4 meters in length with a time constant of 9.18 hours and the NTU used was 35.5. Note the maximum error for the upgraded Hughes model is less than one third that of (6) and agreement everywhere is within one centigrade degree. For this case the temperatures ranged over 30 centigrade degrees implying a discretization error of about 3% for the Hughes (8) model. Thus, at least for this set of conditions, the time and space discretization is adequate.

Comparison with Experimental Data

As a result of the various DOE demonstration programs, numerous data exist. However, the intention of most of these efforts has not been component model validation. In order to check assumptions made in the model, data at several points in the same plane of the rock-bed need to be available, preferably at intervals of less than $\frac{1}{2}$ hour. Data from only one such source (viz. Univ. of Wisconsin) has been obtained thus far. Other sources have been utilized but these have only had axial temperature measurements.

The University of Wisconsin's Arlington House in conjunction with a local electric utility has the ability to store off-peak energy from electric heaters in their rock-bed. It has sufficient instrumentation to check for deviations from plug flow, as there are 16 thermocouples in each of 5 planes for a total of 80 bed temperature measurements taken at 20 minute intervals. The theoretical charging time for this bed is slightly over 10% hours at the flow rate given. Figure 4.1-2 shows a comparison between model

Figure 4.1-2 Internal Bed Temperatures, Charging Test from U.W.

INTERNAL BED TEMPS. (DEG. C.) AT 5 HOURS, WITH SCALED BED LENGTH

predictions and data when the bed is about half charged, that is at five hours. Note the principal discrepancy (between data and predictions) is in the apparent velocity of the thermal wave, with the predicted velocity about 7% greater than the 13 cm./hr. observed. Much better agreement is obtained with a flow rate about 16% less than reported. These flow rates are difficult to measure accurately, but the Wisconsin staff maintains that their error bounds are more likely plus or minus 5%. Since the predicted wave velocity was a bit too high, the temperatures at five hours are also too high by slightly less than 3 centigrade degrees (within a total range of 48 degrees). A 21% smaller heat transfer coefficient would also improve the agreement slightly, and there is at least a 30% uncertainty in the heat transfer coefficient correlation. The NTU value used in the calculations was 26.557. Figure 4.1-3 shows a comparison of exit air temperatures. Again, a slightly lower flow rate would have improved the correlation. However, the 5% response time (i.e., when the exit temperature rises, in this case, 2½ degrees) is about 6.4 hours with a predicted value about 15 minutes sooner or an error of only 4%. Because of the lower value for the predicted response time, the temperature values are off, again by slightly less than 3 degrees.

Recovery of heat from a rock-bed is usually done by introducing air from the cold air return registers in the house up through the rock-bed. This cold air is essentially constant in temperature, especially if the thermostat setting remains unchanged and infiltration rates are low. Again data were obtained from the University of Wisconsin for discharging of their rock-bed over a four hour period. Better agreement (than for the previous case) is obtained between predictions and data, at least in the rock-bed temperatures. There is a negligible error in thermal wave velocity with a maximum bed temperature error of 4.75 degrees at bed exit. (See Figure 4.1-4.) However, a lower flow rate for discharging was used than for charging, so the theoretical time to fully discharge the bed was longer, viz. 18 hours. Compared to 18 hours (to fully discharge the bed), 4 hours of data may not have been sufficiently long for a meaningful check of predictive capability. The NTU value used was 31.25. The value of principal interest, however, is the exit fluid temperature versus time shown in Figure 4.1-5. Here it is apparent that some anomaly at start-up seems to have caused an asymptotic discrepancy of about 4% centigrade degrees with a maximum error of 5.5 degrees.

UNIVERSITY OF WISCONSIN CHARGING DATA VS. CALC.

EXIT AIR TEMPERATURE (DEG. C.) , TIME IN HOURS

Figure 4.1-3 Exit Air Temperatures, Charging Test from U.W.

UNIVERSITY OF WISCONSIN DISCHARGING TEST VS. CALC.

INTERNAL BED TEMPS. (DEG. C.) AT 4 HOURS, WITH SCALED BED LENGTH

Figure 4.1-5 Exit Air Temperatures, Discharging Test from U.W.
20

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EXIT AIR TEMP. (DEGREES CELSIUS) WITH TIME IN HOURS

Colorado State University has been operating several solar houses for a number of years now. Among these is an air system with a rock-bed in their solar house II. We obtained a data tape containing temperatures from several days of operation at ten minute intervals from Colorado State. Theirs is a 10.3 cubic meter bed compared with 19.11 for the University of Wisconsin. The principal difference is in the cross sectional area, as the CSU bed is a bit longer at 1.68 meters (vs. 1.57 for the University of Wisconsin). At the flow rates given, the bed time constant is 9.52 hours. The value of NTU used was 31.38. In Figure 4.1-6, these data are compared with predictions at the end of charging which was after 7-1/3 hours of operation. The large discrepancies are discussed in the next section.

4.1.3 Discussion of Discrepancies

Of the above three comparisons with experimental data, only those for discharging energy from the bed (from the University of Wisconsin) compare favorably with the model predictions and that may be due to the relatively short times available. Some reasons for the discrepancies are explored below:

Non-uniform Flow

A principal assumption of the one-dimensional models is the existence of plug flow. For better utilization of space (rock-beds are already large in comparison to other means of storing energy) plug flow should be a design goal, since without it rock-beds need to be even larger for the same effective capacity. Of course the desire for plug flow needs to be balanced with an avoidance of excessive parasitic energy requirements, viz. too much power required of the fan motor due to a large pressure drop across the bed.

The existence of several temperature probes at the same depth in the Wisconsin rock-bed make the indirect measure of uniformity of flow a simpler task. Were plug flow to exist, all probes at the same depth would record the same temperature at any one time. This was not the case. During bed charging, the standard deviation in temperature measurements exceeded 13 centigrade degrees in each of the first three levels with a maximum of 14.7 in the second level after 3-1/3 hours. At that time the measured

COLORADO STATE UNIVERSITY CHARGING DATA VS. CALC.

INTERNAL BED TEMPS. (DEG. C.) AT 7-1/3 HRS., BED LENGTH SCALED

Figure 4.1-6 Internal Bed Temperatures, Charging Data from CSU.

temperatures in the second level (at three tenths of the way into the bed) ranged from 38.3 to 84.9 degrees centigrade with a mean temperature of 57.04 degrees. Considering that the maximum temperature excursion throughout the entire eight hour charging cycle ranged only from 36.6 to 91.0 degrees, one can see that considerable channeling was occurring. The rock-bed has been modified since these data were taken to improve the uniformity of flow. For the discharge data, the maximum standard deviation was only 4.62 centigrade degrees (at start-up) with no value exceeding 4.5 thereafter. Comparisons between prediction and test are made with unweighted mean values whereby it is almost certain that more flow is occurring in some areas than others. The degree of channeling will most certainly affect the ability of a one dimensional model to predict behavior. For the discharging case, with evidence of considerably less channeling, the apparent thermal wave velocity was negligibly in error contrasted with at least a 7% error in the charging case.

The rock-bed at Colorado State University (CSU) also seems to have some non-uniform flow. As the data were only given along the bed axis, one cannot see as simply evidence of channeling. However, the staff at CSU indicated that there was evidence of a lower void ratio along the center line than out toward the sides of the bed, and thus less flow occurred down the center (where the temperatures were measured).

Air Leaks

Evidence of air leaks is difficult to quantify with the data as they were received. However, confirmation was obtained in discussions with the staff at CSU. Joint separation at the corners of the box holding the rocks had occurred. As much as ten percent of the flow was leaking out the sides of the rock-bed when these data were collected. This was confirmed by adding another flow measurement and observing different flow rates into and out of the bed. Also the lining of the box has yielded, probably allowing flow to bypass the rocks along the outer edges of the box. However, the rock-bed is located within the insulated space of the house, so any leaks are not necessarily losses but merely uncontrolled heat inputs into that space.

Heat Balances

A heat balance to check the consistency of the CSU data was performed. Over an eight hour period approximately 298,000 kilojoules of energy were input to the bed. Allowing for expected heat lost to the environment from a well sealed, well insulated rock-bed (as it was designed) and some heat flow out with the exit air stream, there was still a discrepancy of 100,000 kilojoules (slightly over one third of the total) between the expected stored energy and the actual, based on measured temperatures along the centerline. Of course the heat capacity of the rocks may be suspect, but it was re-checked by the staff at CSU and found to be correct. It is felt that due to non-uniform flow, higher temperatures than those measured may well have existed off the centerline, and thus more heat was stored than the centerline temperatures would indicate. Nonetheless, the data to which validation attempts had been attempted were sufficiently suspect to discount the apparent poor showing of the model's predictive capabilities.

Temperature Data

Charging of the rock-bed can only be adequately modeled if the temperature input to the rock-bed is known with some reliability. In the case of the University of Wisconsin charging data, the measurement of the air temperature was apparently in the streamline of an electric heating element well before sufficient mixing could occur. Again heat balances showed more energy entering the bed than could be accounted for by storage temperatures or normal heat losses. However, since the Wisconsin data were for an essentially constant charging temperature, an asymptotic approach to this temperature is expected in the bed. This apparent asymptote in rock-bed temperatures reached a value from 17 to 20 centigrade degrees lower than the inlet air stream temperatures. The staff at Wisconsin recommended that the inlet temperatures be corrected (to obtain a heat balance) by subtracting a value from them. Thus, the Wisconsin charging data comparison (as presented in Figures 4.1-2 and 4.1-3) has been modified to include a correction of $18\frac{1}{2}$ centigrade degrees to the inlet temperatures.

In Figure 4.1-5 there is a fairly rapid departure of observed values from prediction in the first half hour. It is conjectured that in this discharging test, the bed was first

charged and then sat for some time with no apparent flow. During this no flow condition the top few percent of the bed may have cooled due to heat losses to the environment. The first bed temperature measurements are ten per cent of the way into the bed. It is felt that the initial conditions were insufficiently detailed to account for the cooler part of the bed in the top few percent, which would explain the consistently lower observed exit air temperatures. A good check on this hypothesis would be data for a complete charge and discharge cycle with bed surface measurements especially during the quiescent period.

Additional discussions with the staff of CSU indicate that some of their temperature measurements are also questionable. In the process of conversion from raw data to engineering units a computer program was used which contained some sign errors. As a result some temperatures may be in error by as much as 15%. It is our understanding that corrected values are being distributed.

Comparisons Using Modified Flow Rates

If flow rates are "adjusted" such that an energy balance is obtained, one may assume that the temperatures observed correspond to some plug flow condition at that modified flow rate. Since the CSU data had a discrepancy in the heat balance of 33%, the input flow rate was reduced by this amount and the model was again used to predict the temperature profiles at this new flow rate. The comparison is shown in Figure 4.1-7. As would be expected, the correlation is considerably improved over that of Figure 4.1-6. Of course this does not constitute validation. Instead it has been shown that for consistent data the model performs reasonably well. Any air leaking out the sides of the bed is not modeled and depending upon magnitude and location, that could account for much of the remaining differences.

4.1.4 Need For Effective Rock-Beds

Concern has occasionally been expressed that the parasitic power used for rock-bed blowers needs to be reduced. A more loosely packed (higher void ratio) rock bed of larger particles would have a lower pressure drop and thus lower fan power

Figure 4.1-7 Internal Bed Temperatures with Modified CSU Charging Data.
26

INTERNAL BED TEMPS. (DEG. C.) AT 7-1/3 HRS., BED LENGTH SCALED

requirements. However, the stratification is thereby impaired and the thermal wave soon breaks through the bed. This is simply remedied by making the bed longer. All of which results in additional degradation to the amplitude of the recovered temperature wave (with respect to the charging wave) and lower quality recovered energy, which is less useful. Such a bed is also harder to model since it is less likely to have plug flow. Aside from it being more difficult to model, it may not be cost effective since a larger rock-bed will cost more and occupy additional valuable space.

Using as a design criterion a constant response time for the rock-bed, it will be shown that more compact beds are not more costly, in spite of their higher pressure drop. Response time may be defined as the time for 5% breakthrough. That is, given a steady state, constant temperature, cool bed, introduce a step increase in the inlet temperature. The time at which the exit temperature increases by 5% of this inlet change is the response time. For many solar applications a response time of between 6 and 8 hours is appropriate. For example in Figure 4.1-3 the response time is 6.4 hours or about $5/8$ of the theoretical charge time. Response time may be determined using the closed form solution of Reference (2). Many graphical representations of this solution are available in terms of various dimensionless parameters. A case is made in (5) that the most convenient form is that found in (17), in which response of outlet temperature to a step change in inlet temperature is plotted versus dimensionless time for several values of the parameter NTU, which was previously defined as a dimensionless measure of heat exchange effectiveness.

For a given response time and air mass flow rate, one can iteratively determine the required length for a rock-bed of a particular configuration (i.e., cross section, particle size, void fraction, particle shape factor, etc.). To do so, a correlation for heat transfer coefficient is needed (such as (15)) as well as the response time curves found in (17). This presumes that even low pressure drop beds have plug flow, which is not the case; therefore, the conclusions drawn here will be conservative. Unfortunately, correlations of NTU (or of heat transfer coefficient) versus pressure drop are not readily available.

Given the rock-bed configuration and length, the pressure drop may be determined using an appropriate correlation, such as (18), modified to agree with some recent test data.

Thus, one can obtain a plot of pressure drop versus rock bed volume for a fixed response time and mass flow rate by varying the configuration (e.g., particle size). The volume of a rock bed may be converted to some initial cost by assuming \$100 per square meter of external bed area, installed. This cost in turn may be converted to a yearly cost by assuming a 10% loan over 20 years. Fan power required is a linear function of pressure drop and may also be represented as a yearly cost by assuming 5¢ per kilowatt hour and design chart data which states that a ¼ HP fan at 75% efficiency can move 1500cfm air at a pressure drop of .785 inches of water. The above procedure was followed for a rock bed similar to the one in the Arlington, Wisconsin house. The yearly rock-bed costs were converted to incremental costs by subtracting some nominal rock-bed size. The results are plotted as Figure 4.1-8. The minimum of the sum of the two curves is the optimum pressure drop and is seen to lie somewhere above 60 pascals for a bed about 1.5 meters in length.

There is no claim made in the above that this is a general solution, but rather that by trading bed costs versus fan power one may discover that slightly higher pressure drops look attractive due to the increased effectiveness of the more compact denser bed. Our motivation for the above began with our difficulty at modeling rock-beds with non-uniform flow. Higher pressure drops increase uniform flow and bed effectiveness resulting in improved quality for the recovered energy. A heat transfer effective rock-bed also appears to be cost effective, in spite of the higher pressure drop. It is also easier to model.

4.1.5 Conclusions

Sometimes more detail is necessary in a rock-bed model than is currently available in simulation codes such as TRNSYS (1). (For an example of work requiring a more detailed model, see (5).) Such a computer model (described above) will soon be available and it is planned to have a comparable but more streamlined version available by mid-summer of 1979.

The effectiveness of a rock-bed is sensitive to the details of construction. Only slight carelessness in the design or installation can seriously degrade performance with regard

Figure 4.1-8 Costs for Fan Power and Effective Rock-Beds.

to the quality of recoverable energy. Better constructed rock-beds will cost more, but they are likely to be more cost effective.

Air tight, well insulated, vertical rock-beds with pressure drop adequate to induce plug flow (in excess of 40 pascals/meter of length) are good candidates for reliable modeling. We know of no detailed models for rock-beds of any other kind.

4.2 LIQUID THERMAL ENERGY STORAGE MODELS

4.2.1 Introduction

Computer models of liquid TES components in solar energy systems which have so far been considered in this project all fall into one of two categories: one-dimensional or three-dimensional models. In one-dimensional models heat transfer within the liquid is considered only in the vertical direction. This approximation is appropriate if the isotherms in the tank are horizontal. Otherwise, the three-dimensional nature of the heat transfer phenomenon cannot be ignored and only a properly designed three-dimensional model, such as the COMMIX-SA program developed at Argonne (19) can provide an accurate simulation of the behavior of a storage tank.

We have excluded the COMMIX-SA from further consideration in this project because it is believed to be far too complex and unwieldy to be used as part of a system simulation program such as TRNSYS. This is unfortunate because the three-dimensional approach is necessary whenever such storage tank characteristics are to be analyzed as the length to diameter ratio or the locations, sizes and directions of the various inlets and outlets.

There is, however, one important consideration which favors the selection of one-dimensional models (other than their computational efficiency). It is possible, at least conceptually, to create an ideal liquid storage tank whose storage efficiency is the theoretical maximum. Such an ideal storage tank would be perfectly stratified in the sense that there would be no loss of stratification as a result of heat conduction or convection currents due to turbulence and, of course, heat loss to the environment

would be nil. To represent mathematically the behavior of an ideal storage tank a one-dimensional model is sufficient.

As will be shown later, a well designed storage tank's behavior approximates quite closely that of the ideal storage tank. Thus, the information provided by a one-dimensional model of a perfectly stratified tank is very useful. It represents the upper bound on performance. For example, if in a particular solar energy system the requirement for TES is to be determined for some known pattern of load demand and insolation, then the first step is to determine the minimum capacity required with the ideal tank using a TRNSYS type simulation with the one-dimensional model. The second step is to convert the capacity requirement of the ideal tank into actual storage tank capacity of a particular tank design. This suggests that the liquid TES design problem may be partitioned into two independent sub-problems, the first one of which requires only a one-dimensional model for a time oriented system simulation. The second subproblem is to determine the storage efficiencies of each particular tank design configuration and need not be repeated for each application. This second task may be done by laboratory testing or by the use of the more complex three-dimensional model.

In this project the emphasis is on getting a component model ready for use in system simulations and, hence, the selection of a one-dimensional model is the only practical compromise available.

4.2.2 Temperature Stratification in Storage Tanks

A number of laboratory tests and analytical studies have shown that temperature stratification in storage tanks can play an important role in determining system efficiency (20,21,22). Mixing due to convection currents and turbulence caused by high inlet flow velocities tend to destroy stratification and, consequently, diminish the storage efficiency of the tank. A stratified storage tank is characterized by a thermocline or temperature profile which describes the distribution of thermal energy stored within the tank. The significance of temperature distribution is that it determines the variation in outlet temperature in the load-loop during periods of load demand. Excessive variation in outlet temperature tends to degrade the overall system

efficiency. For example, if the load is a liquid to air heat exchanger in a central space heating system, the rate of heat transfer to the air decreases as the liquid temperature in the outlet drops. There is a point below which adequate air temperature cannot be maintained by solar heating alone and an auxiliary heat source has to be activated. When this point is reached the storage tank may still have thermal energy stored, but this energy cannot be recovered at a satisfactory level of temperature. The more stratified a storage tank is the less likely that this will happen. That is why a great deal of emphasis is being placed on adequately representing temperature stratification in the computer models under consideration.

A schematic diagram of a storage tank and its interconnections with the rest of the solar energy system is shown in Figure 4.2-1. Some of the design characteristics of storage tanks which contribute to temperature stratification are discussed below:

- (1) A slim (tall) cylindrical tank in a vertical position is preferable to a horizontal tank or short (squat) tank because the smaller the cross-sectional area relative to the volume the less mixing due to turbulence, convection currents and conduction is likely to take place. Thus, the need to improve temperature stratification characteristics may conflict with the need to minimize surface area per unit of storage volume in order to reduce insulation and other material costs. However, when the choice is between horizontal or vertical position of a given tank, the latter is always preferable.
- (2) Inlets and outlets should be located near the top and the bottom as indicated in Figure 4.2-1. This accomplishes two objectives:
 - (a) The temperature of the liquid entering the collector is kept as low as possible because the liquid is drawn out of the coolest bottom layer of the storage tank.
 - (b) The hot liquid from the collector enters the tank near the top where it remains stratified. This is desirable as long as the collector outlet temperature is constant or is increasing in time. When the collector

Figure 4.2-1 Schematic Diagram of Storage Tank in Solar Heating System

temperature starts to decrease it may cause mixing in the tank, unless it is equipped with inlet manifolds of the type discussed below.

- (3) Diffusers should be used to reduce the velocity of the incoming liquid from both the load and the collector loops. A typical diffuser is a tube with many relatively small perforations which distribute the flow over a large area.
- (4) The use of special inlet manifolds which inject the incoming fluid at a level where its temperature is near the temperature of the stored fluid helps to preserve stratification. Two types of porous manifolds have been developed by Dr. Richard Loehrke at the Colorado State University (23). Both of these depend on buoyancy forces to control the level at which the incoming fluid enters the tank.
- (5) Well insulated storage tank walls which have low thermal conductivities are preferable. Otherwise, heat loss to the environment and convection currents along the inner walls tend to degrade stratification.

4.2.3 Computer Models for Storage Tanks

The most detailed computer model for the simulation of the hydraulic and thermodynamic processes which characterize liquid storage tanks is a three-dimensional model (19). This model is too complex and computationally intensive to be used as part of a comprehensive solar energy system simulation package. If it is assumed that the storage tank remains stratified with horizontal isotherms (which is a good assumption for properly designed systems), then a one-dimensional model is sufficient.

Four one-dimensional models have been tested so far:

- (1) The Duffie-Beckman model in TRNSYS (1).
- (2) The SOLSYS stratified storage model (24).
- (3) The CSU model developed at the Colorado State University (25).
- (4) A model developed at the University of Alabama (23).

The mathematical derivations of the Duffie-Beckman model and the CSU model of Loehrke and Sharp are almost identical. The CSU model has been used to obtain the results described in this report because its use was more convenient. Testing of the University of Alabama model revealed some numerical instability problems. Further development work would be required to upgrade it for production use. The SOLSYS model was found to be overly simplistic in its approach to modeling thermoclines. However, the basic concept was useful and the SOLSYS algorithm has been extended to provide a more realistic representation of temperature stratification. For these reasons the following presentation will be focused on the development and evaluation of the CSU and the Extended SOLSYS models.

4.2.3.1 CSU Model

The storage tank is divided into N isothermal constant volume segments. Liquid from the collector enters the hottest segment whose temperature is less than or equal to the collector return temperature. Similarly, fluid from the load enters the hottest segment whose temperature is less than or equal to the load return temperature. The energy balance for each segment is written as:

$$MC_p \frac{dT_i}{dt} = q_{ci} - q_{ki} - q_{ei}$$

where

- M = mass of tank segment
- C_p = specific heat of storage liquid
- T_i = tank segment temperature
- q_{ci} = convective energy transfer between segments
- q_{ki} = conductive energy transfer between segments
- q_{ei} = energy loss to environment

In a typical solar energy system with short-term storage the energy loss and conductive energy transfer terms, q_{ei} and q_{ki}, would be relatively small and the convective term, q_{ci}, would dominate the expression. The value q_{ci} depends on the relative magnitudes of the flow rates through the collector and the load, as well as on the position of the segment. There are five possible positions:

- (1) Above collector return
- (2) At collector return
- (3) Between collector and load returns
- (4) At load return
- (5) Below load return

The expression for convective energy transfer is $q_{ci} = c_p F_i$, where the term F_i is given in the table below

Segment position	Expression for F_i when $m_h > m_c$	Expression for F_i when $m_c > m_h$
1	$m_h(T_2 - T_1)$	$m_h(T_2 - T_1)$
2	$m_c T_c - m_h T_2 + (m_h - m_c) T_3$	$m_c T_c - m_h T_2 - (m_c - m_h) T_2$
3	$(m_h - m_c)(T_4 - T_3)$	$(m_c - m_h)(T_2 - T_3)$
4	$m_h T_h - m_c T_4 - (m_h - m_c) T_4$	$m_h T_h - m_c T_4 - (m_c - m_h) T_3$
5	$m_c(T_4 - T_5)$	$m_c(T_4 - T_5)$

m_c = mass flow rate in collector loop

m_h = mass flow rate in load loop

4.2.3.2 Extended SOLSYS

This model also divides the storage tank into segments; however, the segments are not constant volume isothermal segments as in the CSU model. The volume of each segment is determined by the difference between the flow rates in the collector loops and in the load loop. The temperature within each segment is assumed to be distributed

linearly. The input flow rates and temperatures are assumed to be constant within each time increment of length Δt ; at the end of each time increment a new temperature profile is computed.

This is an extension of the basic SOLSYS model which represents the stratified storage in a tank by two variable volume segments. The concepts underlying the Extended SOLSYS model are illustrated in Figure 4.2-2. First, the basic SOLSYS temperature profile is modified by adding an extra segment to represent the mixing zone between the upper and lower volumes. Next, additional segments may be added in each in time period Δt . Two typical cases are illustrated: In the first case the flow rate in the load loop exceeds the collector flow rate, $m_h > m_c$, and the temperature in the load return loop is lower than the storage tank bottom segment temperature. A new segment of volume $\Delta t(m_h - m_c)$ is added at the bottom of the tank and the original temperature profile is shifted to the left by the same amount. In the second case the collector flow rate is the larger and its temperature is somewhere between the upper and lower segment temperatures. In this case, the new segment is inserted at some intermediate point where the temperature of the incoming fluid equals the storage temperature, and the temperature profile below that point is shifted to the right.

In general, after each time step a new segment is introduced whose width and position in the temperature profile depends on the relative magnitude of the flow rates and on the temperatures of the incoming fluid. The computational algorithm is based on solving the energy balance equation in each time interval:

$$E_{in} - E_{out} = \Delta E_{stored}$$

where E_{in} = input energy from collector

$$= k \{T_c(i) - T_o(i)\}$$

E_{out} = energy output to load

$$= k \{T_o(i) - T_h(i)\}$$

ΔE_{stored} = change in energy stored

$$= k \int_0^1 \{T(x,i) - T(x,i-1)\} dx$$

k = constant of proportionality

Figure 4.2-2 Development of the Extended SOLSYS Concept

$T_c(i)$, $T_b(i)$, $T_o(i)$ and $T_h(i)$ are average temperatures in the i^{th} time period at the following locations:

- c = inlet from collector
- b = bottom outlet to collector
- o = top outlet to load
- h = inlet from load loop

$T(x,i)$ designates the temperature profile, that is, the temperature along the x-axis at the end of time period i .

The computational algorithm determines $T(x,i)$, $T_o(i)$ and $T_b(i)$ at the end of each successive time interval for a given set of input values for $T_h(i)$, $T_c(i)$ and $T(x, i-1)$.

4.2.4 Simulation of Storage Tank Performance

The most convincing method of evaluating the performance of different storage tank designs is to compare the "solar fractions" for the entire system in which the storage tanks are utilized. (Solar fraction is the ratio of the energy supplied from solar sources to the total energy requirement of the load.) However, it is outside the scope of this study to analyze an entire solar heating or cooling system.

The approach taken in this study to simulate storage tank operations is to treat the flow rates in both the collector and the load loops and the temperature in the inlets as input variables, and the outlet temperatures as the output variables. Given the state of the system (temperature profile) at the beginning of each time interval and the values of the input variables for that time interval, the corresponding values of the output variables are determined and the state of system is updated.

In order to simplify the description and interpretation of computer simulations in the following sections, it will be assumed that the heat loss from the storage tank to the environment can be ignored without noticeably affecting the results.

4.2.4.1 Steady-State Charge and Discharge Tests

The purpose of a charge test is to observe the accumulation of thermal energy in the storage tank while the load loop is inactive. Under steady-state conditions the mass flow rate through the collector is constant and the temperature in the inlet from the collector is set to some predetermined value. The evolution of the temperature profiles is monitored during the test.

Figure 4.2-3-a and 4.2-3-b show the results of a computer simulation with the CSU model for a steady-state charge test. The following input parameter values were used:

Tank Capacity, $V = 1.0 \text{ m}^3$
Fluid = water (density = 1000 kg/m^3)
Collector flow rate = 1000 kg/hr
Initial storage temperature = 10°C
Collector outlet temperature = 20°C

Temperature profiles were plotted at 0.1 hour time increments; thus, the results show the gradual replacement of the cool water in the tank with the warmer water produced by the collector. The simulation was repeated twice to illustrate the effect of using different number of segments in the CSU model. With only five segments ($N = 5$) the thermoclines are considerably less sharp than with 100 segments ($N = 100$). This suggests that N could be used as a calibration factor to approximate the thermoclines in storage tanks with different degrees of stratification.

The reverse of a charge test is the discharge test. Starting with a storage tank that is fully charged (with constant temperature throughout the tank), the collector loop is closed while a constant flow rate is maintained in the load loop. In the steady-state test the temperature in the load return inlet was maintained constant. Again, simulated temperature profiles at 0.1 hour time increments were plotted to show the gradual depletion of the storage tank (Figures 4.2-3-c and 4.2-3-d). The results were obtained with both the CSU and the Extended SOLSYS models, in order to provide a comparison of the different ways these models represent thermoclines. The slope of

Figure 4.2-3 a. Evolution of Temperature Profiles During Charge Test - Simulation with CSU Model (N = 5)

Figure 4.2-3 b. Evolution of Temperature Profiles During Charge Test - Simulation with CSU Model (N = 100)

Figure 4.2-3 c. Evolution of Temperature Profiles During Discharge Test - Simulation with CSU Model (N = 100)

Figure 4.2-3 d. Evolution of Temperature Profiles During Discharge Test - Simulation with Extended SOLSYS Model

the temperature profiles in the Extended SOLSYS output is controlled by an input parameter so that it can be adjusted to represent tanks with different degrees of stratification.

4.2.4.2 Comparison With Experimental Data

Experimental data for validation purposes have been obtained from three different sources:

- (1) Colorado State University (26)
- (2) National Bureau of Standards (27)
- (3) Illinois Institute of Technology (21)

Well instrumented and carefully controlled experiments have been conducted at CSU by Dr. Richard Loehrke to investigate various methods of enhancing temperature stratification in hot water storage tanks (22). Figure 4.2-4 shows a comparison of a temperature profile observed during a steady-state charge test with the results of simulations using the CSU model. The agreement between the test data and the simulation result with $N = 100$ is exceptionally good.

The results of steady-state discharge tests have been documented at both the National Bureau of Standards and at the Illinois Institute of Technology by Lavan and Thompson. Although these tests were not identical in terms of tank size, flow rates, temperatures, etc., comparison may be made by replotting the results on a dimensionless temperature versus dimensionless time basis. That is,

$$\Delta T = 100 (T_o(i) - T_{in}) / (T_o(1) - T_{in})$$

$$t_d = t/t_f$$

where ΔT = percent outlet temperature

$T_o(i)$ = outlet temperature in the i^{th} time period

T_{in} = inlet temperature (assumed to remain constant)

Figure 4.2-4 Comparison of Temperature Profiles Generated By CSU Model with Test Results

$T_o(1)$ = initial outlet temperature

t_d = dimensionless time

t_f = theoretical fill time (i.e., no losses and perfect stratification)

When computing the theoretical fill time for the NBS data, it was necessary to convert the actual storage tank capacity (which included about 16% of unusable space below the level of the inlets) into the equivalent capacity of an "ideal" tank. Thus, the t_f used here is 0.84 times the t_f reported in Reference (27).

The resulting plot of percent outlet temperature versus dimensionless time is shown in Figure 4.2-5, together with some simulation runs with the CSU model. The difference between the two sets of test data are quite noticeable. The curves produced by computer simulation are quite similar in shape to the experimental curves, but again, there are some notable differences. The observed curves are not as symmetrical as the calculated curves and show a slower decay of temperature rise towards the end. If the storage capacity requirements were to be determined on the basis of these results, the discrepancy between the experimental and the theoretical values would be about five percent.

4.2.4.3 Transient Test

Although the steady-state charge and discharge tests provide important information about the performance of a storage tank, they do not represent typical day-to-day operating conditions in solar energy systems. For example, the collector outlet temperature is not likely to remain constant over a long period of time and the flow rate through the load heat exchangers may cycle on and off in response to the demand.

As an example of transient operating conditions the inlet temperature from the collector and the flow rate in the load loop were assumed to be sinusoidal functions of time, as shown in Figure 4.2-6, while the collector flow rate and the load return temperature were held constant. The following input parameter values were used:

Tank capacity, $V = 1.0 \text{ m}^3$

Collector flow rate, $m_c = 140 \text{ kgs/hr}$

Figure 4.2-5. Outlet Temperature vs. Time During Discharge Test

Figure 4.2-6. Transient Simulation Input Variables

Max. collector outlet temperature = 60 C

Max. load flow rate, $m_h = 200$ kgs/hr

Load outlet temperature, $T_h = 30$ C

The storage tank was assumed to be uniformly at 30 C at the beginning of the test. The temperature profiles which developed during this simulation were quite different from those observed previously during steady-state conditions. Figure 4.2-7 shows simulation results from both the CSU model and the Extended SOLSYS model, obtained at hourly intervals during the first eight hours of the test. The variation in the outlet temperature as the flow rate in the load loop goes through a sinusoidal half cycle is displayed in Figure 4.2-8. Two of the curves represent the perfectly stratified tank; one was obtained by the CSU model with $N = 100$ and one with the Extended SOLSYS model. Again, the results of these two models are very close. A third curve was also produced with the CSU program in which only ten segments were used. The energy output, which is represented by the area under the curve, is appreciably smaller with $N = 10$. The perfect stratification case assumes that the inlet flow from the collector always enters the tank at the right level in order to minimize mixing. Partial stratification results when the collector return flow enters the tank at the top, regardless of the temperature distribution. This case is illustrated by a fourth curve produced by the Extended SOLSYS model.

The significance of these results depends on the load temperature requirement. For example, if the outlet temperature were required to remain above the 80 percent level at all times, then the fully stratified tank would provide a significantly better performance than the partially stratified tank. However, if the load requirements were quite insensitive to outlet temperature, then the partially stratified tank could be considered just as effective a storage device.

This example of transient operating conditions is, of course, only one of many different possibilities. There are numerous control strategies possible each of which may require different assumptions for the input parameters. For example, the flow rate through the collector may be modulated by a variable speed pump in order to maintain a near constant collector outlet temperature. Thus, the results of the simulations are valid only for the particular example presented and no general conclusions should be drawn.

Figure 4.2-7 Development of Temperature Profiles
During Transient Charge Test

Figure 4.2-8 Variation in Outlet Temperature to Load During Transient Simulation

4.2.4.4 Storage Capacity Requirement

The simulation results shown in Figure 4.2-8 have all been obtained with a storage tank volume of 1.0 m^3 . The question arises, what impact would increasing or decreasing storage capacity have on these results? One might expect intuitively that the larger the storage tank the better the performance of the system. But intuition has been known to be misleading, so the simulation has been repeated with different storage tank volumes.

The results are shown in Figure 4.2-9-a for the perfectly stratified tank and in Figure 4.2-9-b for the partially stratified tank. Instead of plotting outlet temperature vs. time, as in Figure 4.2-8, the results are put in a different perspective by displaying storage efficiency as a function of the minimum outlet temperature that would be acceptable at the inlet to the load heat-exchanger. Storage efficiency was defined as the ratio of the thermal energy delivered to the load before the minimum temperature was reached to the thermal energy generated by the collector. Since the minimum acceptable temperature varies from application to application, there is no unique value of storage efficiency associated with a particular tank design.

The envelope of these curves represent the possible best performance when the storage efficiency is optimized with respect to volume. A comparison of the envelopes for the perfectly and the partially stratified tanks is shown in Figure 4.2-10. It is interesting to note that the perfectly stratified tank outperforms the partially stratified tank only if the minimum acceptable outlet temperature is greater than 62%.

Clearly, storage capacity requirements are quite sensitive to the characteristics of the load as expressed by the minimum acceptable outlet temperature. It is therefore difficult to evaluate computer models of storage tanks by themselves, without considering the interactive nature of the entire solar energy system which they are a component of.

Figure 4.2-9a. Storage Efficiency of Perfectly Stratified Tank as a Function of Min. Acceptable Outlet Temperature for Various Values of Storage Volume

Figure 4.2-9b. Storage Efficiency of Partially Stratified Tank as a Function of Min. Acceptable Outlet Temperature for Various Values of Storage Volume

Figure 4.2-10 Comparison of Optimized Storage Efficiency Curves for Perfectly Stratified and Partially Stratified Tanks

4.2.4.5 Performance Bounds

The theoretical upper bound on storage efficiency has been established by the perfectly stratified tank. A theoretical lower bound can also be found by assuming a total lack of stratification, that is, a completely mixed storage tank in which the temperature of the liquid is uniformly distributed. In practice the behavior of this idealized completely mixed tank could be approximated by reversing the positions of the inlets from the load and from the collector. Then, the warmest water would enter at the bottom and the coolest water would return at the top, resulting in a continuous mixing process that would only stop when the temperature distribution became uniform.

Complete mixing may seem to be an extremely inefficient mode of operation, which should not be used in practice and, therefore, simulating it could be considered quite needless. However, it is possible that there are storage systems whose performance is even worse than that of the completely mixed tank. Whenever the inlets and outlets in a tank are located near each other, a hydraulic short circuiting effect is created and much of the tank's volume may be lost as an effective means of heat storage. Short circuiting is, however, a three-dimensional phenomenon which cannot be represented by a one-dimensional model. Thus, for the purposes of this analysis the completely mixed tank shall be considered a representative of the worst possible design configuration.

The transient simulation test was repeated with the completely mixed tank and the results were plotted on Figure 4.2-11. As may be expected, the storage efficiencies are much lower than in the previous cases. The envelope of the storage efficiency curves for the perfectly stratified and the completely mixed tanks are compared on Figure 4.2-12. The zone between the two curves represents the domain of feasible performance. The width of this zone suggests that there is a great deal to be gained by the application of stratification enhancement techniques.

Figure 4.2-12 also shows another interesting fact: Contrary to the intuitive reasoning expressed before, increasing storage volume does not necessarily result in improved performance. In fact, these results imply that for any given combination of energy demand and supply patterns, as well as load and collector operating characteristics,

Figure 4.2-11 Storage Efficiency of a Completely Mixed Tank as a Function of Min. Acceptable Outlet Temperature for Various Values of Storage Tank Volume

Figure 4.2-12 Comparison of Optimized Storage Efficiency Curves for Perfectly Stratified and Completely Mixed Storage Tanks

there is a unique optimum value for storage capacity. Any deviation from this optimal value, regardless of whether it is an increase or a decrease in volume, can degrade system performance. Thus, improper design and/or installation practices cannot always be compensated for by increased capacity.

4.2.4.6 Simulation Accuracy Versus Computer Cost

The results plotted in Figure 4.2-8 indicate that the net energy output to the load increases as the number of segments in the CSU model was changed from 10 to 100. This is due to the fact that the larger the number of segments in the model the steeper the thermocline and, consequently, the effective energy storage capacity becomes greater. Thus, the error due to model accuracy may be measured by comparing the energy outputs obtained with different numbers of segments to the energy output of the perfectly stratified tank represented by the Extended SOLSYS model, as shown in Figure 4.2-13. For example, to keep the computational error below five percent a minimum of 20 segments must be used and to get below the one percent error level, N must be 55 or more. Of course, there is a price to pay for accuracy in terms of computer costs. As an indication of that, the c.p. time required to run each simulation on the CDC Cyber 175 computer was also plotted on Figure 4.2-13. This result shows that the computer cost will increase very sharply when the number of segments is greater than 60. Based on these data, a good compromise between accuracy and cost appears to be achieved when N is set to about 40.

4.2.5 Conclusions

It has been demonstrated that the evolution of temperature profile is an important characteristic of liquid storage tank operations. The CSU model and the Extended SOLSYS model have both been shown to provide a reasonably accurate representation of temperature stratification in well designed storage tanks.

The performance of many TES systems used today may be well below the high level of performance attainable with the perfectly stratified tank. In practice, temperature stratification is not always recognized as a desirable characteristic of TES in liquids

Figure 4.2-13 The Trade-off Between Accuracy and Computer Cost

and the knowledge of what design practices contribute to stratification may be lacking. Neither of these simple, one-dimensional computer models can predict accurately the performance of tanks with imperfect stratification for which the horizontal isotherm assumption is not valid.

However, the main thrust of this project is to provide computer models of components to be used in overall system simulations. Thus, the emphasis must be on modeling the interactive nature of component behavior in the feedback control system environment of active solar energy systems. With an objective of such broad scope, we cannot afford to worry about the numerous variations of component details and their impact on system efficiency.

The one-dimensional CSU and Extended SOLSYS models are capable of predicting storage tank behavior for perfect stratification, partial stratification and complete mixing. Thus, the domain of feasible performance can be established for any particular system with known patterns of solar energy supply and load demand. The exact location of the storage efficiency of a particular storage tank design within this domain cannot be determined without employing more sophisticated and expensive analysis techniques.

5.0 PLANNED FUTURE ACTIVITY

In the next six months, work will be directed toward the further validation of the rock bed model. This validation will include a test program at the University of Wisconsin run specifically to obtain data of the quality and type needed for the validation of the model. Streamlining and incorporation of the rock bed model into the TRNSYS simulation program will be accomplished during this period.

The water tank simulation model has been developed and validated for the highly stratified class of water tanks. The next step is to validate the model for water tanks in which various degrees of mixing has occurred. Negotiations with the University of Colorado have been initiated to obtain the quality and type of data needed for this validation. The streamlining of the model and the incorporation of the model into the TRNSYS simulation program will continue.

Work on the phase change model has been postponed for the next six month period. No plans have been made for exploring these models until the rock-bed and water tank work is well in hand.

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