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Method of Testing for Rating Thermal Storage Devices Based on Thermal Performance

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ABSTRACT

A study has been made at the National Bureau of Standards of the different techniques that could be used for testing thermal storage devices and rating them on the basis of thermal performance. This document outlines a proposed standard test procedure based on that study. It is written in the format of a standard of the American Society of Heating, Refrigerating, and Air Conditioning Engineers and specifies the recommended apparatus, instrumentation, and test procedure.

Key Words: Standard test; thermal storage, thermal performance, solar energy, standard, thermal test

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SECTION 1. PURPOSE

1.1 The purpose of this standard is to provide a standard procedure for determining the thermal performance of thermal energy storage devices that are used in systems to provide the thermal requirements for heating, cooling, and the generation of domestic hot water in buildings.

SECTION 2. SCOPE

2.1 This standard applies to sensible-heat and latent-heat type thermal energy storage systems. In addition, it is limited to those storage devices in which a fluid enters the device through a single inlet and leaves the device through a single outlet. Storage devices having more than one inlet and/or outlet may be tested according to this standard, but each flow configuration involving a single inlet and a single outlet must be tested separately. This standard is not applicable to those configurations in which there is simultaneous flow into the storage device through more than one inlet or simultaneous flow out of the storage device through more than one outlet. The fluid can be either a gas or liquid but not a mixture of the two.

2.2 This standard does not address factors relating to cost or consideration of requirements for interfacing with a specific heating and cooling system. Consequently, the test results do not provide all the information required for a complete evaluation of the thermal energy storage system.

2.3 The test procedure and equipment outlined in this standard are most easily adaptable to thermal energy storage system having capacities on the order of 10^9 J (10^6 Btu) or less.

SECTION 3. DEFINITIONS

3.1 AMBIENT AIR

Ambient air is the air in the space surrounding the thermal energy storage system.

3.2 CYCLING

Cycling of a latent-heat type storage device is a process in which the temperature of the system is raised and lowered in a cyclic manner and the phase of the storage medium is changed twice in each temperature cycle.

3.3 EFFECTIVE CAPACITY FOR HEAT REMOVAL

The effective capacity for heat removal is the amount of heat that can be removed from the storage system during a period of time equal to the fill time and for a specific set of values for t_i, the initial temperature of the storage system; Δt, the temperature difference between t_i and the temperature of the entering fluid; and $\dot{m}$, the mass flow rate of the fluid through the storage system.

3.4 EFFECTIVE CAPACITY FOR HEAT STORAGE

The effective capacity for heat storage is the amount of heat that can be stored in the storage system during a period of time equal to the fill time and for a specific set of parameters t_i, Δt and $\dot{m}$.

3.5 FILL TIME

The fill time is the duration of a single transient test as specified in Section 8 in which energy is either added or extracted from the storage system.

3.6 RATE OF HEAT LOSS

The heat loss rate is the rate that heat is lost from the storage system per degree temperature difference between the storage medium temperature and the average ambient air temperature.

3.7 STORAGE CAPACITY

The storage capacity of a thermal energy storage system is defined as the heat that can be stored in a system undergoing a Δt increase in temperature from its initial value t_i.

3.8 PERFORMANCE COEFFICIENT FOR HEAT REMOVAL

The performance coefficient for heat removal is the ratio of the effective capacity for heat removal to the amount of heat that could be removed from an equal volume of water in an ideal water tank under the same conditions.

3.9 PERFORMANCE COEFFICIENT FOR HEAT STORAGE

The performance coefficient for heat storage is the ratio of the effective capacity for heat storage to the amount of heat that could be stored in an equal volume of water in an ideal water tank under the same conditions.

3.10 STANDARD AIR

Standard air is air weighing 1.2 kg/m^3 (0.075 lb/ft^3), and is equivalent in density to dry air at a temperature of 21.1°C (70°F) and a barometric pressure of 1.01×10^5 N/m^2 (29.92 in. of Hg.).

3.11 STANDARD BAROMETRIC PRESSURE

1.01×10^5 N/m^2 (29.92 in. of Hg.).

3.12 STORAGE MEDIUM

The storage medium is the material in the storage system in which the major portion of the energy is stored.

3.13 STORAGE SYSTEM

The storage system is defined as the container(s) plus all contents of the container(s) used for storing thermal energy in a system. The transfer fluid and other accessories such as heat exchangers within the thermal storage container(s) are considered as part of the storage system.

3.14 SPECIFIC HEAT

The specific heat of a substance is the quantity of energy necessary to produce a unit change in temperature of a unit mass.

3.15 TRANSFER FLUID

The transfer fluid is the fluid that carries energy in and out of the storage system.

SECTION 4. CLASSIFICATIONS

4.1 In this standard thermal energy storage systems are classified according to the method they use to store energy and the type of transfer fluid they employ.

4.1.1 Sensible heat storage systems are those in which the heat absorbed by or removed from the system results in an increase or decrease in the temperature of the storage medium and there is no change of phase of any portion of the storage medium. Typical systems employ pressurized water, unpressurized water, rock, brick or concrete as the storage medium.

Latent-heat storage systems are those involving a change of phase of the storage medium. In this type of system, most of the heat added to or removed from the system goes into changing the enthalpy of the storage medium during a change of phase process. Some heat is also stored as sensible heat, since charging and discharging of the storage device usually involves a finite change in the temperature of the system.

4.1.2 A storage system will use either a liquid or a gas as the transfer fluid. The most common liquids are water or a water-ethylene glycol solution. The most common gas is air.

SECTION 5. REQUIREMENTS

5.1 This standard covers only those thermal energy storage systems that can be treated as black boxes and that do not alter the phase or composition of the transfer fluid passing through them.

5.2 Latent-heat type storage systems evaluated under this standard shall have been cycled (see definition of cycling) through their change of phase at least 30 times prior to being tested.

5.3 The transfer fluid used in evaluating the performance of a thermal energy storage system shall have a known specific heat that varies by less than $\pm$ 0.5% over the temperature range encountered during a test.

5.4 The room where the testing of the storage system is performed shall have its temperature controlled to the extent that the average ambient air temperature t_a, determined by the average of the four temperatures measured as specified in Section 8.9, varies between extremes by less than $\pm$ 1.0°C ($\pm$ 1.8°F) during a test.

SECTION 6. INSTRUMENTATION

6.1 TEMPERATURE MEASUREMENTS

6.1.1 Temperature measurements shall be made in accordance with ASHRAE Standard 41-66, Part 1 [1].

6.1.2 Temperature Difference Measurements Across the Thermal Storage System. The temperature difference of the transfer fluid across the thermal storage system shall be measured with:

a. Thermopile (air or liquid as the transfer fluid)

b. Calibrated resistance thermometers connected in two arms of a bridge circuit (only when a liquid is the transfer fluid)

6.1.3 The accuracy and precision of the instruments and their associated readout devices shall be within the limits as follows:

	Instrument Accuracy[a]	Instrument Precision[b]
Temperature	$\pm$ 0.5°C ($\pm$ 0.9°F)	$\pm$ 0.2°C ($\pm$ 0.4°F)
Temperature Difference	$\pm$ 0.1°C ($\pm$ 0.2°F)	$\pm$ 0.1°C ($\pm$ 0.2°F)

6.1.4 In no case shall the smallest scale division of the instrument or instrument system exceed 2 1/2 times the specified precision. For example, if the specified precision is $\pm$ 0.1°C ($\pm$ 0.2°F), the smallest scale division shall not exceed 0.25°C (0.5°F).

6.1.5 The instruments shall be configured and used in accordance with Section 7. of this standard.

6.1.6 When thermopiles are used, they shall be constructed in accordance with ANSI Standard C96.1-1964 (R 1969) [2].

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[a] The ability of the instrument to indicate the true value of the measured quantity.

[b] Closeness of agreement among repeated measurements of the same physical quantity.

6.2 LIQUID FLOW MEASUREMENTS

6.2.1 The accuracy of the meter including a calibration, if furnished, shall be equal to or better than ± 1.0% of the measured value.

6.3 RECORDERS AND INTEGRATORS

6.3.1 Strip chart recorders used shall have an accuracy equal to or better than ± 0.5% of the temperature difference and/or voltage measured for each test with the exception of the heat loss rate test. In the test to determine the heat loss rate, the accuracy of the strip chart recorder shall be equal to or better than ± 2.0%.

6.3.2 Electronic integrators used shall have an accuracy equal to or better than ± 1.0% of the measured value for each test with the exception of the heat loss rate test. In the test to determine the heat loss rate, the accuracy of the electronic integrator shall be equal to or better than ± 4.0%.

6.4 AIR FLOW MEASUREMENTS

When air is used as the transfer fluid, air flow rate shall be determined as described in Section 7.

6.5 PRESSURE MEASUREMENTS

6.5.1 Nozzle Throat Pressure. The pressure measurement at the nozzle throat shall be made with instruments that shall permit measurements of pressure to within ± 2.0% absolute and whose smallest scale division shall not exceed 2 1/2 times the specified accuracy [3].

6.5.2 Air Flow Measurements. The static pressure across the nozzle and the velocity pressure at the nozzle throat shall be measured with manometers that have been calibrated and are accurate to within ± 1.0% of the reading. The area of the nozzle shall be determined by measuring its diameter to an accuracy of ± 0.20% in four places approximately 45 degrees apart around the nozzle in each of two planes through the nozzle throat, one at the outlet and the other in the straight section near the radius [3].

6.5.3 Pressure Drop Across the Thermal Storage System. The static pressure drop across the thermal storage system shall be measured with a manometer having an accuracy of 2.49 N/m^2 (0.01 in. of water).

6.6 TIME AND MASS MEASUREMENTS

Time measurements and mass measurements shall be made to an accuracy of $\pm$ 0.20% [3].

SECTION 7. APPARATUS AND METHOD OF TESTING

7.1 AIR AS THE TRANSFER FLUID

The relative position of the thermal energy storage system, the temperature measuring instrumentation, the air flow measuring apparatus and the reconditioning apparatus is shown in Figure B1[a].

7.1.1 Test Ducts. The air inlet test duct, between the air flow measuring apparatus and the thermal energy storage system, shall have the same cross-sectional dimensions as the inlet to the storage device. The air outlet test duct, between the thermal energy storage system and the reconditioning apparatus, shall have the same cross-sectional dimensions as the outlet of the storage device.

7.1.2 Temperature Measurement Across the Storage System. A thermopile shall be used to measure the difference between the inlet air temperature and outlet air temperature of the thermal energy storage system. It shall be constructed from calibrated type T thermocouple wire all taken from a single spool of wire. No extension wires are to be used in either its fabrication or installation. The wire diameter must be no larger than 0.51 mm (24 AWG) and the thermopile shall be fabricated as shown in Figure B2. There shall be a minimum of six junctions in the air inlet test duct and six junctions in the air outlet test duct. These junctions shall be located at the center of equal cross-sectional areas.

[a]The recommended apparatus consists of a closed loop configuration. An open loop configuration is an acceptable alternative provided that the test conditions specified herein can be satisfied.

During all tests, the variation in temperature across the air inlet and air outlet test ducts shall be less than ± 0.5°C (± 0.9°F) at the location of the thermopile junctions. The variation shall be checked prior to testing utilizing instrumentation and procedures outlined in reference [1]. If the variation exceeds the limits above, mixing devices shall be installed to achieve this degree of temperature uniformity. Reference [4] discusses the positioning and performance of several types of air mixers.

The measuring junctions of the thermopile should be located as near as possible to the inlet and outlet of the thermal energy storage system. The air inlet and air outlet ducts shall be insulated in such a manner that the calculated heat loss from these ducts to the ambient air would not result in a temperature change for any test of more than 0.05°C (0.09°F) between the temperature measuring locations and the storage system.

7.1.3 Dry and Wet Bulb Temperature Measurements. Thermocouples or other devices giving a continuous reading shall be used to measure the wet and dry bulb temperature at the locations in the air inlet and air outlet ducts shown in Figure B1. ASHRAE Standard 41-66, Part I [1] shall be followed in making these measurements.

7.1.4 Duct Pressure Measurements. The static pressure drop across the thermal energy storage system shall be measured using a manometer as shown in Figures B1 and B3. Each side of the manometer shall be connected to four pressure taps that are connected to an external manifold on the air inlet and air outlet ducts. The pressure taps should consist of 6.4 mm (1/4 in.) nipples soldered to the duct and centered over 1 mm (0.040 in.) diameter holes. The edges of these holes on the inside surfaces of the ducts should be free of burrs and other surface irregularities [5].

7.1.5 Air Flow Measuring Apparatus. The air flow shall be measured with the nozzle apparatus discussed in Section 7 of ASHRAE Standard 37-69 [3]. As shown in Figure B4, this apparatus consists basically of a receiving chamber, a discharge chamber and an air flow measuring nozzle. The distance from the center of the nozzle to the side walls shall not be less than 1 1/2 times the nozzle throat diameter, and diffusers shall be installed in the receiving chamber at least 1 1/2 nozzle throat diameters upstream of the nozzle and 2 1/2 nozzle throat diameters downstream of the nozzle. The apparatus should be designed so that the nozzle can be easily changed and the nozzle used on each test shall be selected so that the throat velocity is between 15 m/s (2960 fpm) and 35 m/s (6900 fpm). Details on nozzle construction and discharge coefficients that may be used are contained in Section 7.3 of ASHRAE Standard 37-69 [3].

An exhaust fan capable of providing the desired flow rates through the thermal energy storage system shall be installed in the end wall of the discharge chamber. The dry and wet bulb temperature of the air entering the nozzle shall be measured in accordance with ASHRAE Standard 41-66, Part I [1]. The velocity of the air passing through the nozzle shall be determined by either measuring the velocity head by means of a commercially available pitot tube or by measuring the static pressure drop across the nozzle with a manometer. If the latter method is used, one end of the manometer shall be connected to a static pressure tap located flush with the inner wall of the receiving chamber and the other end to a static pressure tap located flush with the inner wall of the discharge chamber, or preferably, several taps in each chamber should be connected through a manifold to a single manometer. A means shall also be provided for measuring the absolute pressure of the air in the nozzle throat.

7.1.6 Air Leakage. Air leakage through the air flow measuring apparatus, air inlet test duct, the thermal energy storage system and the air outlet test duct shall not exceed $\pm$ 1.0% of the measured air flow.

7.1.7 Air Reconditioning Apparatus. The reconditioning apparatus shall control the dry bulb temperature of the air entering the storage system to within $\pm$ 1.0°C ($\pm$ 1.8°F) of the desired test values at all times during the tests. Its heating and cooling capacity shall be selected so that dry bulb temperature of the air entering the reconditioning apparatus may be raised or lowered by an amount equal to the largest required in Section 8.

7.2 LIQUID AS THE TRANSFER FLUID

The test setup for thermal energy storage systems employing liquid as a transfer fluid is shown in Figure B5[a].

7.2.1 Temperature Measurement Across the Storage System. The temperature difference between the transfer fluid entering and leaving the storage system shall be measured using either two calibrated resistance thermometers connected in two arms of a bridge or a thermopile made from calibrated, type T thermocouple wire all taken from a single spool of wire. The thermopile shall contain any even number of junctions constructed according to the recommendations in ANSI Standard C96.1-1964 (R 1969) [2].

[a] The recommended apparatus consists of a closed loop configuration. An open loop configuration is an acceptable alternative provided that the test conditions specified herein can be satisfied.

Each resistance thermometer or each end of the thermopile is to be inserted into a well [6] located as shown in Figure B5. To insure good thermal contact, the wells shall be filled with light oil. The wells should be located just downstream of a right angle bend to insure proper mixing [1].

To minimize temperature measurement error, the wells should be located as close as possible to the inlet or outlet of the storage device. In addition, the piping shall be insulated in such a manner that the calculated heat loss from this piping to the ambient air would not cause a temperature change for any test of more than 0.05°C (0.09°F) between each well and the storage system.

7.2.2 Additional Temperature Measurements. The temperature of the transfer fluid at the two locations cited above shall also be measured by inserting appropriate sensors into the wells. ASHRAE Standard 41-66, Part I [1] shall be followed in making these measurements.

7.2.3 Pressure Drop Across the Storage System. The pressure drop across the thermal energy storage system shall be measured using static pressure tap holes and a manometer. The edges of the holes on the inside surfaces of the pipe should be free of burrs and should be as small as practicable but not exceeding 1.6 mm (1/16 in.) diameter [5]. The thickness of the pipe wall should be 2 1/2 times the hole diameter [5].

7.2.4 Liquid Transfer Fluid Reconditioning Apparatus. The reconditioning apparatus shall control the temperature of the transfer fluid entering the storage system to within $\pm$ 1.0°C ($\pm$ 1.8°F) of the desired test values at all times during the tests. Its heating and cooling capacity shall be selected so that the temperature of the liquid entering the reconditioning apparatus may be raised or lowered by an amount equal to the largest required in Section 8.

7.2.5 Additional Equipment. A pressure gauge, a pump and a means of adjusting the flow rate of the transfer fluid shall be provided at the relative locations shown in Figure B5. In addition, a pressure relief valve and an expansion tank should be installed to allow the transfer fluid to expand and contract freely in the apparatus[a].

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[a] Figure B5 should not be interpreted to mean that the relief valve and expansion tank necessarily be located below the thermal energy storage unit.

SECTION 8. TEST PROCEDURE AND CALCULATIONS

8.1 GENERAL

The method that has been most commonly employed in testing of water storage tanks in Japan [11, 12, 13] is to cause the transfer fluid entering the storage device to undergo a step change in temperature and to measure the temperature of the transfer fluid leaving the storage unit. By integrating the difference in temperature between the inlet and outlet over the testing period and multiplying the result by the transfer fluids' mass flow rate and specific heat, one can determine the amount of heat added or removed during this time period[a]. This energy balance is shown conceptually in Figure B6 where the area under the curve represents the energy absorbed during the time period shown. If the time period chosen for the test were some characteristic time depending upon the size of the storage device chosen, the heat storage capability of different devices could be compared. This will be illustrated by citing typical results taken from reference [14].

Yang and Lee [14] performed an analysis to determine the nature of the transient heat transfer between a heat storage unit and a circulating or transfer fluid due to variations in the inlet temperature of the transfer fluid. The configurations chosen for analysis are shown in Figures B7, B8, and B9. Figure B7 shows a specific-heat type storage device in which a liquid storage medium is heated or cooled by a fluid passing through thin tubes inside the container. Figure B8 shows a pebble-bed type unit in which the transfer fluid comes in direct contact with the storage medium. Figure B9 shows as in Figure B7, a heat-exchanger type storage device except in this case, the transfer fluid is circulated around tubes which encapsulate a latent-heat type material such as a salt hydrate.

The basic one-dimensional transient equations governing the temperature distribution of both the transfer fluid and storage medium are presented and solved using the Laplace transformation technique. Yang and Lee point out that the boundary conditions most appropriate to simulate a real storage device would be some arbitrary variation of inlet fluid temperature with time.

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[a] This is strictly true only if the losses from the outside of the storage unit are negligible. Otherwise, the losses must be accounted for in the energy balance.

However, since it is impractical to calculate the system response for every possible inlet variation and since the storage system is described by linear equations, its dynamic characteristics may be conveniently investigated by using a step input or a sinusoidal input. Solutions are given in [14] for both the step input and sinusoidal input for the configurations of Figures B7 and B8 but only for the step input for the latent-heat type device.

Typical results are shown in Figures B10 and B11 for a water tank in which water is also circulated through the heat exchanger as the transfer fluid and the input is a step function. Figure B10 shows the temperature distribution of the transfer fluid as a function of position down the tube and time. Figure B11 shows the same thing for the storage medium or water in the tank. These results were for the case of negligible resistance to heat transfer at the interface between the tubes and the storage medium. One should note that it has been possible to present the results in dimensionless form. The temperature difference between the fluid and the initial value is divided by the difference between the inlet value and the initial value (step function) to get t^*. The space dimension is divided by the total path length through the storage device to get x^* and time has been made dimensionless by dividing it by the time required for a fluid particle to travel through the system of length ℓ, ℓ/u where u is the flow velocity. The process of normalizing the results has been possible due to the fact that the system behavior is described by linear equations. In reality, the response of storage devices will only approximate a linear behavior. Consequently the testing procedure included herein has been written in such a way as to determine the response of the system to different step inputs and for both heat storage and heat removal processes even though the results of the linear theory would indicate this is unnecessary.

To demonstrate how different storage devices can be compared based on their response to a step function input, the results of reference [14] have been used to plot the curves in Figure B12. The curves are plots of dimensionless temperature difference between the inlet and outlet of a storage tank configured as in Figure B7. Both curves are for the same flow rate of water (transfer fluid) through the storage device. The only difference between the two is that on one hand there is a finite resistance to heat transfer on the outside of the pipes (h_o = 56.7 W/($m^2 \cdot {}^oK$) typical of natural convection in the tank and on the other, there is negligible resistance ($h_o = \infty$). The area under the curve is proportional to the amount of energy transferred into or away from the storage unit. As can be seen, the device with the smallest resistance to heat transfer is clearly the more effective one for absorbing or releasing the energy.

Up until this point, emphasis has been placed on discussing the comparison of storage devices based on their response to a step increase in inlet fluid temperature. Other possibilities exist.

A second method that could be employed would be to subject the transfer fluid entering the storage unit to a constant influx of heat, $\dot{Q}$. This would result in raising the temperature of the entering transfer fluid (assuming the specific heat of the transfer fluid is constant) by a fixed number of degrees above the outlet temperature. By measuring the time dependent outlet temperature one could obtain information that would be useful in designing collector-storage systems. While this method simulates more closely the real interaction between a collector and a storage device, it has the disadvantage that one cannot measure the energy storage and removal capability of the unit. This is due to the fact that if one measured the heat absorbed by the storage unit over a period of time, it would just be equal to $\dot{Q}$ x the test period or the amount of energy added to the system. Thus the only way of comparing different storage devices would be to compare plots of outlet temperature versus time for different values of $\dot{Q}$ choosen so as to take into account the different sizes of the storage units being compared. The storage device with the lowest average outlet temperature would probably be considered best because this would tend to maximize the efficiency of a collector.

A third method would be to use a time varying $\dot{Q}$ and to measure the outlet temperature as a function of time during the testing period. This would allow one to simulate the output of a collector over one or more days and to determine the response of the storage device. If the time dependence of $\dot{Q}$ resulted in an oscillating inlet temperature, one would also be able to look at the degree of stratification attained in the storage unit. This method has the same disadvantage as the second method in that it would be very difficult to compare the performance of different storage devices. In addition, one has the problem of deciding on what is the typical cycle for $\dot{Q}$; not an easy task when one considers that the output of the collector depends not only on the weather but on the particular storage unit employed. The major advantage of this method would be that by inserting an array of thermocouples in the storage medium, the experimenter could measure the temperature stratification in the unit.

Stratification, which in water tanks results from different temperature water seeking its own temperature (or density) level, is a desirable characteristic for operation in a solar heating and cooling system since the inlet fluid temperature varies up and down with time. If large stratification results when the inlet temperature is either constant or a monotonic (increasing or decreasing) function of time, it is probably a result of short circuiting flow (ie. dead space in a water tank). This short circuiting of the flow could result in higher fluid temperatures to the collector and thus decreased efficiency, which could easily off-set the advantages of stratification.

To test the transient response of the storage unit, the first method of measuring the response of outlet temperature to a step change in inlet temperature, was chosen as a basis for the test. This method was selected because:

a) it permits the determination of effective storage capacity and thus allows an easy comparison of different types of storage units,

b) it appears to be the most fundamental approach since linear theory shows that the outlet temperature response to a constant or variable heat flux $\dot{Q}$ can be predicted if one knows how the outlet temperature changes with a step change in inlet temperature, and

c) it is felt that the relative performance of storage devices using this method will be the same if either of the other procedures were used.

8.2 STORAGE CAPACITY, FILL TIME, EFFECTIVE CAPACITY FOR HEAT STORAGE, EFFECTIVE CAPACITY FOR HEAT REMOVAL

The storage capacity SC (t_i, Δt) of a thermal energy storage system is defined as the energy that can be stored in a system undergoing a Δt increase in temperature from its initial value t_i. The value SC (t_i, Δt) shall be calculated by adding up the amount of heat absorbed by the individual components in the storage system (storage medium, tank, heat exchanger, insulation, etc.) in raising their temperature from t_i to ($t_i + \Delta t$). If there is an uncertainty in the heat storage ability of the storage medium, a representative sample shall be tested in a calorimeter to determine the amount of heat required to raise its temperature from t_i to ($t_i + \Delta t$).

The concept of fill time as introduced in this test procedure is used for the purpose of specifying the time of the transient tests. Recall that in the analysis of Yang and Lee [14], it was possible to present the response characteristics of the thermal storage unit in terms of a dimensionless time where real time is divided by some characteristic time ℓ/u, the effective path length of the transfer fluid divided by the effective transfer fluid velocity. This allows a convenient way of comparing storage units of the same basic design but of different sizes. However, if one is going to compare, for example, a water storage tank where the transfer fluid and the storage medium are the same with a storage tank and embedded heat exchanger where the volume of water in the heat exchanger (transfer fluid) at any instant is only 1/10 of the volume of the water in the storage tank (storage medium), then comparing the response of the two over the same time period ℓ/u would be unfair. The device with the embedded heat exchanger would appear to have something on the order of 10% of the storage capacity of the other device when compared over the same number of time periods ℓ/u. Consequently, a different time scale is introduced herein that will allow storage units of entirely different designs to be compared on an equitable basis.

If a storage unit has a specified thermal capacity or storage capacity SC (t_i, Δt), and the transfer fluid of specific heat $c_{tf}(t_i)$ is flowing through the device at a constant flow rate $\dot{m}$ and has an inlet temperature Δt, above the initial temperature of the storage device, then the fill time is defined by

$$\tau_F = \frac{SC(t_i, \Delta t)}{\dot{m}\ c_{tf}(t_i)\ \Delta t} \qquad (1)$$

In the testing procedure, all storage units are tested for the same fill time and then the thermal responses compared.

In other words, if two different sensible-heat devices had the same ultimate storage capacity and were being tested over the same Δt but one used water (water tank) and the other air (pebble-bed type) as the transfer fluid, and the flow rates were such that the fluid dwell times (determined by ℓ/u) were identical, it would be unfair to compare the response of the two units over the same real time period. One would be able to charge the water tank with considerably more energy over the same time period (approximately by the ratio of $((\dot{m}\ c_{tf})_{water})/((\dot{m}\ c_{tf})_{air})$. The recommendation here would be to test the two units for the same fill time as defined by equation (1). Consequently, the flow rates would be adjusted so that $((\dot{m}\ c_{tf})_{water})/((\dot{m}\ c_{tf})_{air}) = 1$. In other words, the flow rates for the different devices are adjusted so that the amount of energy entering the device (or leaving for an energy removal test) per unit thermal storage capacity is identical for the two devices.[a]

The fill time τ_F and the flow rate of the transfer fluid $\dot{m}$ are related in an inverse manner according to equation (1). One possibility for specifying the test conditions of the transient tests would be to specify the Δt to be used, allow the experimenter to select an $\dot{m}$, and then, depending upon the properties of the transfer fluid (c_{tf}) and storage capacity of the unit, conduct the tests for some fraction or integral of the fill time. However, the experimenter who was concerned about the optimum performance of his unit would no doubt select a flow rate where the relationship between the energy transfer rate and pressure drop (or power required to push the fluid through the device) was an optimum. In actual installations, the flow rate through and Δt across the storage device is controlled by the flow rate through and Δt across the collector and/or building heating and cooling system. The flow rate is usually proportional to the collector size and in turn the storage capacity is usually proportional to the collector size. Consequently, the ratio of SC/$\dot{m}$ is constant within certain limits which means according to equation (1), the fill time is constant within certain limits. As a result, in the test procedure, two different fill times are specified that are felt to be typical of installed systems, the Δt's are specified according to whether a liquid or air is the transfer fluid[b], and the flow rate to be used is

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[a] One should recognize that for an ideal water tank where there is entirely piston-type flow, the above definition of fill time is identical with the fluid dwell time.

[b] Air heating collectors impose a much higher Δt on the storage device than do liquid heating collectors.

calculated according to equation (1).

The effective capacity for heat storage EC_{hs} $(t_i, \Delta t, \tau_F)$ is defined as the net amount of heat flowing into a storage system during the time period $(\tau_o, \tau_o + \tau_F)$ when the entering transfer fluid, that is initially at temperature t_i, undergoes at Δt step increase in temperature at time τ_o. It may be calculated using the equation:

$$EC_{hs} (t_i, \Delta t, \tau_F) = \dot{m}\, c_{tf} (t_i) \int_{\tau_o}^{\tau_o + \tau_F} \delta t\, (\tau)\, d\tau$$

$$- L\, \tau_F \left[t_i + \frac{\Delta t}{2} - t_a\right] \qquad (2)$$

where $\delta t\ (\tau) = [t_{in}\ (\tau) - t_{out}\ (\tau)]$ is the difference between the inlet temperature t_{in} and the outlet temperature t_{out} at time τ, t_a is the average ambient air temperature, and $\dot{m}$ has the value specified in Section 8.6.

The effective capacity for heat removal EC_{hr} $(t_i + \Delta t, - \Delta t, \tau_F)$ is defined as the net amount of heat removed from a storage system during the time period $(\tau_o', \tau_o' + \tau_F)$ when the entering transfer fluid, which is initially at temperature $(t_i + \Delta t)$, undergoes a Δt step decrease in temperature at time τ_o'. It may be determined using the formula:

$$EC_{hr} (t_i + \Delta t, - \Delta t, \tau_F) = \dot{m}\, c_{tf} (t_i) \int_{\tau_o'}^{\tau_o' + \tau_F} |\, \delta t\ (\tau)\, |\ d\tau \qquad (3)$$

8.3 <u>GENERAL TEST REQUIREMENTS</u>

All of the tests require that the temperature of the storage medium, prior to the start of data taking, be uniform at the desired temperature and that there exist steady flow of the transfer fluid through the storage system during a test. To achieve the former, the transfer fluid shall be circulated through the testing apparatus until steady state conditions are achieved and the inlet and outlet temperatures vary by less than $\pm$ 0.5°C ($\pm$ 0.9°F) during a one hour period. The initial temperature is then defined to be the arithmetic average of the inlet and outlet temperatures. Steady flow of the transfer fluid shall be considered achieved if the flow rate varies by less than $\pm$ 1.0% during a test.

For all tests, with the exception of the one to determine the heat loss rate, the temperature of the storage medium shall remain within the normal operating range of the storage medium. For latent heat-type storage systems this means that the storage medium shall undergo a change of phase as the temperature of the storage medium is both increased and decreased by Δt in the manner discussed in Section 8.1.

During tests involving air as the transfer fluid, the dry bulb temperature shall remain above the dew point temperature, and the inlet dew point temperature shall equal the outlet dew point temperature and both shall remain constant. The latter shall be considered accomplished if the inlet and outlet dew point temperatures vary by less than $\pm$ 0.5°C ($\pm$ 0.9°F) during a test.

8.4 TESTS TO BE PERFORMED

The first test to be performed on a thermal energy storage system is the determination of the heat loss rate, L. The test method for doing this is discussed in Section 8.5. After this is completed, additional tests are required to evaluate EC_{hs} $(t_i, \Delta t, \tau_F)$ and EC_{hr} $(t_i + \Delta t, - \Delta t, \tau_F)$ for a specific t_i and a given pair of parameters τ_F and Δt. The set of parameters τ_F and Δt are to be chosen depending on the type of thermal energy storage system, so that all of the following combinations of the variables τ_F and Δt are tested:

τ_F = 2 hr, 4 hr and Δt = 16°C (28.8°F), 8°C (14.4°F) for thermal energy storage systems using a liquid transfer fluid,

τ_F = 2 hr, 4 hr and Δt = 50°C (90°F), 28°C (50.4°F) for thermal energy storage systems using air as the transfer fluid.

When a phase-change type thermal storage system is being tested that has been designed to be "charged" or "discharged" over a specific time period, this time period shall be used as the fill time for testing in lieu of the above specification.

The value of t_i to be used shall be chosen based on the intended application for the thermal energy storage system. The performance of these tests is discussed in Section 8.6.

8.5 METHOD FOR DETERMINING THE HEAT LOSS RATE

The flow rate of the transfer fluid shall be adjusted to the value (see equation (1)):

$$\dot{m} = \frac{SC\ (t_a,\ 25^\circ C)}{(c_{tf}\ (t_a + 25^\circ C))\ (1\ hr)\ (25^\circ C)} \tag{4}$$

and the temperature of the transfer fluid entering the storage system shall be raised 25 °C (45 °F) above the average ambient air temperature t_a. After the storage system has reached a uniform, steady-state temperature, the average value of δt shall be determined over a one hour period. This average, that will be called $\overline{\delta t}$, is to be obtained by integrating δt over this time period and then dividing by the time period. The rate of heat loss L from the storage system shall then be determined from

$$L = \frac{\dot{m}\ c_{tf}\ (t_a + 25^\circ C)\ \overline{\delta t}}{25^\circ C} \tag{5}$$

8.6 PERFORMANCE OF A TEST INVOLVING A Δt STEP CHANGE IN TEMPERATURE AND A τ_F FILL TIME

The test described in this section is to be performed on both sensible and latent heat-type storage systems. However, because the performance of a latent heat-type storage system is usually affected by its immediate past temperature history, the test herein described shall be performed twice on this kind of storage system. After the test has been completed once, the storage system shall be allowed to reach a uniform temperature and the test is then to be repeated. Only data collected on this second test shall be reduced and reported as discussed in this section and in Section 9.

After the storage medium has reached a uniform initial temperature t_i, the flow rate shall be adjusted to the value:

$$\dot{m} = \frac{SC\ (t_i, \Delta t)}{c_{tf}\ (t_i)\ \Delta t\ \tau_F} \tag{6}$$

The temperature of the transfer fluid entering the storage system shall be increased in a step-like manner to the new value $t_i + \Delta t$ at some time τ_o. During the time period (τ_o, τ_F), the difference between the temperature of the transfer fluid entering and leaving the storage system, $\delta t\ (\tau) = [t_{in}\ (\tau) - t_{out}\ (\tau)]$, shall either be recorded on a strip chart recorder or integrated over time using an electronic integrator. If a strip chart recorder is used, $\delta t\ (\tau)$ shall be manually integrated over time after the test is completed. The integrated value

$$\int_{\tau_o}^{\tau_o + \tau_F} \delta t\ (\tau)\ d\tau$$

plus a knowledge of $\dot{m}$ and $c_{tf}\ (t_i)$ will allow determination of the effective storage capacity $EC_{hs}\ (t_i, \Delta t, \tau_F)$ by means of equation (2).

The last phase of the test involves a step-like decrease in the temperature of the transfer fluid entering the reconditioning apparatus. The temperature of the entering transfer fluid shall be maintained at $t_i + \Delta t$ until the temperature of the transfer fluid leaving the storage system is no longer changing with time. Following this, the temperature of the entering transfer fluid shall be reduced from $(t_i + \Delta t)$ to t_i. Assuming this step change takes place at τ_o', the quantity

$$\int_{\tau_o'}^{\tau_o' + \tau_F} \delta t\ (\tau)\ d\tau$$

is to be evaluated over the time period (τ_o', $\tau_o' + \tau_F$) either by direct electronic integration or by recording δt (τ) on a strip chart recorder and then manually integrating. Equation (3) is then to be used to find the effective heat removal capacity EC_{hr} ($t_i + \Delta t$, $- \Delta t$, τ_F).

8.7 AN INDEPENDENT CHECK ON THE INTEGRAL OF δt

As an independent check, the inlet temperature, t_{in}, and the outlet temperature, t_{out}, of the transfer fluid shall be recorded on strip chart recorders. The quantities

$$\int_{\tau_o}^{\tau_o + \tau_F} \delta t \ (\tau) \ d\tau \quad \text{and} \quad \int_{\tau_o'}^{\tau_o' + \tau_F'} \delta t \ (\tau) \ d\tau$$ shall be calculated using these

recordings and compared with the identical quantities that are obtained by using the primary method which measures $\delta t\ (\tau) = [t_{in}\ (\tau) - t_{out}\ (\tau)]$ directly. In order for a test to be valid, the values obtained on the check must be within $\pm$ 10% of the results obtained using the primary method.

8.8 AIR FLOW RATE CALCULATIONS

The air flow rate through the nozzle is calculated by the following equations:

$$Q_{mi} = 1.41\ C\ A_n\ (P_v\ v'_n)^{0.5} \tag{7a}$$

$$v_n' = 10.1 \times 10^4\ v_n/P_n\ (1 + W_n) \tag{7b}$$

The air flow rate of standard air is then:

$$Q_s = Q_{mi}/(1.2\ v'_n) \tag{7c}$$

8.9 MEASUREMENT OF AMBIENT AIR TEMPERATURE

The ambient air temperature t_a shall be the arithmetic average temperature of the test area, determined by four calibrated, temperature sensors. ASHRAE Standard 41-66, Part I [1] shall be followed in making these measurements. The sensors shall lie in a horizontal plane approximately at the vertical midpoint of the storage system and shall be approximately 0.6 m (23.6 in.) from the sides of the storage system.

8.10 ESTIMATION OF POWER REQUIREMENTS

In order to estimate the power required to move the transfer fluid through the thermal energy storage system, the following equation shall be used:

$$P = \dot{m} \, \Delta P \,/\, \rho \qquad (8)$$

SECTION 9. DATA TO BE RECORDED AND TEST REPORT

9.1 TEST DATA

Table B1 lists the measurements that are to be made during the various tests.

9.2 TEST REPORT

Table B2 specifies the data to be reported in testing a thermal energy storage system. The performance coefficient for heat storage, PC_{hs}, is defined in Table B2 by the formula:

$$PC_{hs} \equiv \frac{EC_{hs}\,(t_i,\ \Delta t,\ \tau_F)/V}{\Delta t\ c_{H_2O}\ \rho_{H_2O}} \qquad (9)$$

where c_{H_2O} and ρ_{H_2O} are the specific heat and density, respectively, of water at the temperature $t_i + \Delta t/2$. The performance coefficient for heat removal is similarly defined by:

$$PC_{hr} = \frac{EC_{hr}\,(t_i + \Delta t,\ -\Delta t,\ \tau_F)/V}{\Delta t\ c_{H_2O}\ \rho_{H_2O}} \qquad (10)$$

The performance coefficients for heat storage and heat removal, as defined above, compare the performance of a thermal energy storage system with the theoretical performance of an ideal water tank of equal volume and having perfect piston-type flow and zero heat loss.

Both PC_{hs} and PC_{hr} are to be calculated for each of the eight tests that determine the effective heat storage and removal capacities for a specific t_i. In addition, a single plot is to be provided showing the time variation in outlet temperature of the transfer fluid for these eight tests. The quantity

$$\frac{t_{out}(\tau) - t_i}{\Delta t}$$

shall be plotted as the ordinate, while the abscissa shall show the time periods $(\tau_o, \tau_o + \tau_F)$ and $(\tau'_o, \tau'_o + \tau_F)$.

SECTION 10. NOMENCLATURE

A_n	area of nozzle, m^2
C	nozzle coefficient of discharge
c_{H_2O}	specific heat of water, J/(kg· °C)
$c_{tf}(t_i)$	specific heat of transfer fluid at t_i, J/(kg·°C)
D	nozzle throat diameter, m
$EC_{hs}(t_i, \Delta t, \tau_F)$	effective capacity for heat storage, J
$EC_{hr}(t_i + \Delta t, - \Delta t, \tau_F)$	effective capacity for heat removal, J
$SC(t_i, \Delta t)$	storage capacity, J
L	heat loss rate, W/°C
$\dot{m}$	mass flow rate of transfer medium, kg/s
PC_{hr}	performance coefficient for heat removal
PC_{hs}	performance coefficient for heat storage
P_n	absolute pressure at nozzle throat, N/m^2
P	estimated power required to move the transfer fluid through the thermal energy storage system, W
P_v	velocity pressure or static pressure difference across nozzle, N/m^2
Q_{mi}	measured air flow, m^3/s

Q_s	standard air flow, m^3/s
t_i	initial temperature of storage system, ^oC
t_{in}	temperature of transfer fluid at inlet, ^oC
t_a	average ambient air temperature, ^oC
t_{out}	temperature of transfer fluid at outlet, ^oC
v_n	specific volume of air at dry and wet bulb temperature conditions existing at the nozzle but at standard barometric pressure, m^3/kg dry air
v_n'	specific volume of air at the nozzle, m^3/kg dry air
V	volume of the storage system, m^3
W	total mass of storage system, kg
W_n	humidity ratio at nozzle, kg H_2O/kg dry air
ΔP	pressure drop across storage system, N/m^2
Δt	step change in temperature, ^oC
$\delta t\ (\tau)$	inlet temperature, t_{in}, minus outlet temperature, t_{out}, ^oC
$\overline{\delta t}$	average of δt (t) during test for L, ^oC
ρ	density of the transfer fluid, kg/m^3
ρ_{H_2O}	density of water, kg/m^3
τ	time
τ_F	fill time, s
τ_o	initial time
τ_o'	time at which temperature of transfer fluid is decreased from $t_i + \Delta t$ to t_i

SECTION 11. REFERENCES

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Table B1 Test Data to be Recorded

Item	Tests Involving Air as the Transfer Medium	Tests Involving a Liquid as the Transfer Medium
Date	X	X
Observer	X	X
Equipment Name Plate Data	X	X
δt (τ) Across Storage System	X	X
Inlet Temperature, t_{in}	X	X
Outlet Temperature, t_{out}	X	X
Inlet Wet Bulb Temperature	X	
Outlet Wet Bulb Temperature	X	
Times	X	X
Liquid Flow Rate		X
Barometric Pressure	X	X
Gauge Pressure at Inlet		X
Gauge Pressure at Nozzle Throat	X	
Nozzle Throat Diameter	X	
Velocity Pressure at Nozzle Throat or Static Pressure Difference Across Nozzle	X	
Dry Bulb Temperature at Nozzle Throat	X	
Wet Bulb Temperature at Nozzle Throat	X	
Pressure Drop Across Storage System	X	X
Ambient Air Temperature	X	X
Initial Temperature t_i	X	X
Step Change Δt	X	X

Table B2 Data to be Reported

General Information

Manufacturer . . . .

Model # . . . .

Serial # . . . .

Storage Medium . . . .

Transfer Fluid . . . .

Weight of Storage System, W . . . .

Volume of Storage System, V . . . .

Normal Operating Temperature Range . . . .

Minimum Transfer Fluid Flow Rate . . . .

Maximum Transfer Fluid Flow Rate . . . .

Maximum Operating Pressure . . . .

Flow Configuration Tested . . . . (picture)

Heat Loss Rate Test

t_a . . . .

SC $(t_a, 25^oC)$ . . . .

$\overline{\delta(t)}$ . . . .

$\dot{m}$ . . . .

$c_{tf}\ (\overline{t_{in}})$ . . . .

$$L = \frac{\dot{m}\ c_{tf}\ (t_{in})\ \overline{\delta t}}{t_{in} - t_a}$$ . . . .

Table B2 (continued)

Transient Tests

t_a . . . .

t_i . . . .

Δt . . . .

τ_F . . . .

$\dot{m}$ . . . .

ΔP . . . .

P . . . .

$c_{tf}\ (t_i)$ . . . .

$SC\ (t_i, \Delta t)$ . . . .

$EC_{hs}\ (t_i, \Delta t, \tau_F)$ . . . .

$EC_{hr}\ (t_i + \Delta t, - \Delta t, \tau_F)$ . . . .

$$PC_{hs} = \frac{EC_{hs}\ (t_i, \Delta t, \tau_F)/V}{\Delta t \quad c_{H_2O}\ \rho_{H_2O}}$$ . . . .

$$PC_{hr} = \frac{EC_{hr}\ (t_i + \Delta t, - \Delta t, \tau_F)/V}{\Delta t \quad c_{H_2O}\ \rho_{H_2O}}$$ . . . . .

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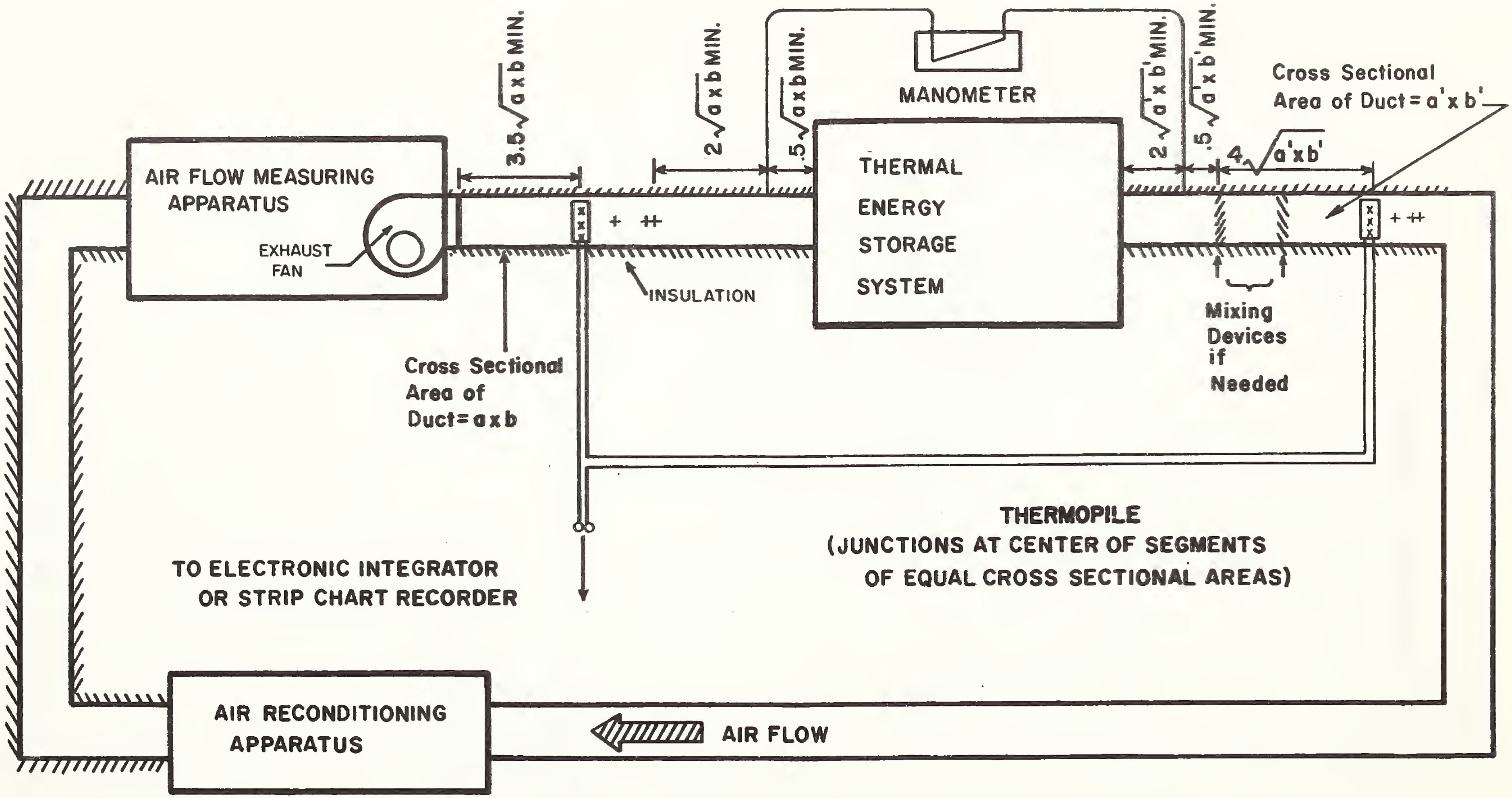


Figure B1 Testing Configuration for the Thermal Storage System When the Transfer Fluid is Air

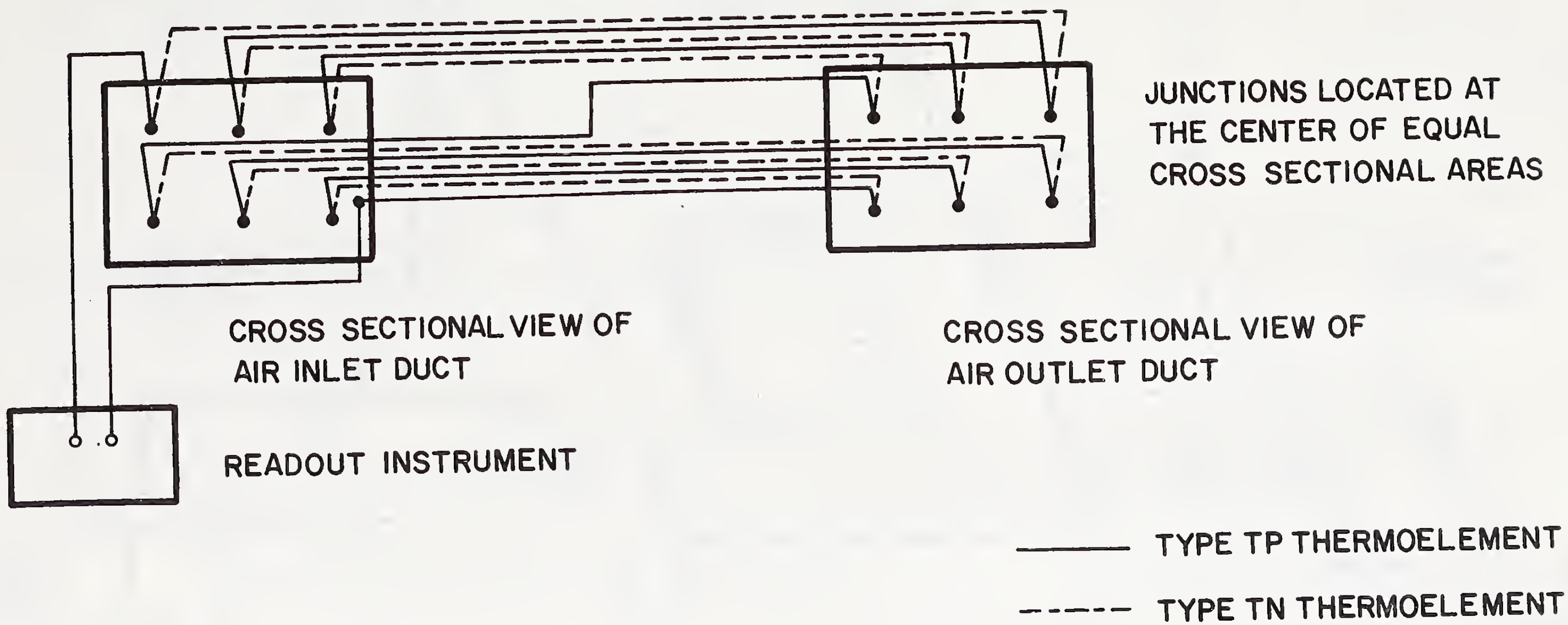


Figure B2 Schematic of the Thermopile Arrangement Used to Measure the Temperature Difference Across the Thermal Storage System

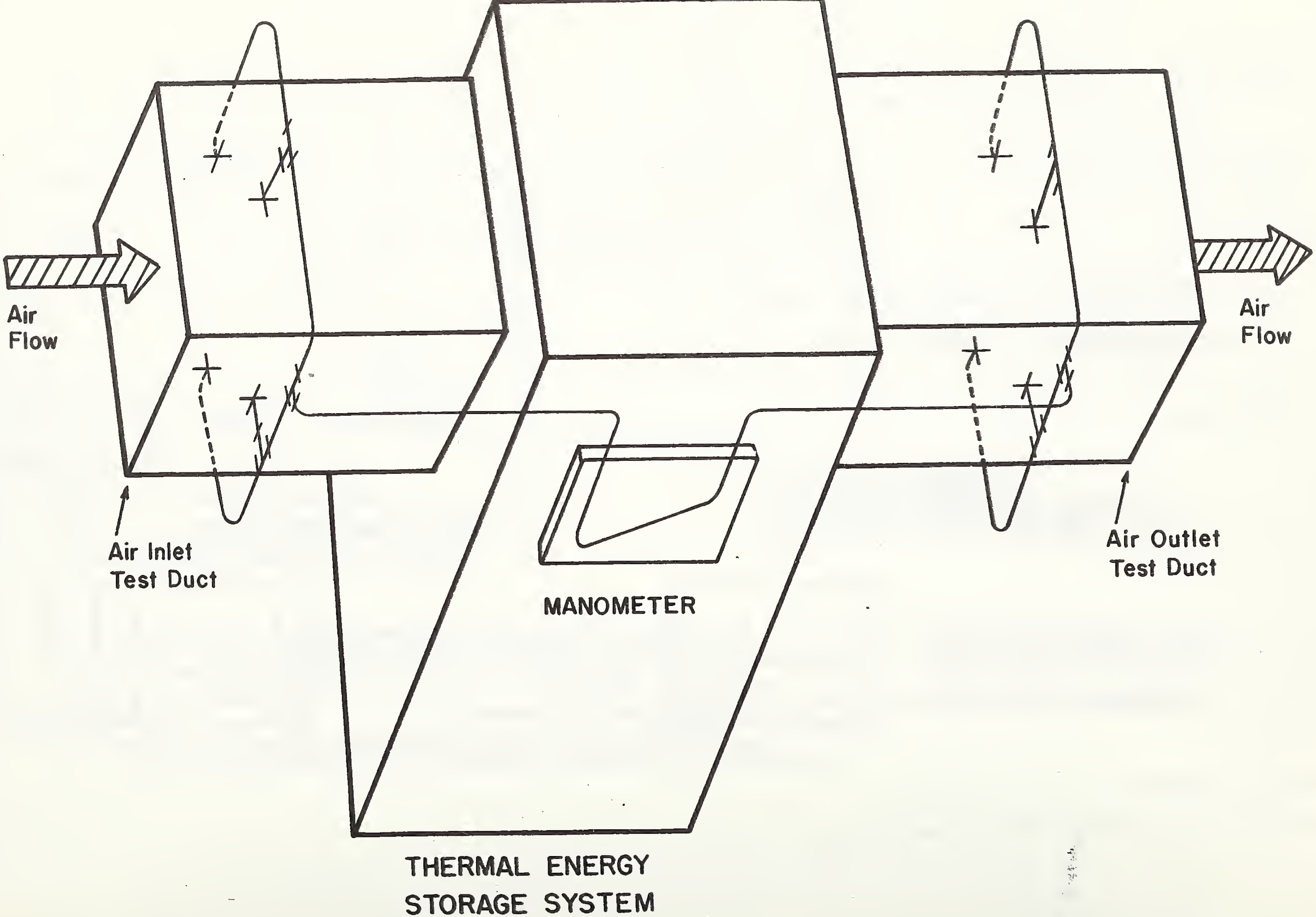
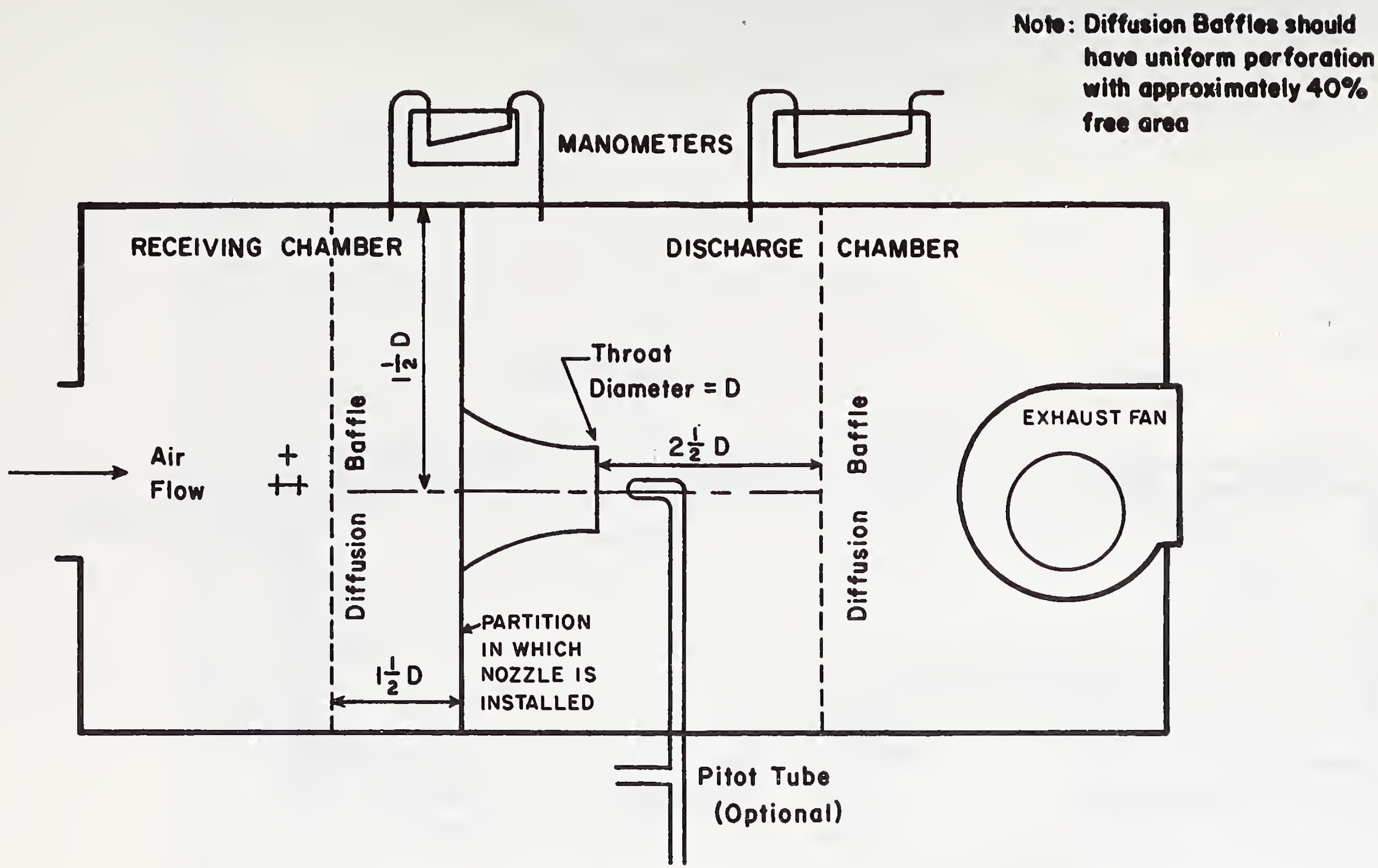


Figure B3 Schematic Representation of the Measurement of Pressure Drop Across the Thermal Storage System When Air is the Transfer Fluid



\+ CALIBRATED THERMOCOUPLE OR THERMISTOR

++ CALIBRATED WET BULB TEMPERATURE MEASURING DEVICE

Figure B4 Nozzle Apparatus for Measuring Air Flow Rate

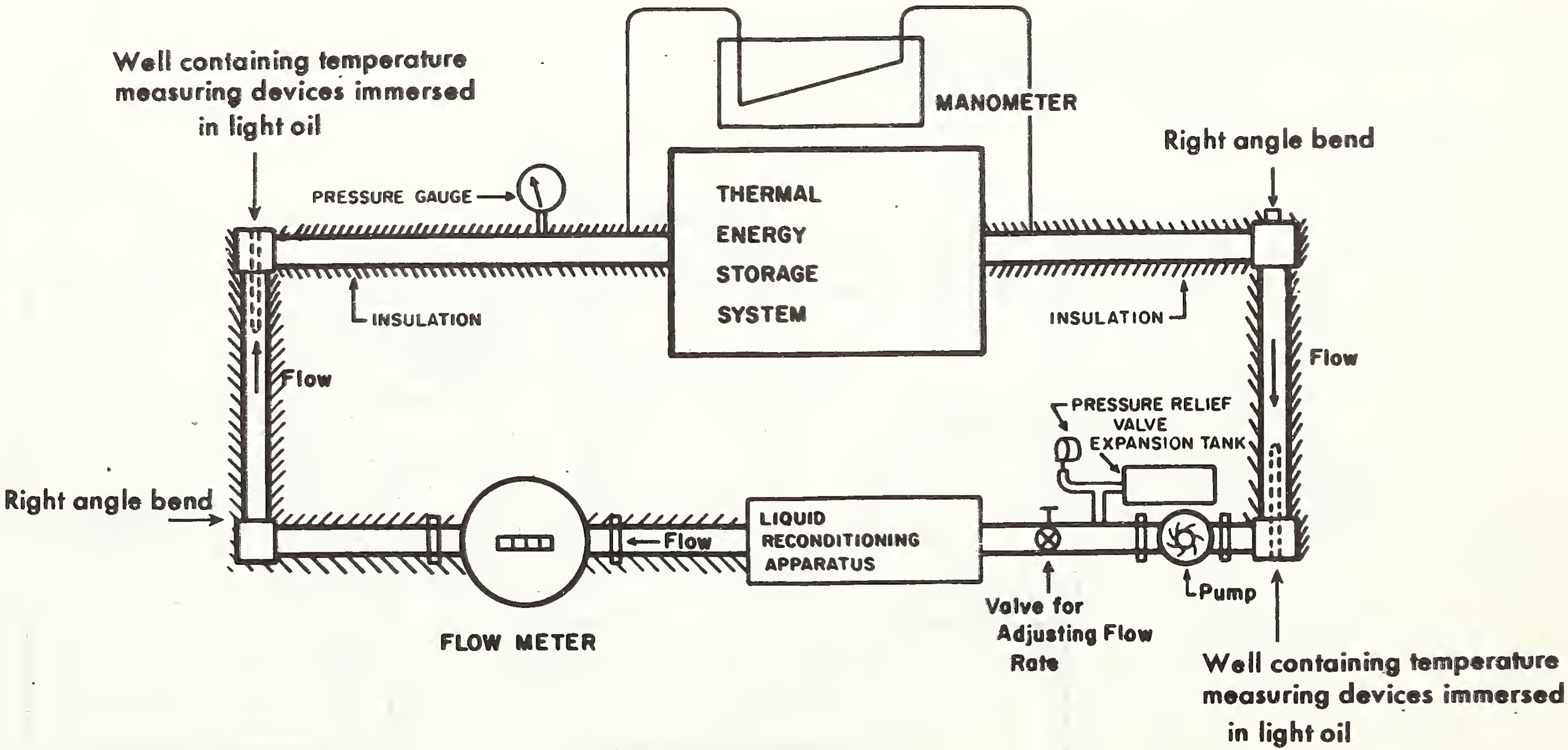


Figure B5 Testing Configuration for the Thermal Storage System When the Transfer Fluid is a Liquid

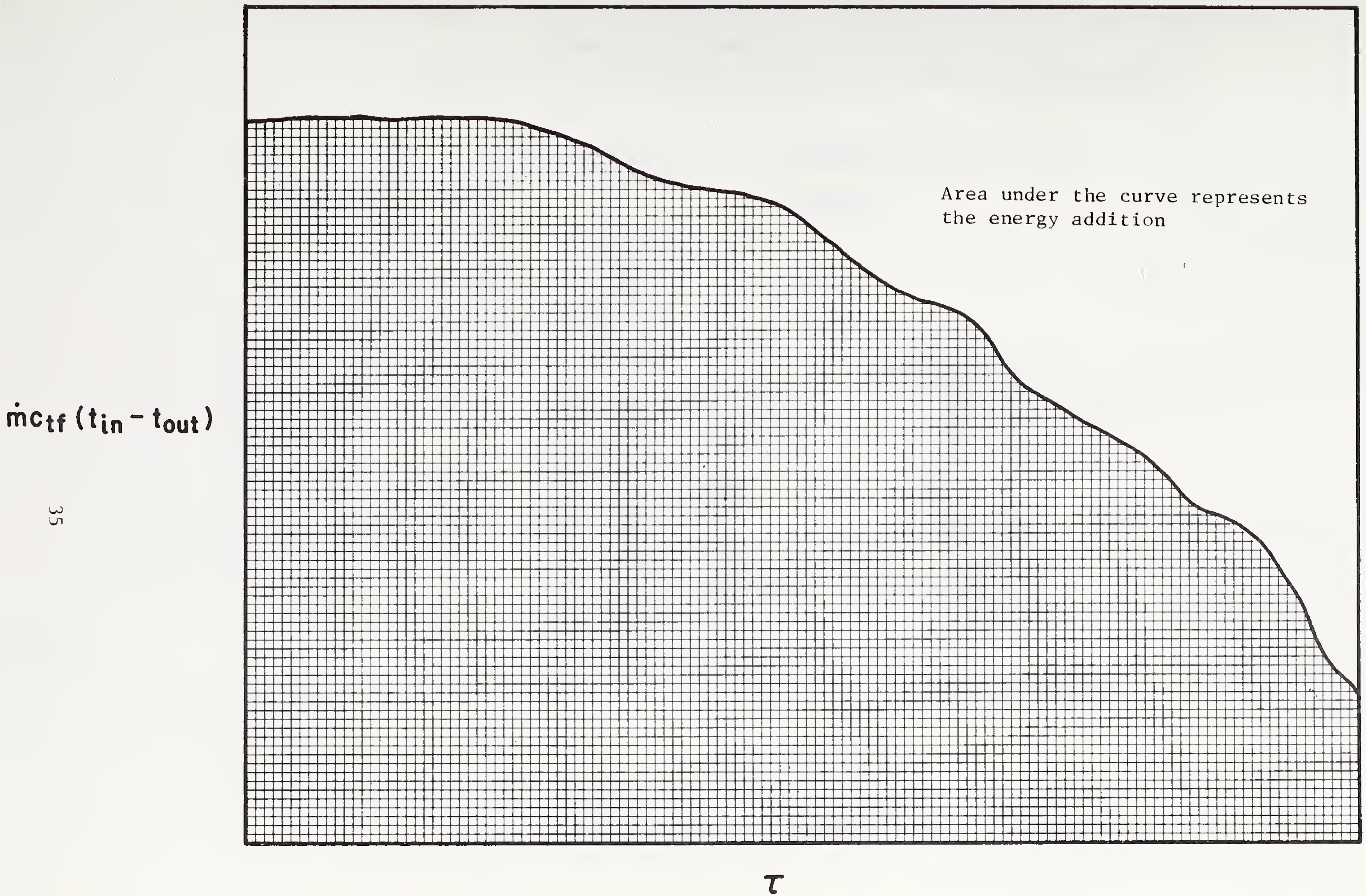


Figure B6 Schematic Representation of Energy Transfer into a Thermal Storage System as a Function of Time When the Inlet Temperature is a Step Function

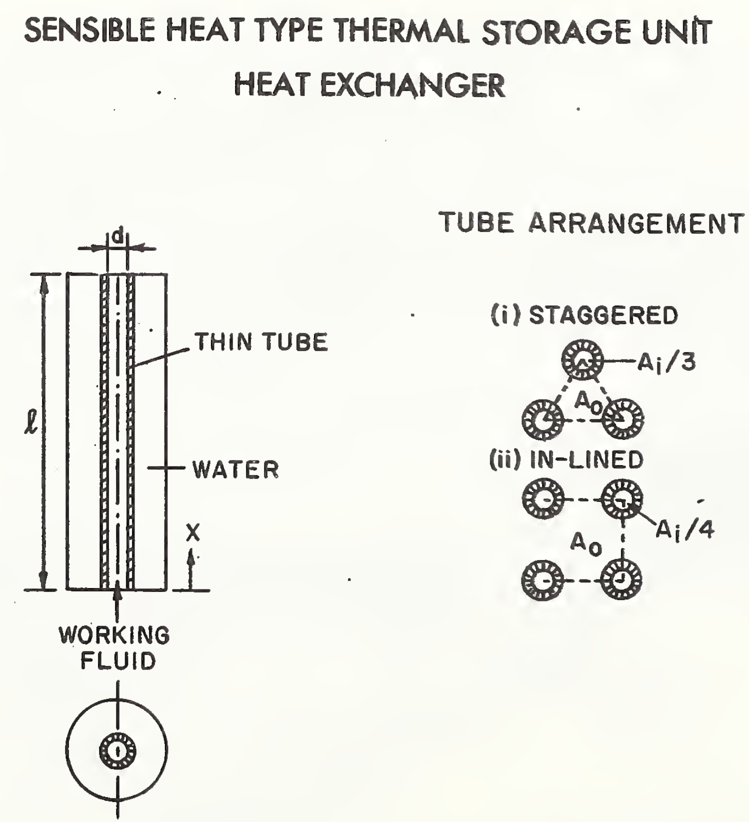


Figure B7 Schematic Representation of a Heat Exchanger-Type Thermal Storage Unit from Reference [14]

SENSIBLE HEAT TYPE THERMAL STORAGE UNIT

PEBBLE BED

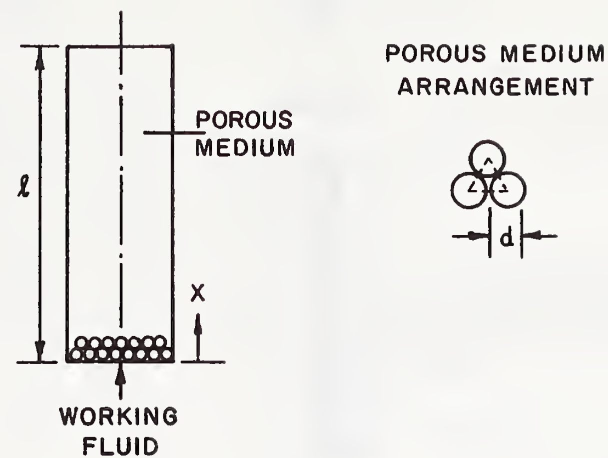


Figure B8 Schematic Representation of a Pebble Bed-Type Thermal Storage Unit from Reference [14]

LATENT HEAT TYPE THERMAL STORAGE UNIT

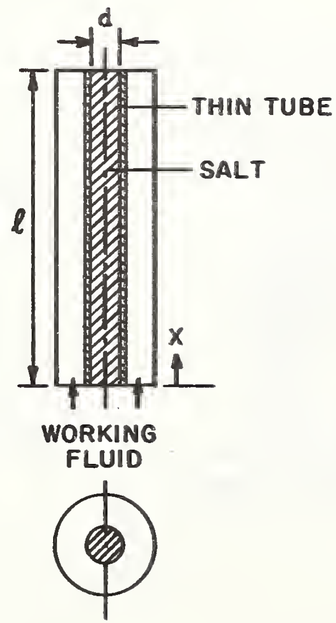
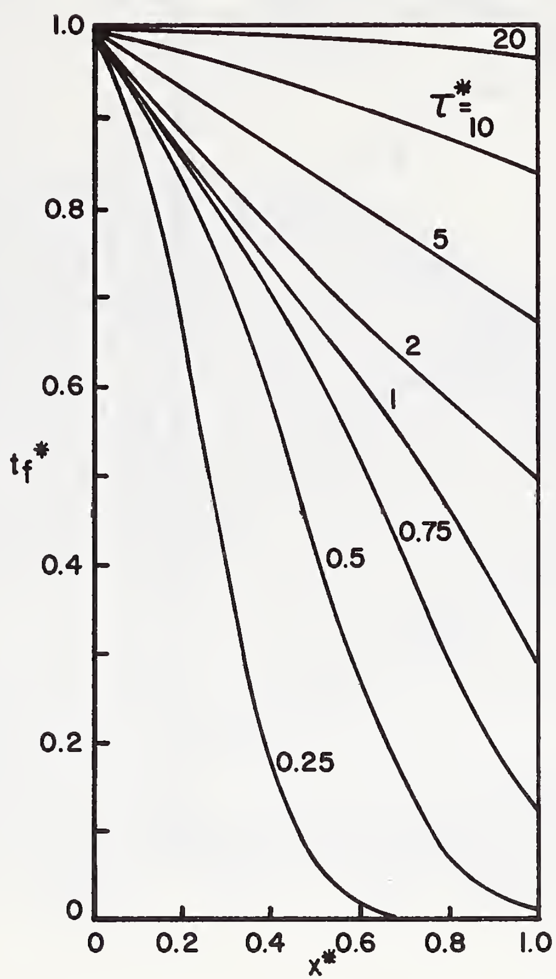


Figure B9 Schematic Representation of a Latent Heat-Type Thermal Storage Unit from Reference [14]

HEAT EXCHANGER TYPE THERMAL STORAGE UNIT
TRANSFER FLUID TRANSIENT TEMPERATURE DISTRIBUTION



$$t_f^* = \frac{t_f - t_o}{t_{fo} - t_o}$$

t_f = transfer fluid temperature
t_o = initial temperature
t_{fo} = temperature at inlet

$$x^* = x/\ell$$

$$\tau^* = \frac{\tau u}{\ell}$$

transfer fluid-water

storage medium-water

$$\frac{\dot{m} c_{tf}}{UA} = 1.1$$

$h_o = \infty$

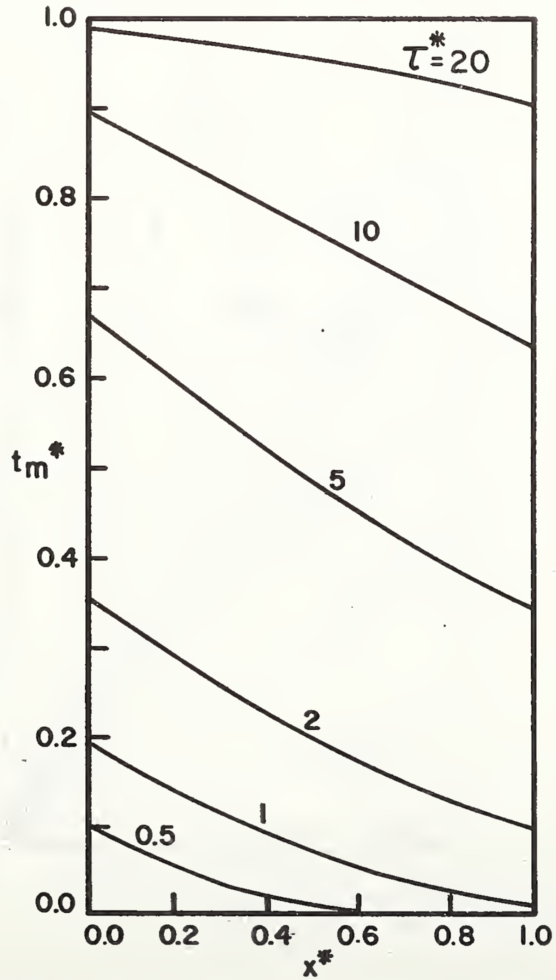
$d = 1.27\text{cm}$

$A_o/A_i = 4$

Figure B10 Transient Temperature Distribution for the Transfer Fluid in a Heat Exchanger-Type Thermal Storage Unit from Reference [14]

HEAT EXCHANGER TYPE THERMAL STORAGE UNIT

STORAGE MEDIUM TRANSIENT TEMPERATURE DISTRIBUTION



$t_m^* = \dfrac{t_m - t_o}{t_{fo} - t_o}$

t_m = storage medium temperature

t_o = initial temperature

t_{fo} = temperature at inlet

$x^* = x/\ell$

$\tau^* = \dfrac{\tau u}{\ell}$

transfer fluid - water

storage medium - water

$\dfrac{\dot{m} c_{tf}}{UA} = 1.1$

$h_o = \infty$

$d = 1.27\,\text{cm}$

$A_o/A_i = 4$

Figure B11 Transient Temperature Distribution for the Storage Medium in a Heat Exchanger-Type Thermal Storage Unit from Reference [14]

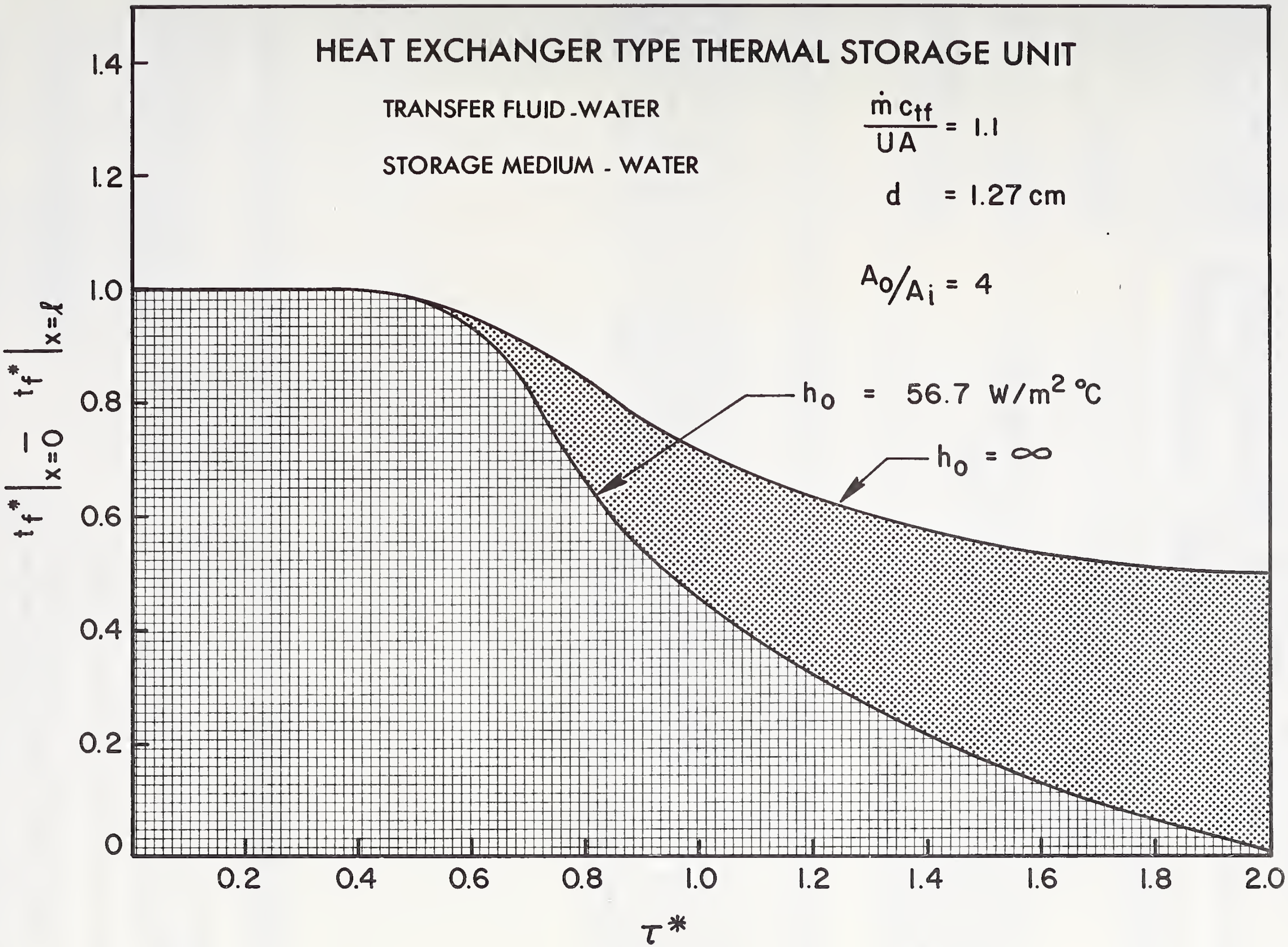


Figure B12 Comparison of the Transient Storage Capacity for Two Heat Exchanger-Type Thermal Storage Units from Reference [14]

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7. AUTHOR(S) George E. Kelly and James E. Hill			8. Performing Organ. Report No.
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12. Sponsoring Organization Name and Complete Address *(Street, City, State, ZIP)* Energy Research and Development Administration 1800 G Street Washington, D. C. 20550			13. Type of Report & Period Covered Interim Report 14. Sponsoring Agency Code

15. SUPPLEMENTARY NOTES

16. ABSTRACT *(A 200-word or less factual summary of most significant information. If document includes a significant bibliography or literature survey, mention it here.)*

A study has been made at the National Bureau of Standards of the different techniques that could be used for testing thermal storage devices and rating them on the basis of thermal performance. This document outlines a proposed standard test procedure based on that study. It is written in the format of a standard of the American Society of Heating, Refrigerating, and Air Conditioning Engineers and specifies the recommended apparatus, instrumentation, and test procedure.

17. KEY WORDS *(six to twelve entries; alphabetical order; capitalize only the first letter of the first key word unless a proper name; separated by semicolons)*

Standard test; thermal storage, thermal performance, solar energy, standard, thermal test

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