# I E A
## SOLAR R & D

**INTERNATIONAL ENERGY AGENCY**

solar heating and
cooling programme

## TASK III

# PERFORMANCE TESTING OF SOLAR COLLECTORS

Recommendations on Test Procedures

for the Thermal Performance of

Solar Domestic - Hot - Water Heating Systems

June 1989

TASK III

Performance testing of solar collectors

# RECOMMENDATIONS ON TEST PROCEDURES

# FOR THE THERMAL PERFORMANCE OF

# SOLAR DOMESTIC-HOT-WATER HEATING SYSTEMS

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This report has been compiled as part of the work within the IEA Solar Heating and Cooling Programme

Task III:  Performance Testing of Solar Collectors

Subtask E:  Development of a Capability to Evaluate Domestic Hot Water
System Performance using Short-Term Test Methods

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# INTRODUCTION TO THE INTERNATIONAL ENERGY AGENCY
# AND THE IEA SOLAR HEATING AND COOLING PROGRAMME

The International Energy Agency was formed in November 1974 to establish cooperation among a number of industrialized countries in the vital area of energy policy. It is an autonomous body within the framework of the Organization for Economic Cooperation and Development (OECD). Twenty-one countries are presently members, with the Commission of the European Communities also participating in the work of the IEA under a special arrangement.

One element of the IEA's programme involves cooperation in the research and development of alternative energy resources in order to reduce excessive dependence on oil. A number of new and improved energy technologies which have the potential of making significant contributions to global energy needs were identified for collaborative efforts. The IEA Committee on Energy Research and Development (CRD), supported by a small Secretariat staff, is the focus of IEA RD&D activities. Four Working Parties (in Conservation, Fossil Fuels, Renewable Energy, and Fusion) are charged with identifying new areas for cooperation and advising the CRD on policy matters in their respective technology areas.

Solar Heating and Cooling was one of the technologies selected for joint activities. During 1976-77, specific projects were identified in key areas of this field and a formal Implementing Agreement drawn up. The Agreement covers the obligations and rights of the Participants and outlines the scope of each project or "task" in annexes to the document. There are now eighteen signatories to the Agreement:

| | |
|---|---|
| Australia | Italy |
| Austria | Japan |
| Belgium | Netherlands |
| Canada | New Zealand |
| Denmark | Norway |
| Commission of the | Spain |
| European Communities | Sweden |
| Finland | Switzerland |
| Federal Republic of Germany | United Kingdom |
| Greece (withdrew in 1986) | United States |

The overall programme is managed by an Executive Committee, while the management of the individual tasks is the responsibility of the Operating Agents. The tasks of the IEA Solar Heating and Cooling Programme, their respective Operating Agents, and current status (ongoing or completed) are as follows:

Task I        Investigation of the Performance of Solar Heating and Cooling Systems - Technical University of Denmark (Completed).

Task II       Coordination of Research and Development on Solar Heating and Cooling - Solar Research Laboratory - GIRIN, Japan (Completed).

Task III      Performance Testing of Solar Collectors - University College, Cardiff, U.K. (Completed).

Task IV       Development of an Insulation Handbook and Instrument Package - U.S. Department of Energy (Completed).

| | |
|---|---|
| Task V | Use of Existing Meteorological Information for Solar Energy Application - Swedish Meteorological and Hydrological Institute (Completed). |
| Task VI | Performance of Solar Heating, Cooling, and Hot Water Systems Using Evacuated Collectors - U.S. Department of Energy (Completed). |
| Task VII | Central Solar Heating Plants with Seasonal Storage - Swedish Council for Building Research (Ongoing). |
| Task VIII | Passive and Hybrid Solar Low Energy Buildings - U.S. Department of Energy (Ongoing). |
| Task IX | Solar Radiation and Pyranometry Studies - KFA Jülich, FRG (Ongoing). |
| Task X | Solar Materials Research & Development - AIST, MITI, Japan (Ongoing). |
| Task XI | Passive Solar Commercial Buildings - Swiss Federal Office of Energy (Ongoing). |
| Task XII | Solar Building Analysis and Design Tools - U.S. Department of Energy (Ongoing). |
| Task XIII | Advanced Solar Low-Energy Buildings - Royal Ministry of Petroleum and Energy, Norway (Ongoing). |

## TASK III
## PERFORMANCE TESTING OF SOLAR COLLECTORS

The overall goal of Task III was by international cooperation to develop and validate common test procedures for rating the performance of solar thermal collectors and solar domestic hot water heating systems.

Task III was initiated in 1977 with three subtasks:

Subtask A:    Standard Test Procedures to Determine Thermal Performance

Subtask B:    Development of Reliability and Durability Test Procedures

Subtask C:    Investigation of the Potential of Solar Simulators

Upon the completion of these subtasks at the end of 1982, the Executive Committee approved an extension of the Task with the following three subtasks:

Subtask D:    Characterization of the Thermal Performance of Solar
              Collectors

Subtask E:    Development of a Capability to Evaluate Domestic Hot Water
              System Performance using Short-Term Test Methods

Subtask F:    Development of a Basis for Identifying the Performance
              Requirements and for Predicting the Service Life of Solar
              Collector System Components

At the end of 1985 a further extension was approved, with a completion date at the end of 1987.

Participants in Task III (those marked * until the end of 1985 only) were:

   Australia*, Austria*, Belgium*, Canada, Denmark, F.R.Germany, Italy,
   Japan*, the Netherlands, Spain, Sweden, Switzerland, United Kingdom,
   United States and the Commission of the European Communities.

# CONTENTS

# EXECUTIVE SUMMARY

Performance test methods for solar equipment provide manufacturers, installers and users with an objective means of measuring and comparing the output of different systems.

This report presents the results of a collaborative effort within Task III to develop performance test methods for single-dwelling solar domestic-hot-water systems. The objectives of this work were to develop the ability to predict from short-term test measurements the long-term (i.e. annual) performance of a system under specified operating conditions in an arbitrary site.

Two approaches to system testing were developed: 'component-based testing', in which the short-term measurements identify individual component parameters; and 'correlation-based testing', in which characteristic system parameters are identified from short-term measurements on the complete system. Component-based testing has the advantage that the sensitivity of the long-term performance to changes in the components can easily be investigated, while correlation-based testing takes into account differences between actual system performance and the performance predicted for an ideal coupling of the separate components.

In component-based testing, the parameters which have to be measured are determined by the model used to predict the long-term performance. The report identifies the component parameters which are commonly required and recommends methods for determining their values. Sometimes a method of calculation is suggested in preference to measurement, and in other cases well-established test methods are recommended. But for some parameters – the storage heat-loss coefficient and the parameters describing immersed heat exchangers and heat exchangers with thermosyphon flow on the secondary side – improved or entirely new test methods have been developed. The validation of models used to predict long-term performance lay outside the scope of the work of the Task, but it is emphasized that the reliability of the prediction depends on the accuracy of the model.

For correlation-based testing the Task has produced a new approach based on a daily utilizability model. The model contains four or five parameters which are identified from non-intrusive measurements in stationary operation – i.e. where there is no net carry-over of stored energy from one day to the next. This simple model takes into account the effects of heat-exchanger effectiveness, stratification in the tank and mixing during charge and discharge, the daily draw-off of heated water, and solar utilizability (including the effect of day length). The long-term performance is predicted using the same model. A recent proposal for a dynamic version of the model, which offers the possibility of *in situ* testing under variable outdoor conditions, is to be the subject of further development within the IEA Solar Heating and Cooling Programme.

The work described in this report is offered as a contribution to the development of common internationally-recognized performance test methods for solar domestic-hot-water systems.

# 1    INTRODUCTION

## 1.1    Objectives

Within the participating countries of Task III there has been much interest in
recent years in the development of short-term test methods for single-dwelling
solar domestic-hot-water (SDHW) systems. A summary of a number of national
proposals for SDHW system test methods was given in an earlier Task III
technical report [1].

Subtask E of the Task was set up with the aim of collaborating on the
development of common test methods. The intended methodology was to identify
system performance parameters which could be determined by short-term test
measurements, and which could be used to estimate the long-term (i.e. annual)
performance of the system for an arbitrary site.

## 1.2    Approaches to SDHW system testing

In the review of national approaches to SDHW system testing [1], the methods
described were classified into five different types of approach. Of these, two
types - the so-called 'simulation methods' and 'system-identification methods'
- both appeared to provide a means of achieving the required objectives.

In the simulation methods the parameters to be identified by short-term
testing are individual component parameters, such as the heat-loss coefficient
of the storage tank or the effectiveness of a heat exchanger. The long-term
performance of the system for a given site and operating conditions is
generally predicted using detailed computer simulations. However, the
long-term prediction could equally well be made using a simplified model
(provided its parameters can be expressed in terms of the component
parameters), and since it is the type of short-term testing that really
distinguishes these methods, the approach was generally referred to by the
Task participants as *component-based testing*.

In system-identification methods the short-term measurements are made on the
complete system. The short-term performance of the system is assumed to follow
a simple equation or 'correlation' containing system-dependent parameters. The
values of these system parameters are determined for the system under test by
best fit of the correlation to the short-term measurements. The annual
performance of the system is predicted from the measured parameters using a
suitable long-term performance correlation or other valid method of
calculation. Reflecting the importance of short-term correlations in these
methods, the approach was generally referred to by the Task participants as
*correlation-based testing*.

Since each approach had its own distinct advantages, the participants decided
to develop test methods of both types in parallel. Accordingly, Part I of this
report describes the methods developed for component-based testing, and
Part II the work done by the Task on correlation-based testing.

## 1.3   System inspection procedure

A system test is sometimes required to fulfil the additional role of providing a check that a system is operating according to its design. However, many potential faults can be identified without a possible expensive measurement of the system thermal performance. The Task participants have therefore compiled a system inspection procedure [2] to complement the performance test methods presented in this report. The procedure contains checks on the system configuration and integrity, and diagnostic tests to uncover faults in the sensors and controllers.

## 1.4   References

1       "Summary of National Approaches to Short-Term SDHW Systems Testing"
        B D Wood and B A Rogers (Eds)
        Task III Technical Report No T.3.E.1
        December 1987
                Available from:    Dr B D Wood
                                   Center for Energy Systems Research
                                   Arizona State University
                                   Tempe, AZ 85287
                                   USA

2       "Inspection Procedure for Solar Domestic-Hot-Water Systems"
        U Frei, J Keller and R Brunner
        Task III Technical Report No T.3.E.2
        To be published
                Distribution:      Swiss Federal Office of Energy
                                   P.O. Box
                                   CH-3003 Berne
                                   Switzerland

# PART I: COMPONENT-BASED TESTING

## 2.1    Basis of approach

In component-based SDHW system testing, the long-term performance prediction
is made using a computer simulation model for which the input data are either
the individual component parameters or system parameters expressed in terms of
the component parameters. Normally, a detailed simulation model would be used
in this approach (§2.2), but a simplified model could also be used (§2.3)
provided it was valid for the system of interest and of sufficient accuracy.

Many different system models may be used to predict annual performance, and
some of these models are discussed in the following sections. However,
Task III has made no comparison or assessment of these models and makes no
recommendation about the choice of model. It is assumed that any system model
adopted for component-based testing has already been adequately validated. The
main objective of the Task has been to contribute to the development of simple
and reliable procedures for measuring the component parameters.

It is tempting when developing component tests to look for detailed physical
models of the complex processes taking place in real components. A full
thermal analysis of the behaviour of a stratified store or heat exchanger, for
example, may in fact be very complicated. In component-based system testing,
however, the level of detail which can be taken into account is limited by the
way in which the component is represented in the system model. If the
component is characterized by a single constant parameter, for example, then
there may be little point in identifying a multi-parameter model of its
performance. Thus it is important to first identify the exact input parameters
which the simulation model requires.

It can justifiably be argued that no procedure can be regarded as a system
test unless it includes measurements on the assembled system, since the
performance of a real system may be significantly less than what would be
predicted from a simulation. For example, if the thermal losses of piping are
calculated rather than measured in situ, then they may not take into account
poor installation of the thermal insulation or the extra losses from
uninsulated valves. To overcome this problem, it is recommended that an
inspection procedure such as [1] is applied to the installed system before it
is operated. Also, measurements of component parameters should be made with
the components in as near to operating conditions as possible.

To validate a component-based test for a given system it would be necessary to
show that the simulation model with the test values of the component
parameters as input data could predict the annual performance of the system
for a range of yearly weather conditions. Although this is not practicable as
a part of routine testing, the approach has been well validated by experience
at the Danish Solar Energy Testing Laboratory [2], the Institute of Applied
Physics at Delft [3], in Switzerland (See Chapter 11, below), and elsewhere.

## 2.2  Detailed simulation models

The following is a brief summary of the features of simulation models used by the Task participants for long-term performance prediction:

BSOL - modular; single collector node (HWB with quadratic temperature dependence and collector heat capacity); internal heat exchanger only; thermosyphon mode; 1-10 tank nodes [4,5]

EMGP2 - modular; arbitrary number of collector nodes; collector heat capacity; internal and external heat exchangers; thermosyphon mode; unlimited number of tank nodes [6]

EURSOL - fixed model, derived from EMGP2, describing 8 different SDHW systems [7]

INTASOL C - not modular; HWB collector model; external heat exchanger only; no thermosyphon mode; fully-mixed tank only [8]

SEU II - modular; multiple collector nodes (HWB with collector heat capacity); internal or external heat exchanger; no thermosyphon mode; 1-20 tank nodes [9]

SIWW - four separate programs (single tank with analytic or iterative treatment of internal heat exchanger, double tank with iterative treatment of internal heat exchanger, and single tank with detailed model of thermosyphon operation); further programs under development should permit the simulation of some 10 generic SDHW and SH systems [10]

TPDZB - modular; collector model HWB with collector heat capacity or Klein formula; no thermosyphon mode; 1-10 tank nodes; internal or external heat exchanger [11]

TRNSYS - modular; no collector heat capacity; external heat exchanger only; thermosyphon mode (for direct systems with thermosyphon flow in the primary loop); practically unlimited number of tank nodes; allows for modifications and user-written modules [12]

WATSUN - six or seven generic models (not modular); enhanced (5-parameter) collector models; hour-by-hour energy balance (not fully transient); SYPHON - a derivative of WATSUN is used for thermosyphon simulation (for direct systems with thermosyphon flow in the primary loop) [13]

## 2.3  Simplified simulation models

The following simplified models are also used for long-term prediction of solar water-heating systems:

f-chart [14,15]

$\bar{\phi}$,f-chart [16]

ICS program - uses daily test results from the SRCC test [17]

INTASOL E [18]

KENN [19]

SEU design method [20]

Task VI method [21]

## 2.4 References

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2   "Performance Testing of Domestic Hotwater and Space Heating Solar
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    J E Nielsen and O Ravn
    Danish Solar Energy Testing Laboratory
    Department for Energy Technology
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    PO Box 141
    DK-2630 Tåstrup
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3   "Results from component-based, indoor and outdoor testing on one solar
    domestic hotwater system"
    E M Keizer-Boogh and A C de Geus
    TNO Institute of Applied Physics
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    Postbus 155
    2600 AD Delft
    Netherlands
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4   "Dokumentation af BSOL"
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W L Dutré and A Debosscher
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Afdeling Toegepaste Mechanica en Energieconversie
Laboratorium voor Warmteoverdracht en Reaktorkunde
Celestijnenlaan 300 A
B-3030 Heverlee, Belgium
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8     "INTASOL C - A computer program to obtain the annual dynamic behaviour of a solar system hourly calculated"
E Mezquida, J L García and A L González
Departamento de Motopropulsión y Energía
Instituto Nacional de Tecnica Aeroespacial
Torrejón de Ardoz
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9     "User Manual for SEU II Hourly Simulation Program"
M C Tough
Departmental Report 1288 (SEU No. 525)
Department of Mechanical Engineering & Energy Studies
University College
Newport Road
Cardiff UK  CF2 1TA
January 1986

10    "Simulation programs SIWW for domestic hot water systems"
M Zogg, M Rieder and R Hungerbühler
Ingenieurschule Burgdorf
Höhere Technische Lehranstalt des Staates Bern (HTL)
Pestalozzistr. 20
CH-3400 Burgdorf
Switzerland

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      TPD-TNO-TH Report No 303.220-II (In Dutch)
      Institute of Applied Physics, TNO-TH
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      Netherlands
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12    "TRNSYS: A transient system simulation program user's manual"
      S H Klein et al.
      Engineering Experiment Station Report No 38
      Solar Energy Laboratory
      University of Wisconsin-Madison
      1500 Johnson Drive
      Madison, Wisconsin 53706, USA
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13    "WATSUN: A simulation program for active solar heating systems"
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      University of Waterloo
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14    "Solar Heating Design by the f-Chart Method"
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      University of Wisconsin-Madison
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      S A Klein and W A Beckman
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      Solar Energy **32**, 687-705 (1984).

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      K Schreitmüller
      Report Number IEA-SHAC-TVI-7
      DFVLR
      Pfaffenwaldring 38-40
      D-7000 Stuttgart
      Federal Republic of Germany
      May 1986

# 3    SOLAR COLLECTOR PARAMETERS

## 3.1    Test methods

The characterization and testing of the thermal performance of a range of
different types of solar collectors is the subject of a separate Task III
technical report [1], to which the reader is referred for a full
discussion.

Most simulation models for solar water-heating systems assume the well-known
Hottell-Whillier-Bliss equation for conventional collectors. To account for
the change in optical efficiency with the angle of incidence of solar
radiation, a semi-empirical or empirical incidence-angle modifier is usually
included. These parameters can be determined using standard collector test
procedures.

Some models also account for dynamic effects by including an effective
collector heat-capacity parameter. Methods of estimating this parameter are
discussed in [2].

It is important to remember that the collector parameters which incorporate a
flow factor will generally need to be adjusted to account for the difference
between the mass flowrate of the heat-transfer fluid in operation and the
flowrate used for testing.

## 3.2    References

1       "The Characterization and Testing of Solar Collector Thermal
        Performance"
        B A Rogers, S J Harrison, H Soltau and B D Wood
        Task III Technical Report No SEU-IEA-TR2
        To be published
                Information available from:    Solar Energy Unit
                                               School of Engineering
                                               Division of Mechanical Engineering
                                                   and Energy Studies
                                               University of Wales
                                               P.O. Box 917
                                               Cardiff UK  CF2 1XH

2       "Solar Collectors - Test Methods and Guidelines"
        W B Gillett and J E Moon
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# 4    STORAGE TANK HEAT CAPACITY

## 4.1    Introduction

The heat capacity of a storage tank is ratio of the heat added to (or lost from) the tank to its resulting increase (or decrease) in temperature. For phase-change storage materials, where supercooling may occur, the heat transferred to a storage device during charge and the heat extracted during discharge may differ. In such cases it becomes necessary to define separate heat capacities for charge and discharge, the values of which will depend significantly on the temperature and flowrate of the heat-transfer fluid. In the present document, however, the scope is limited to sensible-heat storage, and it is therefore sufficient to regard the heat capacity of a store as a well-defined single-valued parameter.

For storage of sensible heat a theoretical heat capacity can be calculated by adding the products of the mass and specific heat capacity of all components of the storage device inside the insulation. A simpler estimate is obtained as just the product $M_s c_p$ of the mass of water in the tank and the specific heat capacity of water. It may be that the simulation model calculates this value automatically, and only requires as input the volume or mass of the fluid content.

The methods of measuring the storage-tank heat capacity usually assume that the product $(UA)_s$ of the heat-loss coefficient and the area of the store are already known. For this purpose either the stand-by value of the heat-loss coefficient can be used (cf. §5.1.1, below), since it can be measured independently of the heat capacity, or the dynamic heat-loss coefficient can be measured assuming a theoretical heat capacity. Alternatively, the dynamic heat-loss coefficient and heat capacity can be determined simultaneously from tests that assume independent equations in $(UA)_s$ and $M_s c_p$ (cf. §5.3.5, below).

## 4.2    Methods of measurement

Methods of measuring the charge and discharge heat capacities of a sensible-heat storage tank are given in §8.5.4 of ANSI/ASHRAE Standard 94.3-1986 [1]. The equations on which the estimates are based are implicit integral equations in the heat capacity, so a partial substitution of the theoretical heat capacity is used to give the equations an explicit form. The charge and discharge heat capacities are compared with the theoretical heat capacity (or 'theoretical storage capacitance'), and the Standard specifies that if the discrepancy between the largest and smallest of these three values is greater than 5%, further testing shall be halted until the discrepancy is resolved.

A method of measuring the energy stored in a sensible-heat storage tank (referred to as the 'storage capacity') is given in §7.2 of the CEC SSTG Recommendation [2]. The method is similar to the method for measuring the heat capacity (or 'heat storage capacity') given in §B.4.1.3 of the test procedures of the Danish Solar Energy Testing Laboratory [3]. In both of these methods the stored energy is calculated as the difference between the energy supplied to the store and the thermal losses from the store. The energy supplied is calculated directly from the mass flowrate of the heat-transfer fluid and the

difference between the inlet and outlet fluid temperatures. To estimate the rate of heat loss from the store at any time the CEC method uses the logarithmic mean of the inlet and outlet fluid temperatures, while the Danish method uses measurements from an array of immersed thermal probes. As with the ASHRAE Standard, the CEC procedure requires a theoretical storage capacity to be calculated for comparison with the measured value.

An approximate method for determining the heat capacity of a storage tank simultaneously with its heat-loss coefficient is outlined in §5.3.5 below.

## 4.3    Discussion and recommendations

It is questionable whether there is a need to measure the heat capacity of a sensible-heat store rather than use a theoretical value. Test procedures usually require the theoretical value to be determined, and the measured value is expected to be close to the theoretical value. In published test procedures the theoretical value is used in the determination of the empirical values for the heat capacity and the dynamic heat-loss coefficient. Also, although there may be a significant difference between the calculated and estimated capacities (when there are 'dead spaces' in the tank, for example), simulation models do not generally distinguish the heat capacity of the store from the fluid heat capacity $M_s c_p$. Where an empirical value for the heat capacity is required, however, any of the published methods [1-3], or the method described in §5.3.5 of this report, may be suitable.

## 4.4    References

1       ANSI/ASHRAE Standard 94.3-1986
        "Methods of Testing Active Sensible Thermal Energy Storage Devices based on Thermal Performance"
        The American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc.
        1791 Tullie Circle NE, Atlanta, GA 30329

2       "Recommendations for European Solar Storage Test Methods (Sensible and Latent Heat Storage Devices)"
        Ir. E. van Galen
        Technisch Physische Dienst TNO-TH
        PO Box 155
        2600 AD Delft
        Netherlands
        March 1984

3       "Performance Testing of Domestic Hotwater and Space Heating Solar Systems"
        J E Nielsen and O Ravn
        Danish Solar Energy Testing Laboratory
        Department for Energy Technology
        Technological Institute
        PO Box 141
        DK-2630 Tåstrup
        Denmark
        November 1985

# 5    STORAGE TANK LOSSES

## 5.1    Introduction

### 5.1.1 Stand-by and dynamic storage losses

The thermal losses from a storage tank will depend to some extent on the flow
conditions through the tank. When the tank is stagnating, for example, the
losses will be governed by natural convection within the tank and its upper
piping connections. If fluid is being circulated through the tank, on the
other hand, the losses will be enhanced by forced convection to an extent
which, for a given tank and connection configuration, depends on the flowrate.
A distinction is therefore made between "stand-by" losses, which occur when
there is no flow into or out of the tank, and "dynamic" losses, which occur
when there is flow through the tank. (Other terms may be used to distinguish
these conditions, but the terms adopted here are the ones commonly used by the
Task III participants.)

It may be that in the laboratory experimental conditions can be found for
which the dynamic losses are significantly different from the stand-by losses.
But at the flowrates normally associated with the charge and discharge of
solar storage tanks the dynamic losses are not expected to be very different
from those under natural convection. In that case, whether the losses are
measured in dynamic or stand-by conditions can be chosen according to
convenience and accuracy. In the following sections, therefore, methods are
given for measuring both dynamic and stand-by losses, and in each case an
analysis of errors is included to enable the accuracy of the method to be
determined for a specific instrumentation and range of experimental variables.

### 5.1.2 Test conditions

Heat losses from a storage tank do not occur only by conduction through the
thermal insulation; they may also occur by air convection, either between the
tank and the tank insulation or through gaps at joints in the insulation, or
by natural convection of the storage liquid between the tank and a colder pipe
sited above or level with the tank [1]. Natural convective flow within a cold
pipe connected to a warmer tank has been directly observed using coloured dyes
[2]. Heat-loss enhancement due to these phenomena by a factor of 2 to 5 has
been reported [3,4]. It is therefore important to measure the storage losses
under realistic conditions, i.e.

-    with the pipes connected to the top and sides of the storage vessel in
     the same configuration as in the actual system, and

-    avoiding air gaps between different sections of the insulation,
     particularly at joints between the piping and the store
     (although this may be less critical at the bottom of the store).

## 5.2 Measurements of dynamic heat-loss coefficient

### 5.2.1 Sources

Essentially similar approaches are referred to as the "steady-state method" in §A3.1 of [5], the "method for determining the heat loss rate at finite flow rate" in §7.3 of [6], and the "measurement of the heat loss coefficient of the storage with fluid circulating in the collector loop" in §B.4.1.1 of [7].

### 5.2.2 Basis of method

$$T_1$$
$$\dot{m}c_p \rightarrow$$
$$T_s^*$$
$$T_a^s$$
$$\leftarrow$$
$$T_o^*$$
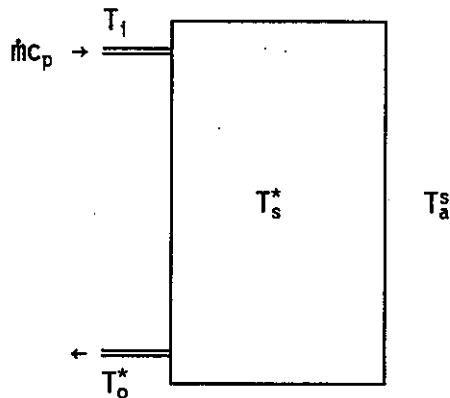
(Note: * denotes *stationary* value)

The store is maintained in a stationary state with the mean temperature in excess of the ambient temperature of the store, and the rate of heat loss

(5.1)     $\dot{Q}_{loss}^* = (UA)_s (T_s^* - T_a^s)$

balanced by the rate of heat gain from the circulating heat transfer fluid

(5.2)     $\dot{Q}_u^* = (\dot{m}c_p)(T_1 - T_o^*)$ .

The value of the heat loss factor is then determined as

(5.3)     $(UA)_s = \dot{Q}_u^* / (T_s^* - T_a^s)$ .

### 5.2.3 Accuracy of method

The accuracy of the evaluation of $(UA)_s$ from Equation (5.3) depends on the accuracy of estimating $T_s^*$, on the validity of the assumption of stationarity, and on the errors of measurement. We consider the effects of each of these factors independently.

*Inaccuracy in estimating mean store temperature* $T_s^*$ is not measured directly, but inferred from other measurements. Hence, according to the assumptions used in the estimation, there will be systematic errors in the estimated value of $T_s^*$. The corresponding relative systematic error in $(UA)_s$ is estimated as

(5.4)        $\Delta(UA)_s/(UA)_s = -\Delta(T_s^*-T_a^s)/(T_s^*-T_a^s)$ .

The mean store temperature $T_s^*$ may be estimated either from the storage inlet and outlet temperatures $T_i$ and $T_o^*$ (as in §7.3.6.4 of [6]), or from internal measurements using an array of submerged temperature sensors (as in §B.2 of [7]).

To estimate $T_s^*$ from $T_i$ and $T_o^*$, the theory of a single-flow heat exchanger in a uniform-temperature heat bath may be assumed, with the store regarded as the heat exchanger and the ambient air as the heat bath. The mean temperature in the store, $T_s^*$, is then calculated as

(5.5)        $T_s^* = T_a^s + (T_i-T_o^*)/\ln[(T_i-T_a^s)/(T_o^*-T_a^s)]$ .

Inequalities for the natural logarithmic function (see, for example, §4.1 of [8]) show that this expression gives a value for $T_s^*$ satisfying

(5.6)        $T_o^* < T_s^* < (T_i+T_o^*)/2$ .

Another simpler assumption for $T_s^*$ is

(5.7)        $T_s^* = (T_i+T_o^*)/2$ .

The accuracy in either case is seen to be of order $(T_i-T_o^*)/2$. For accuracy in estimating $T_s^*$ it follows that we require

(5.8)        $(T_i-T_o^*)/2 \ll T_o^*-T_a^s$ .

Hence this method is useful for estimating storage temperatures much greater than the ambient temperature.

The accuracy of estimating $T_s^*$ may be improved by the use of an array of temperature sensors installed within the tank. The error in calculating the mean temperature from the values at the discrete positions of the sensors is identical to the error of the corresponding numerical integration scheme (see, for example, §25.4 of [8]). In principle this error may be reduced to any desired value by increasing the number of sensors, but in practice the use of internal tank measurements makes routine testing difficult.

*Non-stationarity* Suppose that the store temperature has not reached its stationary value $T_s^*$, but some near-stationary value $T_s$, satisfying

(5.9)        $M_s c_p d(T_s-T_a^s)/dt + (UA)_s(T_s-T_a^s) = \dot{Q}_u$ .

While the true value of $(UA)_s$ satisfies (5.3), the calculated value is

(5.10)        $(UA)_{s,calc} = \dot{Q}_u/(T_s-T_a^s)$ .

The fractional error in estimating $(UA)_s$ is thus

(5.11)        $\Delta(UA)_s/(UA)_s = 1/[\dot{Q}_u/(M_s c_p d(T_s-T_a^s)/dt) - 1]$ .

The error due to non-stationarity can therefore be limited by imposing an upper limit on the rate of change of $T_s-T_a^s$ .

*Random measurement errors* Random errors are treated differently from systematic errors in the same variables and must be considered separately. The relative random error in the estimate of $(UA)_s$ from Equation (5.3) is

(5.12)     $\sigma(UA)_s/(UA)_s$

$$= [\sigma^2(\dot{Q}_u^*)/(\dot{Q}_u^*)^2 + \sigma^2(T_s^*-T_a^s)/(T_s^*-T_a^s)^2]^{\frac{1}{2}} \ .$$

where $\sigma^2(\dot{Q}_u^*)$ and $\sigma^2(T_s^*-T_a^s)$ are the variances in the measurements of the variables $\dot{Q}_u^*$ and $(T_s^*-T_a^s)$.

The random errors of $(T_s^*-T_a^s)$ depend on how it is estimated from directly-measured variables. With $\dot{Q}_u^*$ calculated from Equation (5.2), using measured values of $\dot{m}$, $c_p$ and $(T_i-T_o^*)$,

(5.13)     $\sigma^2(\dot{Q}_u^*)/(\dot{Q}_u^*)^2$

$$= \sigma^2(\dot{m})/(\dot{m})^2 + \sigma^2(c_p)/(c_p)^2 + \sigma^2(T_i-T_o^*)/(T_i-T_o^*)^2 \ .$$

The appropriate relative or absolute values of the random errors of measurement can be substituted in this equation. Normally for measurements of mass flowrate and specific heat capacity the relative errors are fixed, while for temperature measurements the absolute error is fixed. Hence the accuracy of estimation is best for large temperature differences.

## 5.3     Measurements of stand-by heat-loss coefficient

### 5.3.1 Introduction

To measure the stand-by heat-loss coefficient of a storage tank the tank is brought to some initial high temperature, it is then allowed to cool naturally under stagnation, and then is rapidly reheated. Losses during reheating are assumed negligible, so the value obtained for the loss coefficient is the value appropriate to the cooling period. Variations of the method differ in how the initial and final states of the tank are defined.

### 5.3.2 Sources

The method is referred to as the "charge-standby-recharge method" in §A.3.3 of [5], where it is the recommended heat-loss test, and as the "method for determining the relative heat loss during a stand-by period (zero flow rate)" in §7.4 of [6]. In these procedures the initial condition of the tank is steady state (as defined in the method for measuring the dynamic heat-loss coefficient), and after reheating the tank is returned to the same high-temperature state.

In §B.4.1.2 of [7], essentially the same method is described as "measuring of heat loss coefficient of storage tank during cooling period". Here, the initial and final states are fully mixed (which is achieved by rapidly circulating the fluid through the tank), and the tank is not reheated to the starting temperature.

### 5.3.3 Basis of method

The store is charged to an initial mean temperature $T_s^i$, and then allowed to cool naturally in a constant ambient temperature $T_a^s$. It is recommended that the period of cooling $\Delta t$ is sufficiently long that stratification will become fully developed; then the measured loss coefficient will be automatically weighted to account for any local dependence of thermal losses.

During cooling the store temperature obeys the equation

$$(5.14) \qquad M_s c_p d(T_s - T_a^s)/dt + (UA)_s (T_s - T_a^s) = 0 \; ,$$

and after the time interval $\Delta t$ the mean store temperature reaches a value $T_s^c$ given by

$$(5.15) \qquad M_s c_p (T_s^c - T_a^s) = \exp[-(UA)_s \Delta t / M_s c_p] \, M_s c_p (T_s^i - T_a^s) \; .$$

$T_s^c$ is assumed to be not directly measurable because of the stratification in the store.

The store is now charged to a final mean temperature $T_s^f$ (which may or may not be the same as $T_s^i$). If it is assumed that the losses during recharge are a negligible fraction of the energy $Q_u$ supplied during recharge, then

$$(5.16) \qquad M_s c_p (T_s^f - T_s^c) = Q_u \; .$$

Eliminating $T_s^c$ between the last two equations gives

$$(5.17) \qquad (UA)_s = -(M_s c_p / \Delta t) \ln[(Q_s^f - Q_u)/Q_s^i] \; ,$$

where

$$(5.18) \qquad Q_s^f = M_s c_p (T_s^f - T_a^s) \; ,$$

and

$$(5.19) \qquad Q_s^i = M_s c_p (T_s^i - T_a^s) \; .$$

The heat capacity $(M_s c_p)$ of the store is assumed known, but it may be estimated simultaneously using additional measurements, as indicated in §5.3.5.


### 5.3.4 Accuracy of method

*Inaccuracy in estimating mean store temperature* There are systematic errors in the estimation of the mean storage temperatures $T_s^i$ and $T_s^f$, and these can be treated in the same way as described in §5.2.3.

*Assumption of no losses during recharge* A second source of systematic error is the assumption of no thermal losses during the recharge period. This error is represented by an additional term

$$-(UA)_s (T_s^m - T_a^s)\delta t,$$

on the right-hand side of Equation (5.16), where $T_s^m$ is the mean storage temperature during the recharge period $\delta t$. The resulting error in $(UA)_s$ is

$$(5.20) \qquad \Delta(UA)_s = - (M_s c_p / \Delta t) \ln[(Q_s^f - Q_u - (UA)_s (T_s^m - T_a^s)\delta t)/(Q_s^f - Q_u)] \; .$$

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Since $T_s^m < T_s^f$ , it follows that a sufficient condition for

$$\left| \Delta(UA)_s / (UA)_s \right| < r$$

is

(5.21)    $\delta t < (M_s c_p / (UA)_s)(Q_s^i / Q_s^f) x (1 - x^r)$ ,

where

(5.22)    $x \equiv (Q_s^f - Q_u)/Q_s^i = (T_s^c - T_a^s)/(T_s^i - T_a^s)$ .

In terms of x, Equation (5.17) can be rewritten as

(5.23)    $(UA)_s = -(M_s c_p / \Delta t) \ln(x)$ .

Now, for given $Q_s^i$ and $Q_s^f$, the right-hand side of Equation (5.21) is maximised for

(5.24)    $x = (1+r)^{-1/r}$ ,

and with this value for x, the condition for the time of recharge becomes

(5.25)    $\delta t < (M_s c_p / (UA)_s)(Q_s^i / Q_s^f) r (1+r)^{-(1/r)-1}$ .

The right-hand side is further increased when $Q_s^f$ is as small as possible, that is when the recharge is just sufficient to fully mix the store.
[The limiting value of $Q_s^f$ occurs for the case when $T_s^f = T_s^c$, $Q_u = 0$, and $x = Q_s^f / Q_s^i$. The condition on $\delta t$ then becomes: $\delta t < (M_s c_p / (UA)_s) r / (1+r)$ .]

From Equation (5.23), we see that the choice of the value for x given by Equation (5.24) also implies a condition on the time $\Delta t$ of discharge. For small r the value of x given by Equation (5.24) satisfies $x \approx 1/e$, and hence the condition is

(5.26)    $\Delta t \approx M_s c_p / (UA)_s$ ,

i.e. that the discharge time should be approximately equal to the time constant of the store. This can be arranged using prior estimates of the time constant or by using an immersed temperature probe to indicate when

(5.27)    $T_s^c - T_a^s \approx (T_s^i - T_a^s)/e$ .

*Random measurement errors*  For a given value of $M_s c_p / \Delta t$, and assuming the errors in $Q_s^f$, $Q_u$, and $Q_s^i$ are independent, the theory of the propagation of errors [9] gives as an estimate of the variance of $(UA)_s$:

(5.28)    $\sigma^2 (UA)_s = (M_s c_p / \Delta t)^2 [\{\sigma^2 (Q_s^f) + \sigma^2 (Q_u)\}/(Q_s^f - Q_u)^2 + \sigma^2 (Q_s^i)/(Q_s^i)^2]$ .

To reduce the random errors, it follows that the values of both $T_s^c - T_a^s$ and $T_s^i - T_a^s$ should be large compared with the errors of temperature measurements. Since, for a given value of $T_s^i - T_a^s$, $T_s^c - T_a^s$ is determined by the value of $\Delta t$, this implies choosing a sufficiently large value for $T_s^i - T_a^s$.

## 5.3.5 Estimating $M_s c_p$

The heat capacity of the store can be estimated by repeating the test with another appropriate starting value for $T_s^i$, and equating the right-hand sides of Equation (5.17) given by each set of measurements. If, for example, the same value used for $\Delta t$ in each case, then the estimate is given by

$$(5.29) \quad M_s c_p = \{[Q_u/(T_s^i - T_s^a)]_2 - [Q_u/(T_s^i - T_s^a)]_1\}$$

$$/\{[(T_s^f - T_s^a)/(T_s^i - T_s^a)]_2 - [(T_s^f - T_s^a)/(T_s^i - T_s^a)]_1\} \ .$$

Unfortunately, for reliable estimates of $M_s c_p$ large values of $Q_u$ are required, which is contrary to the requirements for small systematic errors in $(UA)_s$.

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7     "Performance Testing of Domestic Hotwater and Space Heating Solar
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# 6    STORAGE TANK STRATIFICATION

## 6.1    Introduction

Whether a storage tank is fully mixed or there is significant thermal stratification within the tank depends on a number of factors, the most important of which is the rate of flow through the tank during the charging and discharging of the tank.

Stratification is accounted for in most detailed system simulation models by dividing the tank into a number of layers, or 'nodes', each of which is uniform in temperature. The mixing effects caused by the injection or draw-off of fluid or by convection are then modelled by the transfer of fluid and heat between neighbouring nodes after each time step in the simulation. (Hence the mass of fluid in each node should not be less than the product of the maximum mass flowrate through the store and the time step for the simulation.)

Usually, the number of nodes has to be supplied as an input parameter by the program user. But the detailed physical processes causing destratification are generally not modelled, so it is this parameter which largely determines the degree of thermal stratification predicted by the model. (A single-node model, for example, always corresponds to a fully-mixed tank.) Hence the problem for the user is how to choose the number of nodes so as to get the best agreement with actual performance.

## 6.2 Recommendations

A sensible approach to dealing with stratification would be to first see whether there was any significant difference in the annual performance predicted with a single node and many nodes (corresponding to fully-mixed and perfectly-stratified storage). Only if there were would there be any justification for trying to model the stratification more closely.

For extreme cases the choice of model can be straightforward. In a direct system (one having no heat exchanger) with high flowrates, the tank will be fully mixed during charge and recharge and a single-node model is appropriate. The stratification that occurs during stand-by can be taken into account using an appropriately weighted value of the heat-loss coefficient as suggested in §5.3 above. Similarly, the limiting case of a large number of nodes (corresponding effectively to plug flow) will be a good approximation for in an indirect system with low flowrates. The problem arises when the flowrates are intermediate and the long-term performance prediction is sensitive to the number of nodes. Then a separate test may be thought necessary.

Clearly the most accurate information on the thermal stratification within a storage tank can be provided by direct measurements of the temperature profiles using internal thermal probes. However, this information cannot readily be interpreted in terms of a best choice for the number of nodes. An alternative under development within the CEC Solar Storage Testing Group (primarily for storage in phase-change materials) is to measure the output temperature profile of the store in response to a step change in inlet temperature, and then to choose as the number of nodes the value which gives the best agreement between the measured response and that predicted by the

computer model. Further details of this approach are given by van Galen [1] and Marshall [2].


## 6.3  References

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## 7.1   Introduction

Although external (counter-current) heat exchangers are not common in small, single-family SDHW systems, there may be an occasional need to measure their thermal performance. This presents no difficulty when the heat exchanger is operated conventionally, with pumped flow on both the primary and secondary sides (cf. §7.2, below). But in solar systems a heat exchanger is now more likely to be operated with natural circulation of heat-transfer fluid in the secondary loop. The Task III participants have therefore devised a test method for external heat exchangers operating in thermosyphon mode. This new method is outlined in §7.3.

The testing of external heat exchangers is based on the conventional theory of counter-current heat exchangers given in standard texts such as [1].
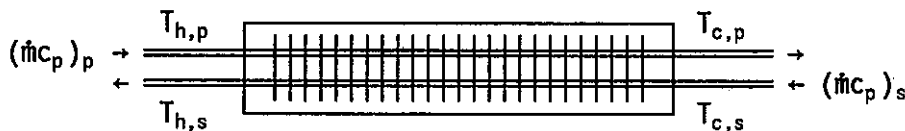
On the primary side the heat exchanger is fed with hot liquid at a temperature $T_{h,p}$ and a heat-capacity flowrate $(\dot{m}c_p)_p$. On the secondary side the heat exchanger is fed with cold liquid at a temperature $T_{c,s}$ and a heat-capacity flowrate $(\dot{m}c_p)_s$. As a result of the exchange of heat between the two streams, the liquid on the secondary side is warmed to a temperature $T_{h,s}$ and the liquid on the primary side is cooled to a temperature $T_{c,p}$. The rate at which heat is supplied to the secondary fluid is

(7.1)        $\dot{Q} = (\dot{m}c_p)_s (T_{h,s} - T_{c,s})$ .

$(UA)_{hx}$, the product of the heat-transfer coefficient and area of the heat exchanger, is given by

(7.2)        $(UA)_{hx} = \dot{Q}/\Theta$ ,

where

(7.3)        $\Theta = [(T_{h,p} - T_{h,s}) - (T_{c,p} - T_{c,s})]/\ln[(T_{h,p} - T_{h,s})/(T_{c,p} - T_{c,s})]$

is the log mean temperature difference.

The effectiveness $\varepsilon$ of the heat exchanger is defined by the expression

(7.4)        $\varepsilon = \dot{Q}/[(\dot{m}c_p)_{min}(T_{h,p} - T_{c,s})]$ ,

where $(\dot{m}c_p)_{min}$ is the smaller of $(\dot{m}c_p)_p$ and $(\dot{m}c_p)_s$.

The parameters $(UA)_{hx}$ and $\varepsilon$ are in general functions of $(\dot{m}c_p)_p$, $(\dot{m}c_p)_s$, $T_{h,p}$, and $T_{c,s}$. In the simulation model, however, they may be regarded as constants. (Following de Winter [2], a constant heat-exchanger effectiveness may simply

be incorporated into a heat-exchanger penalty factor for the solar collectors.) This assumption is not always justified: the temperature sensitivity may be significant, for example, when the heat exchanger is undersized. Hence the first objective of a test should be to establish whether the model is adequate for the particular heat exchanger. This can be done by measuring the dependence of the heat-exchanger parameters on operating conditions and by performing annual simulations to find the sensitivity of annual system performance to likely variations in these parameters.

## 7.2    External heat exchanger with pumped flow on both sides

The testing of external heat exchangers with pumped flow on both sides is straightforward since all the variables used to define the effectiveness or heat-transfer coefficient are directly measurable. The results are generally less temperature and flowrate dependent than for external heat exchangers with thermosyphon flow or immersed heat exchangers with pumped flow.

## 7.3    External heat exchanger with thermosyphon flow on secondary side

### 7.3.1 Introduction

The testing of external heat exchangers with pumped flow on the primary (supply) side and thermosyphon flow on the secondary (load) side is not so straightforward. In this case the flow through the secondary side of the heat exchanger is driven by natural convection due to the transfer of heat. Hence the flowrate in the secondary circuit is limited by buoyancy, and its value must be determined without changing the hydrodynamic resistance of the circuit. The test method suggested below is intended to avoid the necessity of measuring this flowrate.

Since the temperature profile across the secondary conduit in the heat exchanger is different than with forced convection, the heat-transfer coefficient and heat-exchanger effectiveness generally have smaller values than would be measured with pumped flow – even at the same flowrates. The parameters depend on the geometry of the secondary circuit, especially on the height of the heat exchanger relative to the tank. They also depend on the temperature profile within the tank, but to first order the dependence is just on the mean tank temperature. Also, because the processes are thermally driven, the parameters are likely to have a significant temperature dependence.

Simulation models for thermosyphon systems usually have detailed physical models within the program. Providing the heat-exchanger parameters are not too dependent on temperature and flowrate, the following test procedure should provide values which enable the system to be treated as one with pumped flow. It must be stressed that the procedure is a proposal which has not been validated by the Task III participants.

## 7.3.2 Proposed test method
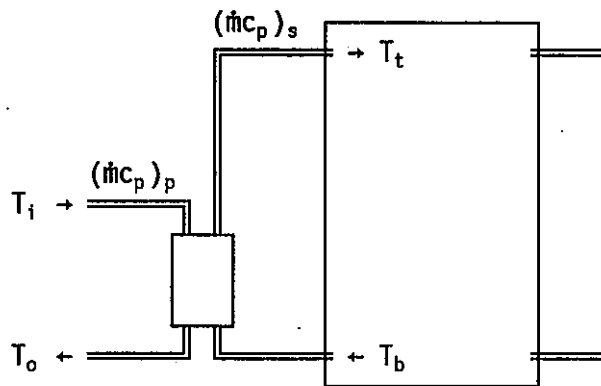


The heat exchanger is installed in the normal counterflow way, with the inlet to the heat exchanger on the secondary side on the same horizontal level as the tank outlet to which it is connected. Temperature sensors are placed at the inlet and outlet to the heat exchanger on the primary (supply) side, and at the inlet and outlet ports of the tank on both the supply side and the load side. The system is filled.

The water in the tank is heated and mixed by circulation between the inlet and outlet ports on the load side of the tank until a uniform temperature $T_s$ (measured at the load-side inlet and outlet ports) is achieved throughout the tank.

Heat-transfer fluid is pumped through the primary side of the heat exchanger at a constant heat-capacity flowrate $(\dot{m}c_p)_p$ and with a constant inlet temperature $T_i$. When the outlet temperature on the supply side of the heat exchanger has become quasi-stationary, its value $(T_o)$ is recorded.

The corresponding quasi-stationary water temperatures at the inlet and outlet of the heat exchanger on the secondary side ($T_b$ and $T_t$, respectively) are measured at the supply-side inlet and outlet ports of the tank. $T_b$ should have the value measured for $T_s$, and it is assumed that during the time in which the quasi-stationary state is achieved the mean temperature in the tank has not changed significantly from this initial value.

The rate of heat output from the heat exchanger is

(7.5) $\qquad \dot{Q} = (\dot{m}c_p)_s(T_t - T_b)$ ,

and the product $(UA)_{hx}$ of the heat-transfer coefficient and area of the heat exchanger is

(7.6) $\qquad (UA)_{hx} = \dot{Q}/\theta$ ,

with, from Equation (7.3),

(7.7) $\qquad \theta = [(T_i - T_t) - (T_o - T_b)]/\ln[(T_i - T_t)/(T_o - T_b)]$ .

The effectiveness $\varepsilon$ of the heat exchanger is, from Equation (7.4), given by

(7.8)    $\varepsilon = \dot{Q}/[(\dot{m}c_p)_{min}(T_i - T_b)]$ ,

where $(\dot{m}c_p)_{min}$ is the smaller of $(\dot{m}c_p)_p$ and $(\dot{m}c_p)_s$, the quasi-stationary heat-capacity flowrate in the secondary side of the heat exchanger. The value of $(\dot{m}c_p)_s$ is not, however, assumed to be measured. When the value is not available we assume in its place that there are no significant heat losses from the heat exchanger. In that case $\dot{Q}$ is also equal to the rate of input of heat into the heat exchanger,

(7.9)    $\dot{Q} = (\dot{m}c_p)_p(T_i - T_o)$ .

Equations (7.5) and (7.9) enable $(\dot{m}c_p)_s$ to be determined from the other measured variables. [If the flowrate is measured directly, using a calibrated magnetic flow meter for example, then measurement could be used to estimate the thermal losses.]

Under the assumption of no thermal losses $\varepsilon$ can be calculated simply as

(7.10)    $\varepsilon = \Delta T/(T_i - T_b)$ ,

where $\Delta T$ is the greater of $(T_i - T_o)$ and $(T_t - T_b)$, the temperature differentials on either side of the heat exchanger.

In this way the parameters $(UA)_{hx}$ and $\varepsilon$ can be measured as a function of $(\dot{m}c_p)_p$, $T_i$, and the mean tank temperature $T_s$.

It must be stressed when quoting the results of the test that they only apply when the heat exchanger is mounted as in the test. If it is known that the heat exchanger is to be mounted at a different height relative to the tank, then test results should be obtained for that configuration. It may even be thought necessary to perform two tests, one with the heat exchanger level with the bottom of the tank and the other with it level with the top, and thus provide best and worst performance characteristics.


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# 8    IMMERSED HEAT EXCHANGERS

## 8.1    Introduction

An immersed heat exchanger consists of a primary side entirely surrounded by a heat bath to which the input heat is transferred. The flow of heat over the heat exchanger is by natural convection, with very little resistance to flow, and hence the temperature differential outside the heat exchanger is small. Regarded as a a counter-current heat exchanger (cf. §7.1) with the tank as the secondary side, the immersed heat exchanger thus has no thermal losses, an unlimited heat-capacity flowrate on the secondary side, and inlet and outlet temperatures on the secondary side both equal to mean temperature $T_s$ of the surrounding fluid.

If the fluid on the supply side has an inlet temperature $T_i$, an outlet temperature $T_o$, and a heat-capacity flowrate $\dot{m}c_p$, then the rate of transfer of heat $\dot{Q}$ is given by

(8.1)    $\dot{Q} = \dot{m}c_p(T_i-T_o)$ .

The log mean temperature difference (cf. Equation (7.3)) reduces to

(8.2)    $\theta = (T_i-T_o)/\ln[(T_i-T_s)/(T_o-T_s)]$ ,

and hence the product $(UA)_{hx}$ of the heat-transfer coefficient and area of the heat exchanger,

(8.3)    $(UA)_{hx} = \dot{Q}/\theta$ ,

is given by

(8.4)    $(UA)_{hx} = \dot{m}c_p\ln[(T_i-T_s)/(T_o-T_s)]$ .

The effectiveness $\varepsilon$ of the heat exchanger is (cf. Equation (7.4)) given by

(8.5)    $\varepsilon = \dot{Q}/[\dot{m}c_p(T_i-T_s)]$ ,

or, equivalently, by

(8.6)    $\varepsilon = (T_i-T_o)/(T_i-T_s)$ .

Note that $(UA)_{hx}$ and $\varepsilon$ are related by

(8.7)    $(UA)_{hx} = -\dot{m}c_p\ln(1-\varepsilon)$ .

The heat-exchanger effectiveness can vary significantly with flowrate and operating temperatures, and some simulation models take this into account. The Danish program BSOL [1,2] assumes a variation in effectiveness that is linear in the variation in temperature from a nominal value, while the Swiss program SIWW [3] contains an explicit analytic model. Some simulation programs (such as TRNSYS [4]) assume a constant effectiveness and treat the immersed heat exchanger in the same way as an external heat exchanger. The Canadian simulation program WATSUN [5] offers the user a choice: either a detailed model can be used, with the dimensions, materials, etc. specified as input, or a constant effectiveness can be specified. A Swiss evaluation of different formulae for the heat transfer in in-tank heat exchangers is reported in [6].

With simulation programs that require only one effectiveness number it may be advisable to perform runs to determine the sensitivity of annual performance to the value of the heat-exchanger effectiveness. This was recommended, for example, in a recent SERI review of models and test procedures for immersed heat exchangers [7].

In the CEC SSTG method of testing immersed heat exchangers [8] a storage charge efficiency is measured as a function of flowrate and from this a flowrate-dependent heat-exchange factor is determined. Upper and lower bounds for the UA value of the heat exchanger are deduced from plots of the heat-exchange factor, and the average of these values is used to calculate a value for the heat-exchanger effectiveness. In the Danish Solar Testing Laboratory approach [9] the value of $(UA)_{hx}$ is deduced directly from measurements of the inlet and outlet temperatures of the heat exchanger and the temperature of the storage fluid surrounding the heat exchanger. The following method of test, developed by the Task III participants, is designed to measure the heat exchanger parameters without the need to make internal measurements of tank temperature.


## 8.2   Test procedure

The inlet and outlet manifolds of the heat exchanger are joined in a closed, thermally-insulated loop containing a controllable in-line heater, a pump with a means of measuring and controlling the flow, and temperature sensors at the inlet and outlet manifolds. The equipment should be capable of rapidly establishing and maintaining a constant flow of fluid through the heat exchanger at a constant inlet temperature, and this should be possible for any combination of flowrate and inlet temperature encountered in normal operation.

Because the heat-exchanger parameters generally depend on the properties of the heat-transfer fluid, it is important that the fluid in the heat-exchanger loop is identical to the fluid which will be used in operation.

The inlet and outlet ports of the tank are similarly joined in a closed insulated loop containing a pump. Valves are installed in the pipework at the inlet and outlet ports, but in this loop there is no requirement for a heater or temperature sensors. The tank and secondary loop are filled with mains water.

The test consists of a pre-conditioning stage, in which the tank and heat exchanger are brought to a common uniform temperature, followed by a heating stage, in which the heat-exchanger parameters are measured. The second stage should follow immediately after the first.

*Preconditioning stage*

$T_s$
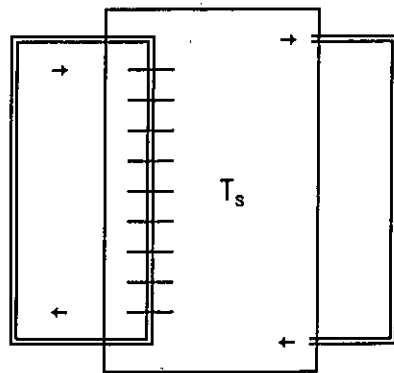
With the valves at the tank inlet and outlet ports open and the inline heating turned off, both pumps are operated so as to slowly circulate heat-transfer fluid through the heat-exchanger and water through the tank. The temperatures at the heat-exchanger inlet and outlet manifolds are monitored until they have both stabilized with the same steady value. The tank and heat exchanger are then assumed to have a uniform temperature $T_s$ equal to this value. ($T_s$ can, of course, be pre-set to any approximate value by previously heating the tank.)

*Heating stage*

$T_i$

$\dot{m}c_p$

$T_s$     $T_a^s$

$T_o$
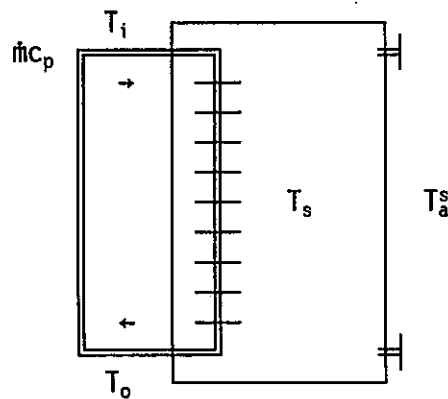
The pump in the secondary loop is now switched off, and the valves at the inlet and outlet ports of the tank are closed. The heater and pump in the heat-exchanger loop are operated so as to maintain a constant inlet temperature $T_i$ at a constant mass flowrate $\dot{m}$. The values of $T_i$ and $\dot{m}$ are those at which the heat-exchanger parameters are required to be measured.

It is assumed that in the time which it takes for the temperature at the outlet of the heat exchanger to reach a quasi-stationary value $T_o$, natural convection around the outside of the heat exchanger has prevented the temperature of the surrounding fluid from rising significantly above the initial value $T_s$.

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The product of the heat-transfer coefficient and area of the heat exchanger is calculated according to Equation (8.4), as

(8.8)    $(UA)_{hx} = \dot{m}c_p \ln[(T_i-T_s)/(T_o-T_s)]$ ,

and its effectiveness according to Equation (8.6), as

(8.9)    $\varepsilon = (T_i-T_o)/(T_i-T_s)$ .

By repeating the two stages of the test under different conditions, these parameters can be determined as a function of the variables $T_i$, $T_s$, and $\dot{m}c_p$.

Note that the procedure gives a value just for the product $(UA)_{hx}$. If the simulation program requires separate values for the heat-transfer coefficient and area, then a nominal value can be given for one and the other chosen to give the correct product.


## 8.3   Experimental trial

The test method outlined in the previous section has been subjected to laboratory trials by the Paul Scherrer Institute in Switzerland, and the results have been reported by Suter and Brack [10].

The method was compared, using five different combinations of storage tank and heat exchanger, with the normal "three-sensor" procedure in which the store temperature around the immersed heat exchanger is measured directly. The method tended to give about a 5% lower value for $(UA)_{hx}$ than the three-sensor procedure, a difference which would normally have a negligible influence on the prediction of long-term performance.

A major difficulty encountered with the three-sensor method was a strong dependence of the results on the position of the sensor measuring the store temperature. This was especially the case for small log mean temperature differences or for vertically-mounted heat exchangers. If the storage sensor is not positioned correctly, the discrepancy between this method and the new procedure could be as large as 20%.

It was found during the trials that a significant temperature gradient could develop over the outside of a heat exchanger mounted vertically in the tank. This showed that it is important to perform the preconditioning stage carefully before each heating stage to achieve a thorough mixing and thus ensure a uniform temperature.

One test showed a significant rise in the mean storage temperature during the time in which the inlet and outlet fluid temperatures of the heat exchanger reached their quasi-stationary values, but in this case the storage volume was 250 l and the power input was equivalent to the maximum from a collector array of area 7.5 m² – an unusually low ratio of storage volume to collector area. And even in this unfavourable case the observed rise in mean storage temperature was only 3% of the log mean temperature difference.

Further details of the experimental design and a summary of the results are given in the paper.

Suter and Brack concluded that the method proposed in the present document could be recommended in preference to the three-sensor procedure.

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# 9    PIPE PARAMETERS

## 9.1  Recommendations

For determining the piping parameters, i.e. the thermal capacitance and heat-loss coefficient per unit length, calculation is normally adequate.

To avoid underestimating the real pipe losses it is recommended that checks be made to ensure that the insulation is well installed. A procedure for checking this and other aspects of the integrity of the system is given in the Task III inspection procedure [1].

## 9.2  References

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# 10    SENSORS AND CONTROLLERS

## 10.1  Recommendations

The sensor and controller parameters, such as temperature differentials and switching times, are prescribed as part of the system design. Before the nominal values are used for predicting system performance some checks are recommended to ensure that the actual parameters are in agreement with them. Methods for the determination of controller parameters are recommended in the inspection procedures document [1].

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# 11 EXAMPLES OF COMPONENT-BASED SYSTEM TESTING EXPERIENCE

## 11.1 Danish Solar Energy Testing Laboratory

Component-based SDHW system testing at the Danish Solar Energy Testing
Laboratory provides the national standard for the Danish Solar Industry. The
method of testing is summarized in §2.6 of [1].

The test procedures were developed by the Testing Laboratory in close
collaboration with the Thermal Insulation Laboratory of the Technical
University of Denmark. Details of the procedures are published as a Testing
Laboratory report [2], to which frequent reference has been made in the
present document.

As part of the test, measurements are made on a complete system installed by
the manufacturer on an outdoor test stand at the Laboratory. A comparison
between the monitored data and the results of a computer simulation with the
measured component parameters as input data is used for validation and to
reveal installation and operational defects.

The test results are presented on standard data sheets of which examples are
given in both [1] and [2]. The results include annual performance predictions
for different collector areas, loads, and system configurations, based on the
Danish Institute of Building Research Test Reference Year.

## 11.2 TNO Institute of Applied Physics

At the TNO Institute of Applied Physics Heat Systems Department in Delft a
component-based test on an SDHW system was recently compared with indoor and
outdoor tests on the same system, and the results were reported to the Task
III participants [3].

The system was of drain-back type, with selective-absorber, single-glazed
collectors directly connected to the storage tank. The mains water was heated
to the demand temperature using an immersed heat exchanger in the store, an
in-line auxiliary heater, and a cold-water mixing valve.

The component tests were made according to the CEC SSTG recommendations [4],
with the collectors tested in a simulator using to the CEC CSTG procedure [5].
A full table of measured parameter values is given in [3]. The annual
performance was simulated with these values as input data using the TPD SDHW
computer model.

The measured value of $(UA)_s$ was $(1.0 \pm 0.2)$ W/K compared with a theoretical
value of 1.1 W/K, and the measured storage heat capacity was 0.45 MJ/K
compared with a theoretical value based on the water content of 0.50 MJ/K.
With the tank fully mixed, and the mains water input to the heat exchanger at
a flowrate of 5 l/min and temperature of 10 °C, the value of $(UA)_{hx}$ was found
to be 770 W/K, corresponding to an effectiveness $\varepsilon$ of 0.89. These values were
used in the simulation, but it was noted that the effectiveness would in
practice be influenced by a difference in flowrate and stratification in the
storage tank.

Under standard representative conditions, with a fixed mains-water temperature, demand temperature, draw-off volume and load profile, and a specific set of meteorological data, the solar contribution of the system was calculated in the simulation as 3500 MJ.

The indoor test was performed as described in §2.8.2 of [1], the method being of 'adaptive simulation' type. The system was operated in a solar simulator with the irradiance and the ambient, collector-inlet, collector-outlet, and mean storage temperatures each monitored. The TPD SDHW simulation model was then used to simulate the test, with the input parameters chosen to obtain the best fit to the monitored data. The estimated value of $(UA)_s$ was 2.4 W/K, of the storage heat capacity was 0.5 MJ/K, and of $(UA)_{hx}$ was 700 W/K. Assuming the parameter values estimated from the indoor test, the simulation model predicted an annual solar contribution under the standard conditions of 3100 MJ.

The outdoor test was performed according to the method described in §2.8.3 of [1]. The performance of the system was monitored over a six-week period at the TNO-TH outdoor test facility and the data used to define efficiency indices. These indices were then used to extrapolate to the annual performance under the standard conditions. The measured storage efficiency suggested a value of 2.0 W/K for the storage losses, and the annual solar contribution for the standard conditions was in this case estimated as 3250 MJ.

The agreement between these three methods was within about 12%.

## 11.3 Swiss Participants of Subtask E

Following a review of SDHW performance test methods [6] to identify an approach which would meet the needs of the Swiss Professional Association of Solar Energy Firms (SOFAS), it was decided to adopt the adaptive-simulation method of testing, with in-situ measurements on the system used to identify the parameters of a detailed simulation program. An introduction to the Swiss methodology was given in §2.10 of the summary of national approaches [1].

Although the test method under development within the SOFAS programme was not a component-based test, direct contributions to the Task III work on the development of component performance tests were made from theoretical and experimental research on system components, system sizing, and system analysis carried within another SOFAS programme [7-9], at the Burgdorf School of Engineering [10], and at the former Swiss Federal Institute for Reactor Research (EIR) - now the Paul Scherrer Institute. Among these may be mentioned the identification of the most important technical factors that determine the energy balance of a system [11], investigations of the heat loss mechanisms in hot water storage tanks [7-9,12,13], and the validation of the SIWW simulation programme package [10] using both monitored data from five side-by-side SDHW systems and separately measured component parameters [7-9]. On an annual basis this validation showed an agreement between simulation and experiment within 5% for most of the terms in the energy balance - as close as possible given the errors of measurement.

In addition, as described in §8.3 of this report, EIR carried out on behalf of the Task an extensive experimental validation of the proposed method for testing immersed heat exchangers.

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# PART II: CORRELATION-BASED TESTING

## 12.1   Introduction

ANSI/ASHRAE Standard 95-1981 [1] provides a uniform method for testing the thermal performance of solar domestic-hot-water systems under independently-specified test conditions. The Standard gives a system performance rating that is appropriate to operating conditions that are on average close to the test conditions.

The purpose of the test method outlined below is to provide a means by which a suitable series of short-term ASHRAE 95 test measurements may be used to predict the long-term (annual) performance of solar water-heating systems under arbitrary meteorological conditions and with a variable load.

The method assumes a model for the stationary performance of the system based on a correlation proposed by Klein and Fanney [2] combined with a new correlation that avoids the necessity of measuring internal tank temperatures.

If the testing is performed in a solar simulator, then the separate solar-collector performance test required by ASHRAE Standard 95 is not needed as part of the short-term tests. However, knowledge of the incidence-angle modifier is needed to predict the long-term performance.

The method was developed and validated jointly by the participants of Task III to meet a set of commonly agreed criteria. The theoretical basis for the method is summarized in Appendix A of this report.

## 12.2   General

### 12.2.1 Scope of test

The test procedure is intended to be applicable to any system designed for the solar heating of domestic hot water only.

The system may or may not incorporate a heat exchanger in the collector-storage loop, and the circulation may be pumped or by natural convection. The system may or may not incorporate an auxiliary thermal source.

The procedure may be applied to integral collector-storage systems and other systems employing a collector/heat-transfer-fluid combination which cannot be tested according to ASHRAE Standard 93 provided only that the incidence-angle modifier of the solar-collector component can be estimated with sufficient accuracy to adequately account for its effect on the long-term performance of the system.

## 12.2.2 Definitions, classifications, requirements, instrumentation, and apparatus

For this test procedure the definitions, classifications, requirements, instrumentation, and apparatus are as specified in the ANSI/ASHRAE Standard 95-1981 [1].


## 12.2.3 Nomenclature

*Note: The nomenclature used here is not generally equivalent to that of ASHRAE Standard 95, where the same symbols may be used with significantly different meanings.*


## 12.3  Short-term tests


### 12.3.1 Test procedure

In the ASHRAE Standard 95 test procedure, the system to be tested is operated for a succession of days under identical conditions until a stationary performance has been achieved (so that the carry-over of stored energy to the following day is equal to that left from the previous day). The system performance that is measured by the Standard is the stationary performance achieved in the last day of the sequence, which we shall refer to as a "test day". A test day is therefore just one of stationary system performance, which may in the laboratory take two or three actual days to achieve.*

The fixed daily test conditions - irradiance profile, temperature of the water supplied to the system, collector ambient temperature, set temperature for the water supplied by the system, total mass and draw-off profile of the heated water, and so on - are not specified in the ASHRAE Standard but by rating standards such as the SRCC [3]. The conditions imposed by the rating standard are usually intended to ensure that the measured test performance is representative of typical operating conditions.

In the present test procedure the ASHRAE 95 test is repeated for a number of test days, each corresponding to a different set of stationary test conditions. The test results are used to estimate characteristic system parameters which describe the sensitivity of the performance to different operating conditions, and which may then be used to predict the long-term performance of the system. Hence the test conditions for the different test days are generally not chosen to be typical, but to give a sufficient variability that the individual system parameters can be reliably identified.

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* There may be systems which never reach stationarity, e.g. through hysteresis of the auxiliary-heater controller.
(D Proctor and S R James, Solar Energy 36, 345-360, 1986.)

## 12.3.2 Test conditions

The test conditions should ensure a sufficient independent variability in the measured variables that precise estimates of system parameters can be determined from the test data. A proper experimental design has yet to be performed, but the following minimum requirements are recommended:

- There should be at least seven different test days.

- The length of the simulated solar days should vary from about six to about ten hours independently of the daily total irradiation (so that both high and low irradiation is simulated on long and short days).

- The irradiance profiles should have approximately the sinusoidal shape of clear-sky conditions.

- The difference between the mains water temperature and the ambient temperature of the collector should vary over a range of at least ±10 °C within the set of test days.

- At least two different daily draw-off volumes should be used, one close to the volume of the solar storage tank, and one significantly greater.

- For any test day the draw-off mass should be sufficiently great that, in view of the daily total irradiance, collector area, demand temperature and mains supply temperature, saturation cannot occur. (Hence, e.g., no cold-water mixing should occur at draw-off.) However, the incident solar energy on the collectors should represent at least 25% of the load.

- The draw-off profile should be such that the draw off occurs mostly during simulated daylight hours.

## 12.3.3 Data to be collected and recorded

The tester should record:

- the number of test days;

- the number of equal time increments $\Delta t$ in one (24-hour) test day, n;

For each test day the following variables should be recorded:

- the test-day number;

- the total draw-off mass of water from the system during the day, $M_L$ (kg);

- the average over the day of the mains water temperature, $T_m$ (°C);

- the average collector ambient temperature over the day, $T_a$ (°C);

- the average store ambient temperature over the day, $T_a^s$ (°C);

- the total energy delivered by the system during the day, $Q_L$ (MJ);

- the total auxiliary energy supplied for water heating during the day, $Q_{AUX}$ (MJ).

Additionally, for each of the n time increments of the test day should be recorded:

- the average value over the time increment of the total irradiance normal to the collector array multiplied by the incidence-angle modifier for the collector array, $I_T K_{\tau\alpha}$ (W/m$^2$).

In a solar simulator with the radiation incident normal to the collector array, $K_{\tau\alpha} = 1$.

If the system is operated with a thermostat-controlled auxiliary heat source, then we may expect that the energy supplied by the system $Q_L \approx M_L c_p (T_{set} - T_m)$, where $T_{set}$ is the set temperature (or 'demand temperature'). In practice, the thermostat and heater will not respond perfectly, so $Q_L$ should be determined directly from measurements at the delivery point. Note that parasitic energy is not included in $Q_{AUX}$.

For systems without an auxiliary heat source, a value of zero is recorded each day for $Q_{AUX}$.

## 12.3.4 Example

The table of data shown on the following page has been constructed from the test data presented in the paper by Klein and Fanney [2]. These data do not satisfy the conditions set out in §12.3.2 (and the consequences of this for the identification of the system parameters are seen in §12.4.3), but they illustrate the form of data set required.

There are 9 test days, each containing 48 half-hourly time increments.

The values of draw-off mass (which were not given in the original data) have been estimated for each day as $Q_L / (T_{set} - T_m)$. These values are enclosed in brackets to show that they are estimated. The daily draw-off volumes had the same nominal values, so the variability in $M_L$ is less than would be needed for the present test method. The values of $Q_{AUX}$ were obtained from $Q_L$ and the solar fraction $f = (Q_L - Q_{AUX})/Q_L$.

Test days 7 and 8 are days in which there was no circulation of heat-transfer fluid to the collectors. (Such days are excluded from the present test method because the solar contribution to the load is not significantly greater than the ambient gains or losses.) The value given for the collector ambient temperature on these days is an arbitrary value that has been set sufficiently low that no ambient gains from the collector are assumed in the analysis of data. These values are accordingly also put in brackets. For test day 8 the data are corrected according to information supplied by S A Klein.

The ambient temperature around the store is in this example different from the collector ambient temperature, but in some cases - such as systems with collector-integrated storage - they may be physically identical.

Note that although in these data there is a significant variation in total daily irradiance, the variation in solar day length (excluding the no-solar days) is only of one or two hours.

9 48

1 (272.2) 10.6 2.2 20.0 56.280 44.799
   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.
   0.   0.  44.  94. 150. 208. 265. 316. 360. 393. 413. 420.
 413. 393. 360. 316. 265. 208. 150.  94.  44.   0.   0.   0.
   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.

2 (266.4) 19.9 8.1 20.0 44.714 28.617
   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.
  38.  92. 152. 217. 283. 349. 411. 467. 513. 548. 569. 577.
 569. 548. 513. 467. 411. 349. 283. 217. 152.  92.  38.   0.
   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.

3 (254.7) 19.7 24.0 20.0 42.961 15.036
   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.
 122. 186. 253. 322. 390. 456. 516. 569. 613. 646. 666. 672.
 666. 646. 613. 569. 516. 456. 390. 322. 253. 186. 122.   0.
   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.

4 (263.2) 10.9 25.0 20.0 54.080 13.520
   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.
   0.   0. 189. 293. 403. 514. 619. 714. 793. 854. 891. 904.
 891. 854. 793. 714. 619. 514. 403. 293. 189.   0.   0.   0.
   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.

5 (281.1) 10.8 2.2 20.0 57.946 24.511
   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.
   0.   0. 189. 293. 403. 514. 619. 714. 793. 854. 891. 904.
 891. 854. 793. 714. 619. 514. 403. 293. 189.   0.   0.   0.
   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.

6 (287.8) 20.5 31.0 20.0 47.578 3.568
   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.
 141. 242. 348. 455. 562. 664. 758. 840. 907. 958. 989. 999.
 989. 958. 907. 840. 758. 664. 562. 455. 348. 242. 141.   0.
   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.

7 (261.0) 19.6 (0.0) 20.0 44.132 47.530
   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.
   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.
   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.
   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.

8 (263.3) 11.0 (0.0) 20.0 54.000 53.838
   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.
   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.
   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.
   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.

9 (252.5) 19.9 24.0 20.0 42.376 15.086
   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.
 122. 186. 253. 322. 390. 456. 516. 569. 613. 646. 666. 672.
 666. 646. 613. 569. 516. 456. 390. 322. 253. 186. 122.   0.
   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.   0.

## 12.4 Analysis of short-term test data

### 12.4.1 Correlation model

The correlation model fitted to the test data can be expressed in the coupled equations:

(12.1)
$$(1+c_3 c_4 D/M_L c_p)Q_S + c_3 D(T_m - T_a^s)$$

$$= (c_1 \Delta t)\sum[(I_T K_{\tau\alpha}) - c_2\{(T_m - T_a) + c_4 Q_S/M_L c_p\}]^+$$

and

(12.2)
$$Q_L - Q_{AUX} = Q_S - c_5 D(T_w - T_a^s) \ ,$$

where

$D$ is the length of the day (i.e. 24x60x60 s),

$M_L$ is the total draw-off mass of water from the system during the day,

$c_p$ is the specific heat capacity of water,

$Q_S$ is the energy delivered during the day from the collectors and the solar storage parts of the system,

$T_m$ is the mains water temperature for the day,

$T_a^s$ is the store ambient temperature for the day,

$\Delta t = D/n$ is the time increment for irradiance measurements,

$I_T$ is the average total irradiance normal to the collector array over the time increment $\Delta t$,

$K_{\tau\alpha}$ is the average incidence-angle modifier over the time increment $\Delta t$ ($K_{\tau\alpha} = 1$ for normal incidence),

$T_a$ is the collector ambient temperature for the day,

$Q_L$ is the total energy delivered by the system during the day,

$Q_{AUX}$ is the total auxiliary energy supplied for water heating during the day,

$T_w = Q_L/M_L c_p + T_m$ is the weighted average temperature over the day of the water delivered by the system.

The summation on the right-hand side of Equation (12.1) extends over all the n short time increments $\Delta t$ which make up the day. The superfix '+' on the outer square brackets indicates that the term within the brackets is to be included in the summation only if positive.

The coefficients $c_1$, $c_2$, $c_3$, and $c_4$ in Equation (12.1), and $c_5$ in Equation (12.2), are system parameters to be identified by best fit to the test data. The coefficients may be given the following interpretations:

45

(12.3)    $c_1 = A_a (F_R'/F_R) K_s F_R (\tau\alpha)_{e,n}$ ,    $[m^2]$

(12.4)    $c_2 = F_R U_L /F_R (\tau\alpha)_{e,n}$ ,    $[W/(m^2\ K)]$

(12.5)    $c_3 = (UA)_s$ ,    $[W/K]$

(12.6)    $c_4 = 1/J_s$ ,    [dimensionless]

(12.7)    $c_5 = (UA)_{AUX}/J_{AUX}$ ,    $[W/K]$

where

$A_a$ is the collector aperture area,

$F_R'/F_R$ is the de Winter collector-heat exchanger penalty factor [4],

$K_s$ is the Phillips and Dave stratification coefficient [5],

$F_R (\tau\alpha)_{e,n}$ and $F_R U_L$ are collector efficiency parameters,

$(UA)_s$ is the product of the surface area and heat-loss coefficient of the solar store,

$J_s$ is a second stratification coefficient for the solar store (defined in Appendix A),

$(UA)_{AUX}$ is the product of the surface area and heat-loss coefficient of the auxiliary tank,

$J_{AUX}$ is the equivalent of $J_s$ for the auxiliary tank.

Equations (12.3) and (12.4) are for an array with a single collector: for an array of N parallel rows each containing M collectors the corresponding expressions for $c_1$ and $c_2$ can be derived as in Appendix B of [1].

The estimated value of $c_5$ should be effectively equal to zero if the system contains no auxiliary heat source, and may be taken to be $(UA)_{AUX}$ if there is an auxiliary heater (since in most cases $J_{AUX}$ is not expected to be significantly different from 1).


12.4.2 Method of parameter estimation

The recommended method of parameter estimation is to choose the values of the c-parameters which give the least-squares fit to Equation (12.2) with $Q_s$ satisfying Equation (12.1). Thus the function to be minimized with respect to trial values of $c_1$, $c_2$, $c_3$, $c_4$ and $c_5$ is

(12.8)    $S = \sum_{\text{test days}} [Q_L - Q_{AUX} - Q_S + c_5 D(T_w - T_a^s)]^2$ ,

in which for each test day $Q_L$, $Q_{AUX}$, $T_w$, and $T_a^s$ have their measured values, and $Q_s$ is obtained by solving Equation (12.1) with the measured values of $M_L c_p$, $I_T K_{\tau\alpha}$, $T_m$, $T_a^s$, and $T_a$. Equation (12.1) is a piecewise linear function in $Q_s$, which is easily solved by iteration - e.g. by Newton's method, which typically converges in one or two iterations. S, the sum of the squares of the residues, is a continuous, non-linear function of the parameters with discontinuous first derivatives, and the parameter fit can be obtained by any appropriate multi-variate non-linear method.

It is important when using fitting routines to repeat the fit with different starting values of the parameters. This is necessary to ensure that a local minimum is not chosen in preference to the global minimum.


### 12.4.3 Example

Using the method of parameter estimation recommended in §12.4.2 (but modified to account for the assumption that there was no collection of night-time collector ambient gains), the data tabulated in §12.3.4 have been analysed by Spirkl [6], who obtained

$$c_1 = 2.31 \ (\pm 0.72) \ m^2$$

$$c_2 = 5.55 \ (\pm 0.89) \ W/(m^2 \ K)$$

$$c_3 = 6.88 \ (\pm 1.67) \ W/K$$

$$c_4 = 0.38 \ (\pm 0.36)$$

$$c_5 = 1.18 \ (\pm 0.25) \ W/K$$

The figures in brackets are ± the standard errors of estimation.

Since $K_s$ (which is not known) is expected to have a value slightly greater than 1, we would estimate from the component parameters measured by Klein and Fanney [2], that

$$c_1 = A_a \ (F_R^!/F_R) K_s \ F_R \ (\tau\alpha)_{e,n} > 4.19 \ m^2 \ x \ 0.833 \ x \ 0.805 = 2.81 \ m^2 \ .$$

Similarly, the expected value of $c_2$ based on the component parameters is

$$c_2 = F_R U_L /F_R (\tau\alpha)_{e,n} = (4.73 \ W/m^2)/0.805 = 5.88 \ W/m^2 \ .$$

Thus the values of $c_1$ and $c_2$ obtained by fitting the data are consistent with the measured values within the standard errors of estimation.

The parameters $c_3$, $c_4$, and $c_5$ cannot be directly estimated from the component parameters given in [2], so similar comparisons are not possible for them.

The relatively large standard errors of estimation of the parameters can be understood by inspection of the covariance matrix of the estimates [7]:

| $c_1$ | $c_2$ | $c_3$ | $c_4$ | $c_5$ | |
|---|---|---|---|---|---|
| 1.00 | 0.27 | 0.61 | 0.99 | -0.24 | $c_1$ |
| | 1.00 | 0.19 | 0.24 | -0.38 | $c_2$ |
| | | 1.00 | 0.51 | 0.22 | $c_3$ |
| | | | 1.00 | -0.35 | $c_4$ |
| | | | | 1.00 | $c_5$ |

This shows an almost complete correlation between $c_1$ and $c_4$, and substantial correlations between these parameters and $c_3$. Thus the estimation has a large degree of degeneracy. The cause of this degeneracy is the lack of variability in the data set, which was not of course intended for the present test method.

Although we are unable to determine the parameters with much precision in this case, the data set does give support for the model. For even with these uncertain parameter estimates, the estimated error of prediction of the daily values of $Q_L - Q_{AUX}$ from the model is 1.07 MJ (about 2% of the load).

## 12.5 Long-term performance prediction

### 12.5 1 Calculation on a daily basis

The prediction of the long-term performance of the system at a given site can be made by assuming the same model – given by Equations (12.1) and (12.2) – that was fitted to the short-term test data. For the long-term prediction the test-day values of the environmental variables are replaced by values from a TMY or other meteorological data set appropriate to the site of interest. Usually hourly-averaged values of $I_T$ are available, corresponding to $\Delta t = 1$ hour. Daily averages are used in place of the fixed test-day values of $T_m$, and $T_a$, and suitable daily values must be chosen for $M_L$ and $T_a^s$. The parameters $c_1$, $c_2$, $c_3$, $c_4$, and $c_5$ are given the values obtained from the fit to the short-term test data. With these substitutions, Equation (12.1) is solved for $Q_S$ exactly as for the short-term test.

If the system is operated without an auxiliary heat source, $Q_{AUX}$ and $c_5$ are both zero, and $Q_S$ gives directly the delivered energy $Q_L$. If the system is operated with an auxiliary heat source, then $T_w$ is fixed at the set temperature $T_{set}$, $Q_L$ is put equal to the corresponding load,

$$(12.9) \qquad Q_L = M_L c_p (T_{set} - T_m) \ ,$$

and the auxiliary energy required by the system to exactly meet this load can be estimated (cf. Equation (12.2)) as

$$(12.10) \qquad Q_{AUX} = [M_L c_p (T_{set} - T_m) + c_5 D(T_{set} - T_a^s) - Q_S]^+ \ .$$

In this way the delivered energy or auxiliary energy (depending on whether or not there is an auxiliary energy source) is calculated on a day-by-day basis and summed over all days to give a prediction of long-term system performance.

It is important to note that the individual daily energy values have no physical significance, and it is only the sum of these values over a period of a year or more that can be expected to give a valid prediction of system performance. The daily calculations assume stationarity, and although this will not be true in general, a cancellation of errors has been found to result in a valid long-term prediction [8].

### 12.5.2 Calculation on a monthly basis

If for the site of interest only monthly-averaged daily values of the meteorological variables are available, then these may be used to estimate average daily energies for each month. The daily averages are then simply multiplied by the number of days in the month to give the monthly totals.

## 12.5.3 Alternative methods of calculation

Very many detailed simulation models or simplified design methods have been developed for predicting the performance of solar water-heating systems, and the tester may wish to use one of these methods in preference to (12.1) and (12.2) to estimate the long-term system performance.

Although no alternative model requires as input just the parameters $c_1$, $c_2$, $c_3$, $c_4$, and $c_5$, it is an assumption of this test method that the long-term performance of a system in given environmental and operational conditions is completely determined by the values of these five parameters. Hence, in principle, any component parameters could be used in an alternative method so long as they are consistent with the estimated values of $c_1$, $c_2$, $c_3$, $c_4$, and $c_5$.

There are no standard ways of including the stratification coefficient $J_s = 1/c_4$ in simulation models, but the parameter $c_4$ can be taken into account implicitly. Firstly, inspection of Equation (12.1) shows that the same values of $Q_s$ would be obtained if $c_4$ were replaced by

$$c_4' = 1 ,$$

and $M_L$ by

$$M_L' = M_L/c_4 .$$

Hence, for a system without auxiliary heating we can estimate the long-term value of $Q_L$ (the energy delivered by the system) using a simulation model that assumes a fully-mixed solar store and an effective value $M_L'$ for the draw-off mass of water.

Similarly, for a system with an auxiliary heat source the net energy delivered by the system [i.e. $Q_L - Q_{AUX}$ with $Q_L$ given by (12.8)] is identical to that with the effective values $c_4' = 1$ in place of $c_4$ and $M_L' = M_L/c_4$ in place of $M_L$ .

Thus to calculate using any convenient simulation model the long-term energy delivered by a system without an auxiliary heat source (or the net delivered energy from a system with an auxiliary heat source), we could assume the following:

An area of collector

$$A_a = c_1 ,$$

with

$$F_R(\tau\alpha)_{e,n} = 1 \quad \text{and} \quad F_R U_L = c_2 ;$$

no heat-exchanger in the collector loop, so that

$$F_R' = 1 ;$$

a fully-mixed solar store (for which $J_s = 1$ and $K_s = 1$) with

$$(UA)_s = c_3 ;$$

and an auxiliary store with

$$(UA)_{AUX} = c_5 ;$$

provided that the draw-off of water is scaled by a factor of $1/c_4$.

Note, however, that if for a system with an auxiliary heater we want to estimate the long-term solar fraction

(12.11) $\qquad f = (Q_L - Q_{AUX})/Q_L$ ,

then a simulation model that assumes a fully mixed tank with an effective value $M_L'$ for the draw-off mass of water will predict an incorrect solar fraction $f'$. From this value $f'$, however, the true solar fraction can be obtained as $f = f'/c_4$.

## 12.6 References

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       "Methods of Testing to Determine the Thermal Performance of Solar
       Domestic Water Heating Systems"
       ASHRAE Publications and Sales Department
       1791 Tullie Circle NE, Atlanta, GA 30329

2      S A Klein and A H Fanney
       "A rating procedure for solar domestic water heating systems"
       ASME Journal of Solar Energy Engineering 105, 430-439 (1983)

3      SRCC Standard 200-82 (Revised Nov. 1984)
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       Systems"
       Solar Rating and Certification Corporation
       1001 Connecticut Ave. NW, Washington, DC 20036

4      "Heat exchanger penalties in the double loop solar water heating
       systems"
       F de Winter
       Solar Energy 17, 335-338 (1975)

5      W F Phillips and R N Dave
       Solar Energy 26, 111-120 (1982)

6      "SDHW short term test simulation using a utilizability based
       correlation model"
       W Spirkl
       Sektion Physik der LMU München
       (Unpublished communication to Task III meeting, Seville, Dec. 1987)

7      W Spirkl
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       23/9/87

8      T A Reddy
       Solar Energy 39, 123-133 (1987)

# 13   COMPUTER SIMULATIONS AND LABORATORY MEASUREMENTS

## 13.1   Introduction

The theoretical basis of the stationary test method outlined in Chapter 12 and Appendix A was developed by Rogers in a series of unpublished papers [1-7]. The original model (of which six slightly different alternative forms were suggested) was a four-parameter model in which the first and third parameters had an implicit dependence on the draw-off mass: hence it was applicable only to performance with a fixed daily draw-off. The method required a no-solar test day corresponding to each of the solar test days, although it was shown in [3] that this could be avoided by adding a fifth parameter.

In September 1986, Rogers proposed a restructured form of the four-parameter model, in which the dependence on draw-off mass was made explicit and the four parameters $c_1$, $c_2$, $c_3$, and $c_4$ were each independent of draw-off. To this Spirkl [8] suggested adding the correlation for losses from the auxiliary heater, firstly to avoid the need for no-solar test days and secondly because identification of the fifth parameter $c_5$ enabled the long-term prediction to be of delivered energy rather than of auxiliary energy savings.

The first validations of this model using computer simulations was reported to the Task III participants by Spirkl [9] in December 1987, just a few days before the formal end of the Task. Although it was not possible therefore to experimentally validate the test method, the assumptions on which it is based have been supported by many computer simulations and experimental measurements undertaken by the participants as a contribution to the work of the Task. These contributions are summarized in the following sections.

## 13.2   Solar Energy Unit, Cardiff

At Cardiff Marshall and Barragan de Ling performed a series of computer simulations of short-term tests to validate the four-parameter model for fixed draw-off. Fourteen different sets of results were reported [10], with combinations of good and bad collectors with well- and poorly-insulated storage tanks, both fully-mixed and stratified. The parameters identified from the simulated test data generally agreed to within two standard errors of estimation with the values predicted from the component parameters. (Note, however, that the derivation of the parametric models from the correlation of Klein and Fanney does not provide a means for predicting the value of $J_s$ from component parameters.)

On the basis of their results Marshall and Barragan de Ling showed that the form of the model which best fitted the data from among the six alternatives was the fixed-draw equivalent of the present model. The standard error of prediction with this model was in every case less than 1 MJ per day.

Recently Marshall and Barragan de Ling have performed a second series of test simulations to validate the model with a variable draw-off mass. They reported [11] that again the fitted parameters agreed well with the values expected from the component parameters on which the simulations were based, and that the standard error of prediction of the model was less than 1 MJ/day.

A series of experimental validations of the method have begun at Cardiff, using the solar simulator to simulate daily irradiance profiles. Results from these tests are not yet available, however.

## 13.3  Solar Calorimetry Laboratory, Kingston, Ontario

At the Solar Calorimetry Laboratory the work of the Task has been supported both by testing in the laboratory and by simulation studies of short-term tests using the WATSUN simulation program from the University of Waterloo.

The work at Kingston has shown that under a wide range of test conditions the performance of a system in the short-term tests can be accurately represented by a simple multilinear model in which the delivered energy is linearly dependent on the solar energy incident on the collectors and on the difference between the collector ambient temperature and the mains water temperature. The parametric model expressed by Equation (12.1) is an extension of this so-called 'input-output' model to include the case when not all the solar energy is utilizable.

A large number of test simulations performed at the Solar Calorimetry Laboratory in November 1987 confirmed both the validity of the input-output model when all solar energy is utilizable and the non-linearity which results from effects such as changes in day length which affect the utilizability. Parametric fits to these data again showed standard errors of prediction of less than about 1 MJ/day.

Experimental trials of the test method are not complete, but some initial data are shown in Figures 1 and 2 on the following pages. In Figure 1 experimental points are compared with simulated data for the same values of mains water temperature and collector ambient temperature. The straight-line dependence is obvious, but the agreement between the simulated and experimental data is poor. Figure 2 shows that this discrepancy is well within the range due to uncertainty in the degree of stratification in the store.

## 13.4  Studsvik Energy

As a contribution to the validation of the test methods Perers and Walletun selected seven days of stationary performance from data monitored at the Swedish National Test Facility in Borås [12]. The data did not, however, fit the four-parameter model for fixed draw-off with sensible results. Perers and Walletun concluded that a new data base would be needed to test the models for the Swedish climate.
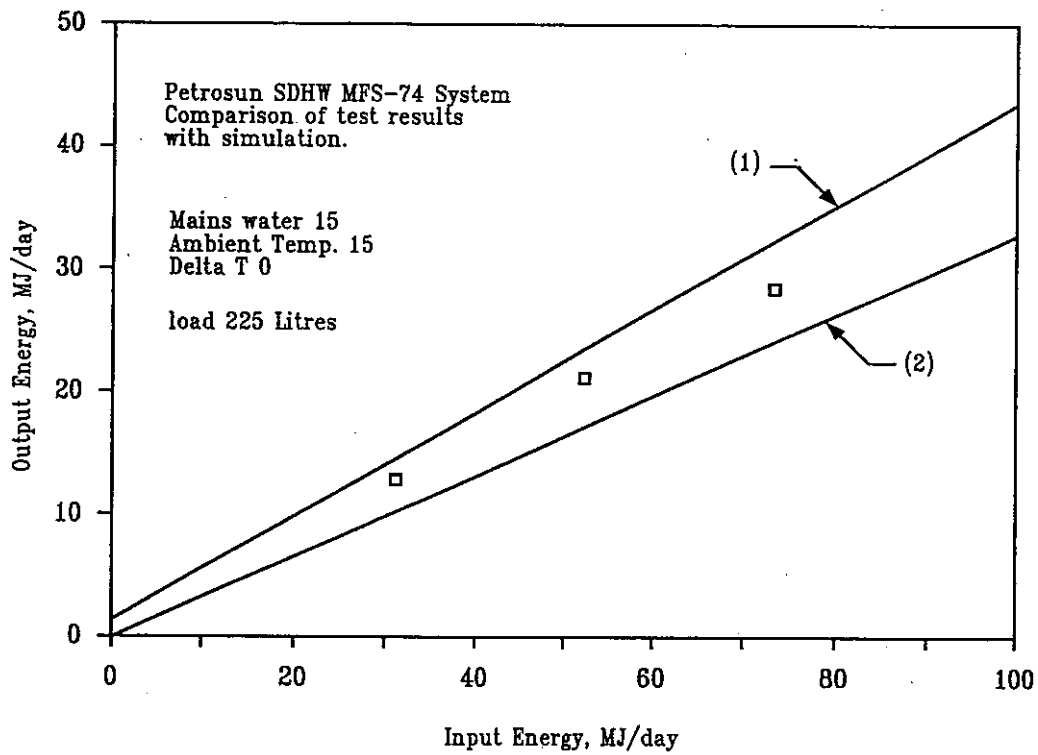
## 13.5  TNO Institute of Applied Physics

A large body of simulated data including long-term simulations was contributed from TNO-TPD at Delft. These data were analysed by Spirkl at Munich, and they helped to determine the conditions which test data needed to satisfy.

Petrosun SDHW MFS-74 System
Comparison of test results
with simulation.

Load 225 Litres
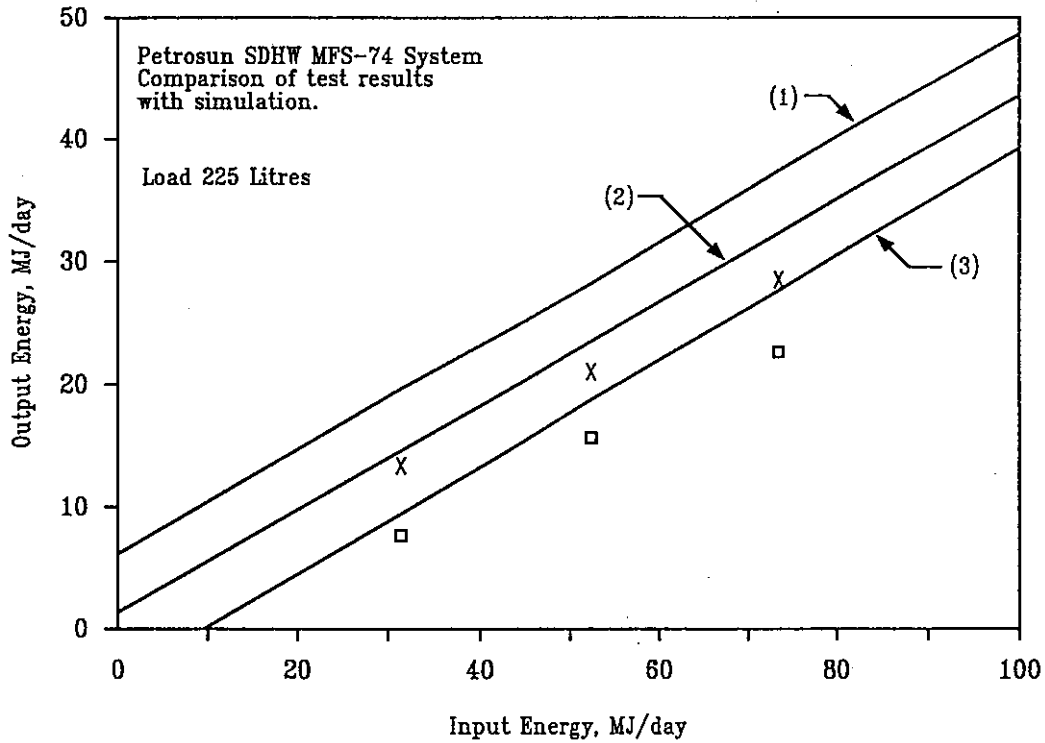
|  | Tmains | Tambient | Delta T | Efficiency (%) |
|---|---|---|---|---|
| (1) Simulated test | 5 | 15 | -10 | 42.4 |
| (2) Simulated test | 15 | 15 | 0 | 42.2 |
| (3) Simulated test | 25 | 15 | 10 | 43.3 |
| X Actual test points | 15 | 15 | 0 | 35.9 |
| □ Actual test points | 25 | 15 | 10 | 35.6 |

Figure 1

Petrosun SDHW MFS-74 System
Comparison of test results
with simulation.

Mains water 15
Ambient Temp. 15
Delta T 0

load 225 Litres

(1)

(2)

Output Energy, MJ/day

Input Energy, MJ/day

(1) Stratified tank simulation

(2) Mixed tank simulation

□ Actual test points

Figure 2

## 13.6 University of Munich

At the University of Munich valuable contributions to the development of the stationary method were made by Spirkl, in private communications as well as the working paper [9] - also referred to in Chapter 12. After considering the statistical aspects of the data analysis, for example, Spirkl suggested improvements in the experimental design and proposed the method of data analysis now recommended.

Recently [13] Spirkl proposed a dynamic test procedure for SDHW systems which generalizes the present method to non-stationary conditions. The dynamic model has a storage heat-capacity term to account for the net carry-over of stored energy from one day to the next, and a filter method is used to identify the model parameters from a continuous sequence of test data. The dynamic test avoids the need to achieve stationarity, which takes two or three actual days for each test day, and it could be used outdoors to test systems *in situ*.

A working group has been formed from among interested participating countries to assist in the development and validation of this new approach.

## 13.7 References

1       "Correlations and simplified system models as a basis for SDHW system testing"
        An informal discussion document for the Task III experts' meeting on 2nd - 6th June, 1986
        B A Rogers
        March 1986

2       "Proposal for a SDHW system test"
        An informal discussion document for the Task III experts' meeting on 2nd - 6th June, 1986
        B A Rogers
        March 1986

3       "Evaluation of alternative models as bases for SDHW system testing"
        B A Rogers
        July 1986

4       "Program to analyse SDHW data"
        B A Rogers
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# APPENDIX A    THEORY OF STATIONARY SYSTEM TEST METHOD

## A.1    Types of system considered

The systems we shall consider are illustrated in Figures 1, 2 and 3 following this Appendix. Figure 1 shows a solar preheat tank, or 'solar store', with no auxiliary heating, Figure 2 shows a system consisting of a solar preheat tank and a separate tank with an auxiliary heater, and Figure 3 shows a system with a combined tank that is heated by solar energy in the bottom portion and by an auxiliary heater at the top.

When we consider the system illustrated in Figure 3 it is convenient to imagine the combined tank divided into two separate stores and to use the terms 'solar store' and 'auxiliary store' to refer to the appropriate sections of the combined tank. By using such simple conventions of terminology it is possible to discuss the three systems interchangeably.

## A.2    Energy-balance equations for stationary performance

### A.2.1 Collector-storage loop

The daily energy supplied to the solar store is

(A.1)        $Q_u = A_a (F'_R/F_R) F_R (\tau\alpha)_{e,n} \Delta t \sum [(I_T K_{\tau\alpha}) - c_2 (T_i - T_a)]^+$ ,

where

(A.2)        $c_2 = F_R U_L / F_R (\tau\alpha)_{e,n}$

is a constant system parameter, and where $F'_R/F_R$ is the de Winter collector-heat exchanger penalty factor [1]. (For the present $T_i$ and $T_a$ have incremental values rather than the daily average values.)

In order to make use of a correlation due to Klein and Fanney [3], we first express $Q_u$ in terms of the mean store temperature $T_s$ instead of $T_i$. We therefore write

(A.3)        $Q_u = A_a (F'_R/F_R) K_s F_R (\tau\alpha)_{e,n} \Delta t \sum [(I_T K_{\tau\alpha}) - c_2 (T_s - T_a)]^+$ ,

where $K_s$ is the Phillips and Dave stratification coefficient [4],

(A.4)        $K_s = [I_T K_{\tau\alpha} - U_L (T_i - T_a)/(\tau\alpha)_{e,n}]/[I_T K_{\tau\alpha} - U_L (T_s - T_a)/(\tau\alpha)_{e,n}]$ .

$K_s$ depends on the collector-store heat-exchanger effectiveness and the thermal-capacity flowrate through the store, but we shall otherwise take it to be constant for the system. The use of $K_s$ assumes that only for a negligible number of time increments will we have

but

$I_T K_{\tau\alpha} - U_L (T_i - T_a)/(\tau\alpha)_{e,n} > 0$ ,

$I_T K_{\tau\alpha} - U_L (T_s - T_a)/(\tau\alpha)_{e,n} < 0$ .

Secondly, we assume daily-averaged values for $T_s$ and $T_a$ in Equation (A.4). Finally, therefore, we have

(A.5) $\qquad Q_u = (c_1 \Delta t) \sum [(I_T K_{\tau\alpha}) - c_2 (T_s - T_a)]^+$ ,

with

(A.6) $\qquad c_1 = A_a (F_R' / F_R) K_s F_R (\tau\alpha)_{e,n}$

another constant system parameter.


A.2.2 Solar store

The daily energy balance for the solar store requires

(A.7) $\qquad Q_u = Q_S + Q_{LOSS,s}$ ,

where $Q_S$ is the daily energy delivered from the solar store, and $Q_{LOSS,s}$ is the daily thermal loss from the solar store.

(In general, there would be another term representing the difference in carry-over energies between one day and the next, but by definition this is zero for stationary conditions.)

$Q_{LOSS,s}$ is expressed as

(A.8) $\qquad Q_{LOSS,s} = c_3 D (T_s - T_a^s)$ ,

where D is the day length (24 hours). The parameter

(A.9) $\qquad c_3 = (UA)_s$

is regarded as a third constant system parameter.

Combining Equations (A.5), (A.7) and (A.8), we have

(A.10) $\qquad Q_S + c_3 D (T_s - T_a^s) = (c_1 \Delta t) \sum [(I_T K_{\tau\alpha}) - c_2 (T_s - T_a)]^+$ .

For a system without an auxiliary heater, $Q_S$ is identical to $Q_L$, the daily energy delivered by the system.


A.2.3 Auxiliary-energy store

An energy balance for the auxiliary store requires

(A.11) $\qquad Q_L - Q_{AUX} = Q_S - Q_{LOSS,AUX}$ .

We shall assume that

(A.12) $\qquad Q_{LOSS,AUX} = c_5 D (T_w - T_a^s)$ .

where

(A.13) $\qquad c_5 = (UA)_{AUX}$

is another system parameter.

Hence for the auxiliary store we have the equation

(A.14)    $Q_L - Q_{AUX} = Q_S - c_5 D(T_w - T_a^s)$ .

## A.3   Correlations

Using TRNSYS simulations of the performance of a solar water-heating system under the stationary conditions of an ANSI/ASHRAE Standard 95-1981 test, Klein and Fanney [3] found a straight-line correlation between the solar fraction (ignoring parasitic energy),

$(Q_L - Q_{AUX})/Q_L$ ,

and the ratio of the utilizable energy to the load,

$F_R Q_u / F_R' K_s Q_L$ .

It follows from this observation that we can write

(A.15)    $Q_L - Q_{AUX} = m(F_R/F_R' K_s)Q_u + cQ_L$ ,

with m and c independent of the daily total radiation.

Now, using Equations (A.7) and (A.8) to substitute for $Q_u$, we can rewrite Equation (A.15) in the form

(A.16)    $Q_L - Q_{AUX} = (mF_R/F_R' K_s)[Q_S + c_3 D(T_s - T_a^s)] + cQ_L$ ,

If we then substitute this value for $Q_L - Q_{AUX}$ in the right-hand side of Equation (A.11) and rearrange terms, we get

(A.17)    $T_s - [(F_R' K_s/mF_R - 1)/c_3 D]Q_S = T_a^s - (F_R' K_s/mF_R)(cQ_L + Q_{LOSS,AUX})/c_3 D$ .

Now on the assumption that the losses from the auxiliary store are given by Equation (A.12) and that $T_w \approx T_{set}$, the right-hand side of this equation is independent of the daily total radiation; and so therefore is the left-hand side. But if we reduce the daily total radiation just to the point where $Q_S = 0$ (which we can always do when the water supply temperature $T_m$ is greater than the ambient temperature of the store $T_a^s$), then at this point we have $T_s = T_m$. Hence, in general,

(A.18)    $T_s - [(F_R' K_s/mF_R - 1)/c_3 D]Q_S = T_m$ .

If the solar store were fully mixed and the draw-off uniform over the day, we would simply have

$Q_s = M_L c_p (T_s - T_m)$ (fully-mixed store).

Hence we shall write in general

(A.19)    $Q_s = J_s M_L c_p (T_s - T_m)$ ,

where, from Equation (A.18),

(A.20)    $J_s = [(UA)_s D/M_L c_p]/[(K_s F_R'/mF_R) - 1]$

$J_s$ - defined by Equation (A.19) as the ratio of the energy delivered from the solar store to the energy which would be delivered from the solar store if it were fully mixed - acts as a second stratification coefficient, analogous to $K_s$. Whereas $K_s$ accounts for the effect of stratification draw-off on the energy collected, however, $J_s$ accounts for their effects on the solar output.

In general, we should expect $J_s$ to depend on the ratio of the daily draw-off of hot water to the tank capacity, but computer simulations have shown that $J_s$ is insensitive to draw-off. When the ratio of draw-off to tank capacity is one, we should expect $J_s > 1$ if the tank is stratified and $J_s = 1$ if it is fully mixed.

The assumption of Equation (A.19) with a constant $J_s$ cannot be justified near to saturation, when the store temperature reaches the demand temperature, but conditions of saturation can easily be avoided in the short-term test. The assumption may also be invalid when the solar contribution to the output from the solar store is small compared with the ambient gains. The error in this case, however, is likely to be small in terms of the total energy.

Thus we have as a correlation for the mean storage temperature

(A.21)     $T_s = T_m + c_4 Q_s / M_L c_p$ ,

where the parameter

(A.22)     $c_4 = 1/J_s$

is our final system parameter.

Substituting (A.21) in (A.10) then gives

(A.23)     $(1 + c_3 c_4 D / M_L c_p) Q_s + c_3 D(T_m - T_a^s)$

$$= (c_1 \Delta t) \sum [(I_T K_{\tau\alpha}) - c_2 \{(T_m - T_a) + c_4 Q_s / M_L c_p\}]^+ ,$$

which together with Equation (A.14) defines the stationary model.

## A.4     References

1     "Heat exchanger penalties in the double loop solar water heating systems"
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2     ANSI/ASHRAE Standard 95-1981
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3     S A Klein and A H Fanney
"A rating procedure for solar domestic water heating systems"
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4     W F Phillips and R N Dave
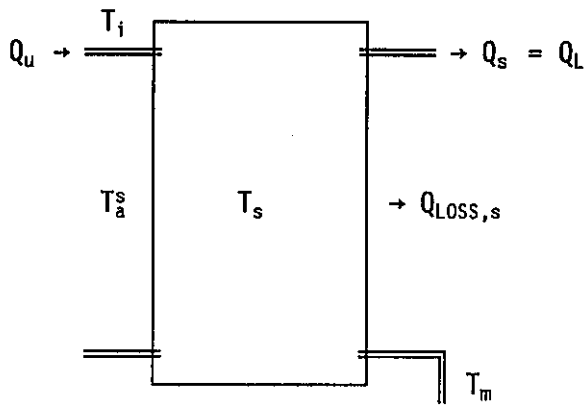Solar Energy 26, 111-120 (1982)

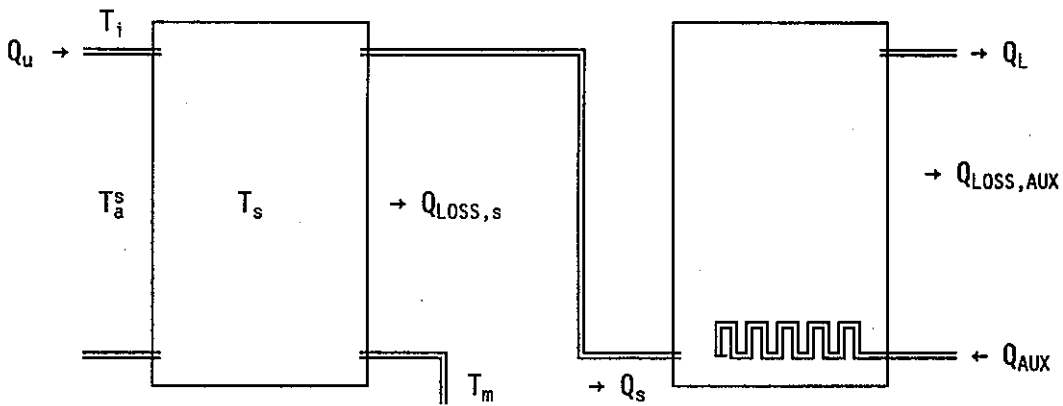FIGURE 1: SOLAR PREHEAT TANK WITH NO AUXILIARY HEATER



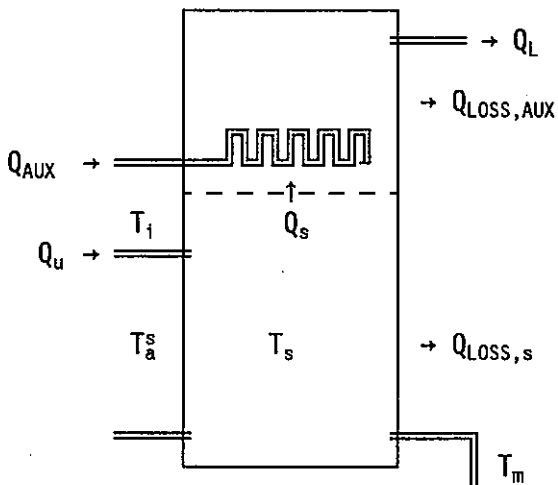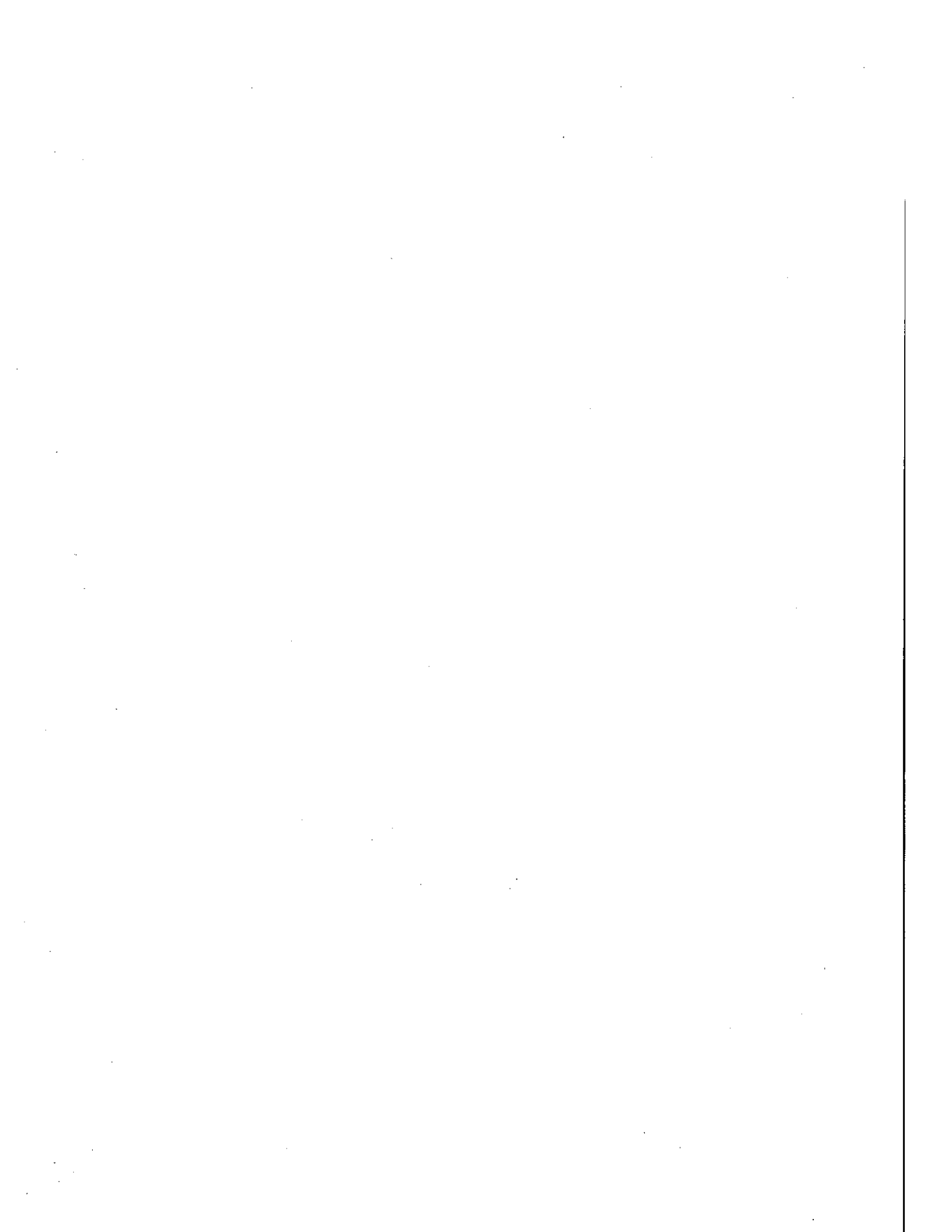FIGURE 2: SOLAR PREHEAT TANK WITH SEPARATE AUXILIARY-HEATING TANK



FIGURE 3: SOLAR PREHEAT TANK COMBINED WITH AUXILIARY HEATER

This report is part of the work of the
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within

Task III :    Performance Testing of Solar Collectors

Subtask E :   Development of a Capability to Evaluate

Domestic Hot Water System Performance

using Short - Term Test Methods