FACTORS AFFECTING THE DESIGN OF A SOLAR ENERGY STORAGE UNIT

Thesis for the Degree of Ph. D.

MICHIGAN STATE UNIVERSITY

John Jett McDow

1957

This is to certify that the

thesis entitled

FACTORS AFFECTING THE DESIGN OF A
SOLAR ENERGY STORAGE UNIT

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of the requirements for

Ph. D. **degree in** Agricultural Engineering

James S. Boyd

Date

FACTORS AFFECTING THE DESIGN OF A SOLAR ENERGY STORAGE UNIT

By

John Jett McDow

AN ABSTRACT

Submitted to the School for Advanced Graduate Studies of
Michigan State University of Agriculture and
Applied Science in partial fulfillment of
the requirements for the degree of

DOCTOR OF PHILOSOPHY

Department of Agricultural Engineering

Year 1957

Approved James S. Boyd

ABSTRACT

The objectives of this project were to (1) statistically analyze daily radiation data for East Lansing in the determination of frequency of various rate levels for each month, as would affect utilization and storage of solar energy, (2) mathematically design and construct a solar storage unit, and (3) perform operational tests on the solar energy storage unit under laboratory conditions.

Quantity of available solar radiation at any locality is the determining criterion in the design of solar utilization equipment. Daily solar-energy data for 14 years at East Lansing were analyzed. Charts developed were monthly probability curves and monthly coefficients of variation for normal, maximum, and minimum daily rates. The probability for a given job is selected by balancing between the importance of consistent energy rates and the allowable investment in equipment. Once this selection has been made, the probability curves give quantitative rates expected. The coefficients of variation aid in selecting a probability and adapting solar-energy equipment to other localities.

A study of heat storage methods proved that rocks would be the best material for agricultural use. Analysis by heat transfer principles indicated that the 4-inch diameter rock would provide the maximum rate of heat storage at minimum pressure drop across the system. Thermal conductivity, specific heat, and density of a special concrete mixture were determined. The 4-inch diameter spheres of

this mixture were used in a laboratory storage unit.

Copper-constantan thermocouples in 15 control spheres provided information on rate of heating, retention of heat, and rate of heat recovery from the spheres. Observations proved the spheres to react very closely to theoretical solutions. Lower mass air velocities of 320 lb per (hr)(sqft) provided the greatest heat transfer effectiveness and the most economical operation. This velocity provided a surface conductance coefficient of two for the spheres. The heating and cooling of the spheres could be considered essentially Newtonian.

Tests showed that the effectiveness of the spheres in heat absorption was reduced considerably after subjected to heated air for three hours. About 68 percent of the available heat was absorbed during this period, with the top layers heating rapidly at first and then the lower layers.

Up to 78.6 percent of the stored energy was calculated to be recoverable, when using a 12-foot cube storage unit within the building where the heat was utilized. It was also found that 68 percent of the energy stored could be remaining in the storage unit after three days. Faster heat losses at ends of storage unit during storage indicated that convection currents must be reduced to conserve heat.

In a prototype unit, the storage material would be of well sized and selected 4-inch diameter field rocks. Placement of the storage unit within the building where the heat is utilized will increase efficiencies. The shape should be cylindrical or cubical. Calculations showed that a 7-ft cube storage unit with stones could furnish a poultry house 25,700 Btu/hr for drying 16 hours a day. This was based on an 80 percent probability in January.

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I. INTRODUCTION

The energy from the sun has in the past been the ultimate source of all man's power. It is a continuing power which will be available as long as life exists on the earth. Man has utilized this energy in a multitude of ways, always selecting the means most easily harnessed with his meager devices. Direct radiation for keeping warm, animals and vegetables for nourishment and comfort, and fossil fuels have all been ways the solar energy provided man with the necessities for existence.

Solar energy is being constantly furnished to man through photosynthesis in plants, direct heating of surroundings, wind power, and water power. But the storage of surplus energy in fossil fuels or any other form known to man is positively not taking place at the rate energy is being utilized. The rapid depletion of these known sources of stored power has stimulated man to capture energies in other forms. Atomic energy will soon attain a respected position of furnishing useful power over wide areas; but the sources of raw materials for power generation by fission are limited and will eventually become exhausted. The fusion process, when fully developed, will not be limited by the lack of raw materials.

Throughout the mechanical age, man, being aware of the vastness of the daily solar energy received on the earth, has devised many ways of successfully capturing and utilizing it on a broad scale. His incentive has been dampened by the availability and the abundance of cheap fossil fuels in the form of coal, oil, and gas. The wide scale use of solar energy as a direct power source is a very difficult problem, but is not impossible. The main difficulties lie in its wide scattering, variability, and low temperatures compared with those under which present day machines operate. Daniels (19) indicates that these restrictions are not impossible in the following statement:

> If a tiny fraction of the effort which has been given to atomic energy were now to be invested in research on utilization of solar energy, significant progress would certainly be forthcoming.

This can well be true when considering that the average daily supply of solar energy in the more thickly populated areas of the world is about 500 kilocalories per square foot (19) or 1985 Btu per square foot.

Methods for accelerated use of solar energy are of a wide variety. Some of the important energy-conversion means are as follows: (1) Photosynthesis is used to store up energy in plants such as algae for a potential fuel. The tonnage of dry matter production is about ten times that of common crops. (2) Solar flat-plate collectors are the most common means of converting the radiant energy into useful sensible heat in the low temperature ranges. (3) Solar furnaces collect the radiant energy for higher temperatures up to 3000°C. (4) Photo-electric cells of the photovoltaic type are used to convert the energy into electrical power, a form of energy readily utilized.

Possible Use of Solar Energy in Agriculture

The possibility of incorporating solar energy as a power source is not limited to any one industry or area of work. However, owing to its availability over large geographical areas at comparatively low energy concentrations (as compared with fossil fuels), utilization of solar energy as a power source would conceivably be more readily adapted to the rural areas. A single farming unit presents a diversified number of power-requiring activities scattered over a wide area in contrast to the concentration of large power requirements in cities and industrial areas. Elimination of other uses of this dispersed form of energy is not to be implied, but accentuation is placed on the locations allowing immediate economical use of solar energy.

Farm operations and activities which could easily utilize the low-temperature solar heat are too numerous to duscuss fully here. However, the two outstanding ones which should be investigated first are mentioned. Crop drying and processing are probably the foremost functions which could well use solar energy. Although agricultural people have used the sun in drying crops in fields for centuries, modern practices of placing the crop in protective shelter as quickly as feasible for higher quantity and quality of production make this practice out-dated. Concentrated energy is, therefore, needed at the building site to finish the job of drying. This energy can be provided with a solar collector, as proved by Buelow (12).

Improper ventilation of animal shelters is an outstanding hindrance to building preservation and sanitary conditions for the occupants. Giese (24) states that "... ventilation should not only improve the purity of stable air and eliminate odors, but, perhaps primarily, remove moisture and prevent condensation which may have a detrimental effect upon the structural elements." Additional heat is necessary to help eliminate the moisture. Some of the other possible uses of solar energy on the farm are air-conditioning (heating and cooling) houses, heating and pumping water, preventing frost, and heating work areas.

II. OBJECTIVES

Satisfactory utilization of solar energy in agricultural work can be accomplished only after careful study of the problems involved. The boundaries of the problems studied in this work are set up in the following objectives:

1. Statistical analysis of daily solar-radiation data for East Lansing to determine frequency of various rate levels for each month of the year, as would affect utilization and storage of solar energy.

2. Mathematical design and construction of a solar-energy storage unit.

3. Operational tests of solar-energy storage unit under laboratory conditions.

III. REVIEW OF LITERATURE

A. Availability of Solar Energy

The basic consideration in solar energy utilization is that of its availability. Only by having adequate knowledge of its intensity and frequency of occurrence can a workable collector and storage unit for a locality be properly designed. Much effort has been exerted to devising maps indicating the average amount of solar energy received per square centimeter or per square foot in a horizontal plane. Baum (7) states that this is a valuable tool for gross planning, and its importance should not be minimized. But, still more important for local use, variations in the intensity and frequency due to atmospheric pollution, altitude, cloudiness, ground reflectivity, season, orientation, and latitude must be incorporated in the planning. Becker (8) made a study of these factors for several localities in the United States, and his results can be used to predict the local variation.

Hand (25) has developed a system of isolines denoting the average solar heat in Btu per square foot per average day for the United States. Its limitations have been pointed out; however, this should not overshadow its general benefits. These data have been further amplified by Crabb (18) in his solar radiation investigations in Michigan, which relate the average radiation for any one day of the year at East Lansing with that at other localities in the United States.

B. Storage of Solar Energy

1. General Historical Review

Storage of solar energy for periods of cloudiness or nighttime is a pressing problem. Robinson (35) states:

> The question of power storage by cheap and simple methods for the time of absence of solar radiation is one of the most important problems in the exploitation of solar energy. A good solution of this problem would enable the use of solar energy in places where this is impossible now.

Telkes (39) emphasizes its importance in connection with house heating:

> The storage of solar heat is one of the major problems to be solved; economically acceptable solutions must use a relatively small heat-storage volume within the house, because the cost of space is at a premium.

Concern for storage of solar heat has been evident in all projects leading up to utilization of this form of energy. Examination of patent claims and descriptions bears this out in early devised apparatus for handling solar energy. Calver (14) in 1883 made claims for his "Apparatus for Storing and Distributing Solar Heat," whose actual realization would be welcomed today.

> Claim 1. A solar-heat storage device comprising a reservoir completely surrounded, except at the heat-supplying orifice, when open, with non-conducting material, and a non-conducting door, substantially as specified.

The same principles of storing the heat were advocated in the late 19th century as are today. However, this is not to imply that improvements and progress have not been made over the models which were used then. Weston (41) promoted the idea of storage by a thermopile and storage cell in 1882. He was closely followed by Cottle (17) in 1897, who also advocated converting the heat to electricity:

> .... a thermo-electric generator adapted and arranged to convert heat from said body into energy of electricity ... .

But, Cottle proposed using a body of stones as a reservoir for the heat. Many of the early workers on solar energy did not specify the exact storage material but only stated "a body of heat-retaining material."

Examination of more recent patents indicates the emphasis placed on storage material having the heat of fusion taking place at relatively low temperatures. Howe and Katuck (27) patented the following heat-storage material in 1955:

> Claim 1. A storage material consisting essentially of tetrahydrate of calcium nitrate containing a nucleating agent selected from the group consisting of barium hydroxide octahydrate, cadmium hydroxide, sodium hydroxide, potassium hydroxide, and strontium hydroxide, the said nucleating agent being present in sufficient quantity to saturate said calcium nitrate tetrahydrate at a temperature above the melting point thereof.

Schaefer (36) defined the same year his patented heat storage material more specifically:

> A heat storage material consisting essentially of 5% to 15% by weight diphenyl ether and 30% to 50% by weight oleic acid, the balance consisting of stearic acid.

2. Characteristics of Specific Materials

Application of heat-storage materials in recently constructed solar heating systems has been limited to three kinds, rocks, water, and phase-change material. Each has shown distinct advantages and disadvantages, which govern the selection of the proper storage material for a specific application. In general, the final selection will be determined by the type of solar collector in the system, allowable space for storage, general design of system, and/or in summary the over-all cost of utilization of each material. A brief discussion is given for each material's use and, in addition, Table I summarizes the governing characteristics for these materials along with some others for possible use.

a. Water as storage material

Water has been a popular solar-energy storage material in several of the projects undertaken in recent years. It is to be the storage material for the fourth solar-heated house sponsored by the Massachusetts Institute of Technology. Whillier (42) expresses the outstanding advantages of water in the statement:

> Water was selected as a storage medium for several reasons, of which the most important are that the water is also to be used for removing the solar energy that is absorbed by the collector, and that the somewhat higher heat-storage capacity per unit volume of phase-change storage materials is not sufficient to justify their higher costs.

Data in Table I indicate that water is superior as a sensible-heat type storage material over others listed except a phase-change

TABLE I

PHYSICAL AND THERMAL CHARACTERISTICS OF HEAT STORAGE MATERIALS

Material	Ref.	Density, ρ lb/ft^3	Specific Heat, c Btu/lb °F	Thermal Conductivity, k Btu/ft^{2}hr °F/ft	Thermal Diffusivity, α ft^2/hr	Heat Storage Capacity with 26%[a] Void Btu/30°F ft^3	Relative Heat Storage Capacity, %
Water	(11)	61.84	0.997	0.368	0.00587	1371	100
Rock	(32)	170	0.2	1.5	0.0442	754	54.9
Salt Hydrates[b] ($Na_2SO_4 \cdot 10\,H_2O$)	(39)	92	1	—	—	9125	665
Concrete	(32)	144	0.25	0.54	0.015	799	58.2
Steel	(32)	492	0.106	26	0.498	1158	84.3
Mortar[c]	—	123.3	0.21	0.372	0.01435	575	41.9
Soybeans[d]	(2)	48	0.47	—	—	677	49.4
Wheat[d]	(2)	48	0.39	0.079	0.0042	562	41

Note: Characteristics are given for approximate temperature range of 70°F to 150°F.

a Based on close-packed spheres.

b Heat of fusion 104 Btu/lb.

c Specific mix used in current experiment.

d Bulk densities are given for grains without reduction of material for heat storage capacity.

material. However, the fact remains that a relatively expensive heat exchanger is necessary to obtain a suitable configuration pattern for heat transfer from water to air in the final phase of utilizing the heat. Recommendations for size of storage tank are based on collector size, with the optimum of about three gallons of water per square foot of collector (26).

b. <u>Phase-change material</u>

Materials having a large phase-change enthalpy within the normal storage-temperature range (90°F to 120°F) are preferable to sensible-heat type materials. On the surface Table I would seem to eliminate other materials, with the phase-change substances having a relative heat storage capacity of more than six times that of water and even more over the others. The cost of the raw material, in the range of $10 to $20 per ton, is not excessive. Telkes (39) reports several limitations of this material as experienced in tests made in the Dover house. First, the process of recovering the stored heat is not promptly reversible as the salts may not solidify upon cooling but will undercool below their normal melting points. This delay, however, can be overcome by the use of crystallization catalysts, or nucleating agents. A more important restriction in the use of Glaubers salts is the limited crystallization velocity of about "0.02 inch per hour per °F temperature difference between the solid and liquid (39)." Relatively thin, expensive containers would be necessary to obtain a suitable geometrical configuration for sufficient heat transfer. This obstacle casts a shadow on the immediate possibility of heat of fusion materials in agricultural work.

c. <u>Rocks as storage of sensible heat</u>

Rocks and the other solid materials listed in Table I do not offer highly desirable characteristics in heat storage. However, their low cost and other physical characteristics constitute the advantages for their use. The shape of small rocks and their normal placement within a container produces adequate surface-to-volume ratio for rapid heat transfer to the circulating fluid. Solar-energy storage units have been constructed with stones in sizes from three-fourths inch diameter [Löf (30)] to those having an approximate diameter of four inches [Bliss (9)].

Löf (29) advocates the use of gravel about 1½ inches in diameter after having made investigations with 3/4-, 1-, and 1½-inch size material. This type of storage unit is usually constructed in form of a column through which the air passes at a low velocity of about 1 to 2 feet per second. A heating front takes place at first at the entrance end and continues moving toward the exit end. A sufficient length of column enables the heat exchange to be very efficient since the exit-air temperature is nearly the same as the rock temperature until the heated front has reached that end. The air is usually circulated back through the collector for higher efficiency. The stratification of the heat is due mainly to a very low rate of heat transfer from pebble to pebble.

Bliss (9) in a 100-percent solar-heated house near Tucson selected four-inch diameter field stones as optimum. The channels for air flow are larger in this case, and accordingly the heat-transfer rate changes. This unit was 10 feet by 12 feet by 12 feet, and held 65 tons of stones. It provides winter heating and summer cooling of an average-size house. Bliss reports that under typical winter conditions this rock pile was adequate for heating during four cold cloudless days with its average temperature ranging from 90°F to 140°F. Under normal conditions, it was believed that there was a 25 percent heat loss which could be reduced by placing the storage unit directly under the house.

Utilization of the heat in both cases was obtained by reversing the direction of air flow. Both designers advocated dual use of the storage unit by blowing cool air through the bed at night, and circulating house air through the unit for day-time cooling. Auxiliary heaters were recommended for standby condition during wintertime operation.

d. Other storage methods

The door to other methods of solar-heat storage is certainly not closed. Robinson (35) states that every energy-change process for converting solar energy to potential energy is feasible. He suggests the possibility of electrolysis of water, if the storage problem of hydrogen and oxygen were simplified. Conversion to electrical energy would be a very desirable means, but the efficiency is low and the cost is high. Photosynthesis process will continue to be explored for storage of fuel in some areas.

IV. AVAILABILITY OF SOLAR ENERGY AT EAST LANSING

Effective utilization of solar energy in a given locality can be accomplished only when adequate basic data are readily available in usable form. The quantity of available solar energy holds the top position among data which should be known for design work. However, variability of the solar energy available in any one locality limits the use of average quantities for any given time. Solar radiation will vary by the hour, day, month, season, and even from year to year.

Of course, for any given date in the year, the probability of not receiving the maximum quantity of radiation will be because of cloudiness. This phase of the weather is not predictable for long periods of time, and, consequently, cannot be predicted from year to year for a given date. Average values could be used, but only to the extent that the average solar radiation rate for a specified date will be reached on a future day 50 percent of the time. Activities utilizing solar energy may be required to operate at an output level higher than one specified for 50 percent of the time. It may be necessary to operate on a basis of 75 percent, 85 percent, or some other probability.

Such needs indicate that a closer examination and analysis of the solar energy are necessary. For maximum use of the data, it is desired to know (1) rate of energy received for any probability, (2) coefficients for the adjustment of solar energy equipment designs in various localities due to difference in variation, (3) minimum and maximum rates of energy received for any probability, (4) variability of minimum and maximum which can be expected from year to year. A detailed statistical study of the solar energy data for a given locality is necessary in order to accomplish these purposes.

A. Source of Data

Solar radiation data, obtained from the Michigan Hydrologic Research Station under the United States Department of Agriculture and Michigan State University Agricultural Experiment Station, covered the period from December 12, 1942 to August 5, 1956. Failure of the pyrheliometer and/or the recording equipment owing to various reasons, including damage by hail, prevented the maintaining of a continuous record. However, absence of data during these few days did not prevent making a statistical analysis; the results were affected only by a slight reduction in degrees of freedom for some months.

There has been some question as to the validity of the data during some periods, owing to a change in pyrheliometers. Notes in the records of raw data indicate that the data taken during the period January 18, 1953 to November 5, 1954, were calculated with an incorrect "factor," and the recorded data should be multiplied by a factor of 1.24. However, a footnote in Climatological Data - National Summary (40) states with reference to the East Lansing Station:

> A study of available information about instrumental equipment and radiation indicates that data published prior to November 5, 1954, are systematically low. This condition has been corrected in data published in CDNS for that station beginning Nov. 5, 1954.

Assuming that the data were low only for the period January 18, 1953 to November 4, 1954, the proposed correction factor was applied to data taken in April during this period. A discrepancy in the results to follow was in most cases well below 3 percent. At any rate, the discrepancy is in the conservative direction of design when using the data.

All data published by the above station and other United States Weather Bureau Stations are given in units of gram calories per square centimeter (one gram calorie per square centimeter equals one Langley). These data were converted to British Thermal units by a multiplication factor of 3.69 to obtain units normally used in engineering work.

B. Analysis of Data

Solar energy at East Lansing and most localities is known to vary widely from month to month. It was, therefore, desirable to study the data for each individual month. Analysis of shorter periods of time, such as bi-monthly or weekly, may prove profitable in the future in order to increase the accuracy of prediction. Calendar months were used in this analysis for simplification of presentation and utilization of data. Periods with the beginning and terminal dates in step with the vernal and autumnal equinox would slightly increase the accuracy of prediction. Analysis of shorter periods would be advantageous with the latter system.

The mean, $\bar{x}$, and the standard deviation, s, from the 14 years of data were calculated for each month. By using the probabilities determined from the Z-Tables in Dixon and Massey (21), calculations were made to determine points for plotting the cumulative probability curves in Figure 1. These curves are classified as normal curves calculated about their means. Confidence in the validity of these normal curves based on 14 years of data was improved by analyzing data by another method. The raw data for the 14 years were tabulated in such a manner to enable construction of the cumulative probability curves for expected daily solar radiation in Figure 2.

Comparisons of the radiation-rate variability are possible by comparing the magnitude of the coefficients of variation, C, where $C = s/\bar{x}$. The coefficient unit is dimensionless and provides a means of comparing the variance of radiation among months or between two localities.

The minimum and maximum radiation rates for a given geographical location are some of the factors needed in the design of solar radiation equipment. The lower rate expected for a given month in some cases will be the determining criterion of design for an activity utilizing solar heat in which the incoming heat rate is of a critical nature. However, an auxiliary heating system may be required under such conditions. The upper extreme rate of incoming solar heat will require an adequate air flow unit to reduce temperatures below the point of danger of damage to component parts of the

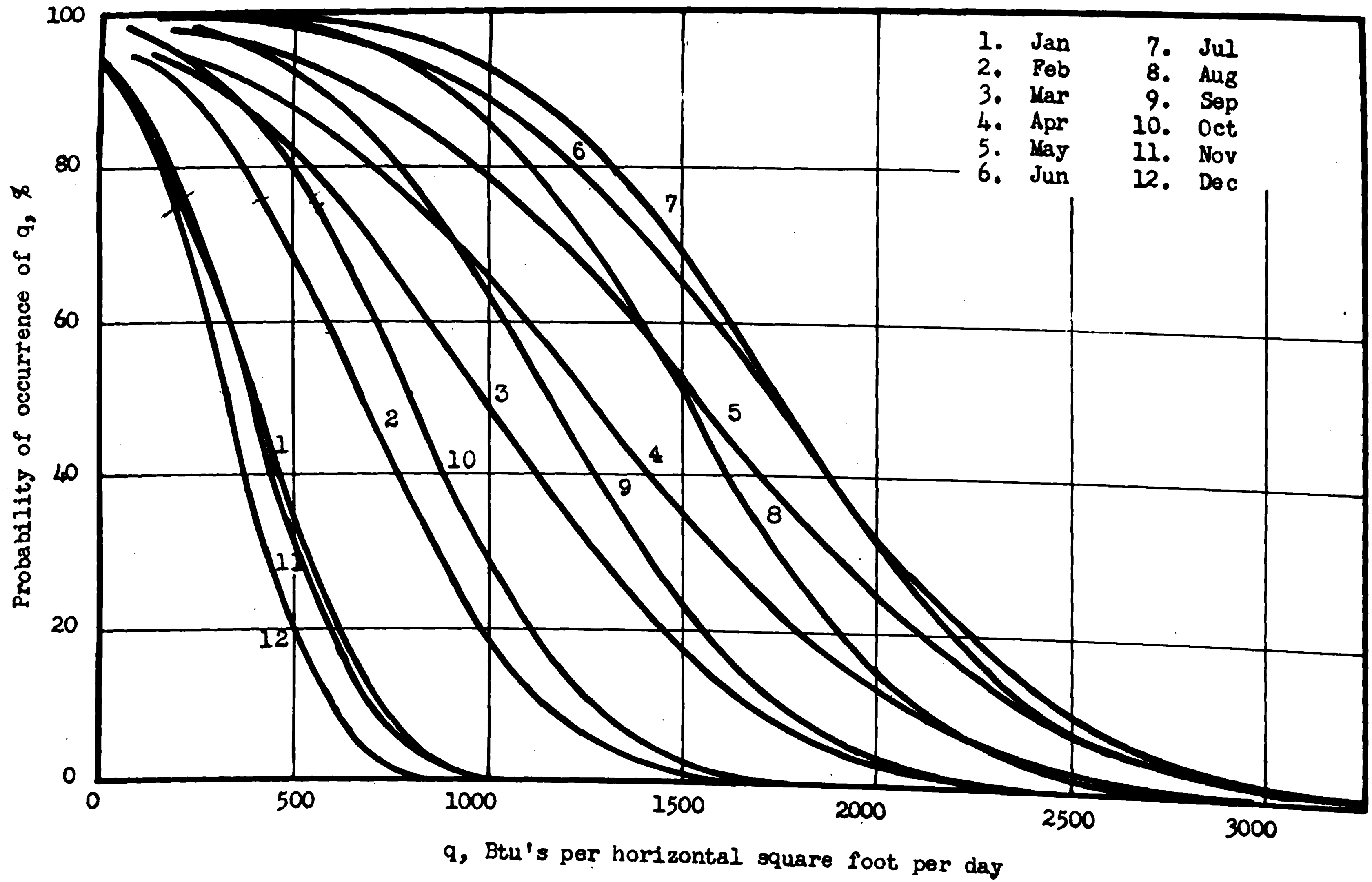


Fig. 1. Cumulative probability curves for daily solar radiation rates calculated about the mean for East Lansing, Michigan.

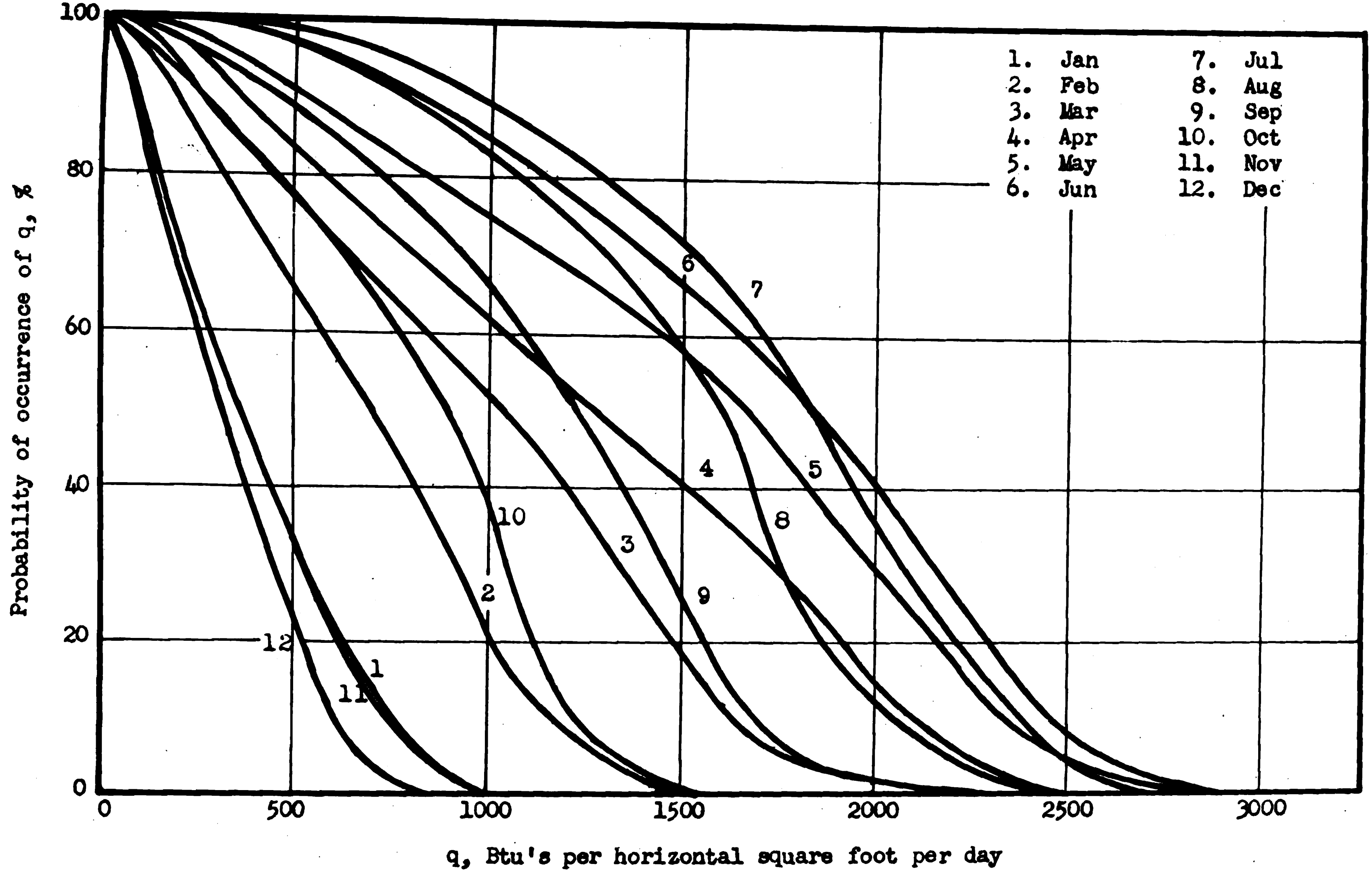


Fig. 2. Cumulative probability curves for daily solar radiation rates summarized from 14 years of data taken at East Lansing, Michigan.

system. For example, Buelow (13) experienced breakage of collector cover because of high temperatures reached. Also, some agricultural products being dried by solar heat would be damaged by air which is too hot.

The cumulative probability curves for monthly minimum and maximum solar radiation rates were developed by a method similar to the one used with the previous probability curves. Only normal curves were developed in these cases, as the variation within a given month was small with the exception of the minimums for the summer months. In like manner, the coefficients of variation for minimum and maximum daily rates were computed by the method previously described.

C. Results

The primary results of this phase of work are presented in the accompanying graphs. Figures 1 and 2 both represent cumulative probability curves for daily radiation rates. The predicted normal distributions about the means have been calculated for each month in Figure 1. Whereas, the curves in Figure 2 were plotted directly from tabulated daily rates for the 14 years of data. Close comparison of the two sets of curves will indicate the closeness of the paths for the two methods. Differences will be found mainly at the two extremes. The extreme values may be of importance in some engineering design work; therefore, more detailed treatment is given to the minimum and maximum rates study.

Utilization of these curves is explained in the following typical design problem. It is desired to use solar energy for the drying of corn in October. By predetermining the needs of the system, an estimation is made that the drying operation by utilizing solar energy could operate satisfactorily if a specified rate of heat would be available from a solar collector 75 percent of the days. From Figure 1, it is indicated that approximately 575 Btu per horizontal square foot could be obtained during the daylight hours. Knowing the total quantity of heat necessary for this particular job, the size, in square feet, of the solar collector can be calculated. Tilting the collector so that the rays of the sun are perpendicular to the collection plate will increase the amount of incoming heat. Such angles of tilt are discussed by Becker and Boyd (8).

Since the rate of solar energy varies each hour, efficient utilization of heat cannot be had by channeling all heated air through the grain continuously. Near solar noon, the rate of heat received could boost the air temperature high enough to damage the grain. During these periods of high intensity, part of the heated air could be diverted to a heat storage unit. The cumulative probability charts provide the total available energy rates for one square foot per day in a given month for these calculations.

The magnitude of the coefficients of variation for each month may be compared in Figure 3. The bar graphs show, as might be expected, that the solar radiation rates are much more variable about their means in winter months than in summer months. Hence, a lower probability must be selected in design work for winter use as compared with summer use.

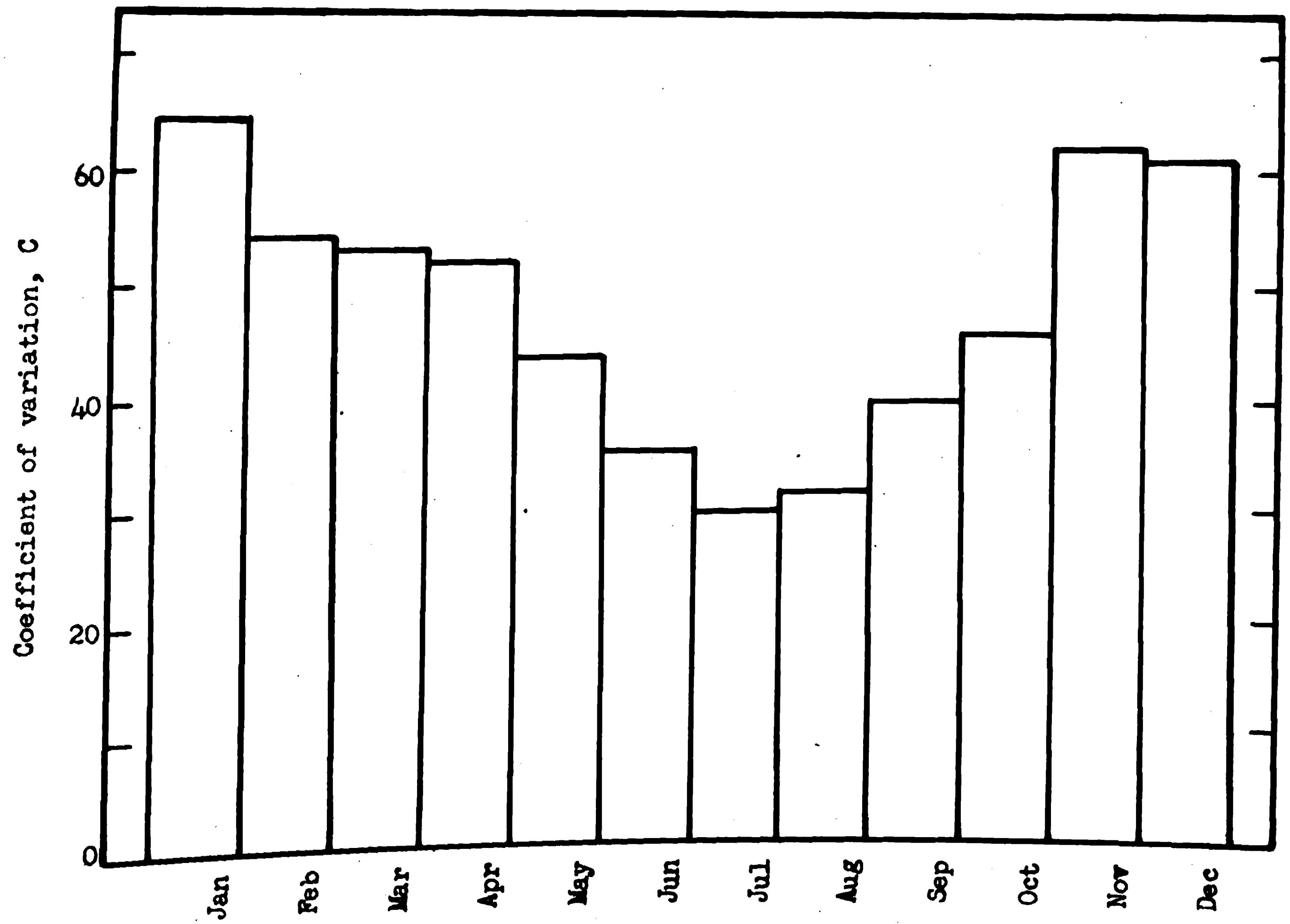


Fig. 3. Coefficients of variation for daily solar energy rates at East Lansing, Michigan.

Cumulative probability curves for minimum rates of solar energy (Figure 4) have a large variation from month to month, with December having a minimum of less than 51 Btu per horizontal square foot for 88.5 percent of the years. The steep slopes of the curves in the colder months indicate that these low rates will be reached nearly every year. However, a month such as July will have a wide variation in minimum rates from year to year. The negative slopes of the June and July curves are much less than those of other months.

The cumulative probability curves about the means for monthly maximum rates, shown in Figure 5, do not have the variation that is found among the minimum curves. The slopes for all months are very nearly the same, although the magnitude of the maximum value for a given probability varies from month to month. The steepness of the curves indicates that maximum radiation expected for any one month does not have a wide variation. For example, June's maximum will always stay within the range of 2290 to 2790 Btu per horizontal square foot 80 percent of the years. Also, the December maximum will vary only from 580 to 850 Btu per horizontal square foot for 80 percent of the years.

A study of the coefficients of variation for the minimum and maximum values in Figure 6 will indicate several obvious factors. Minimum expected rates for any one month vary widely from year to year. This is due to the wide variation of cloudiness that may occur for any given month in one year. On the other hand, the maximum rates have a very small coefficient of variation. The

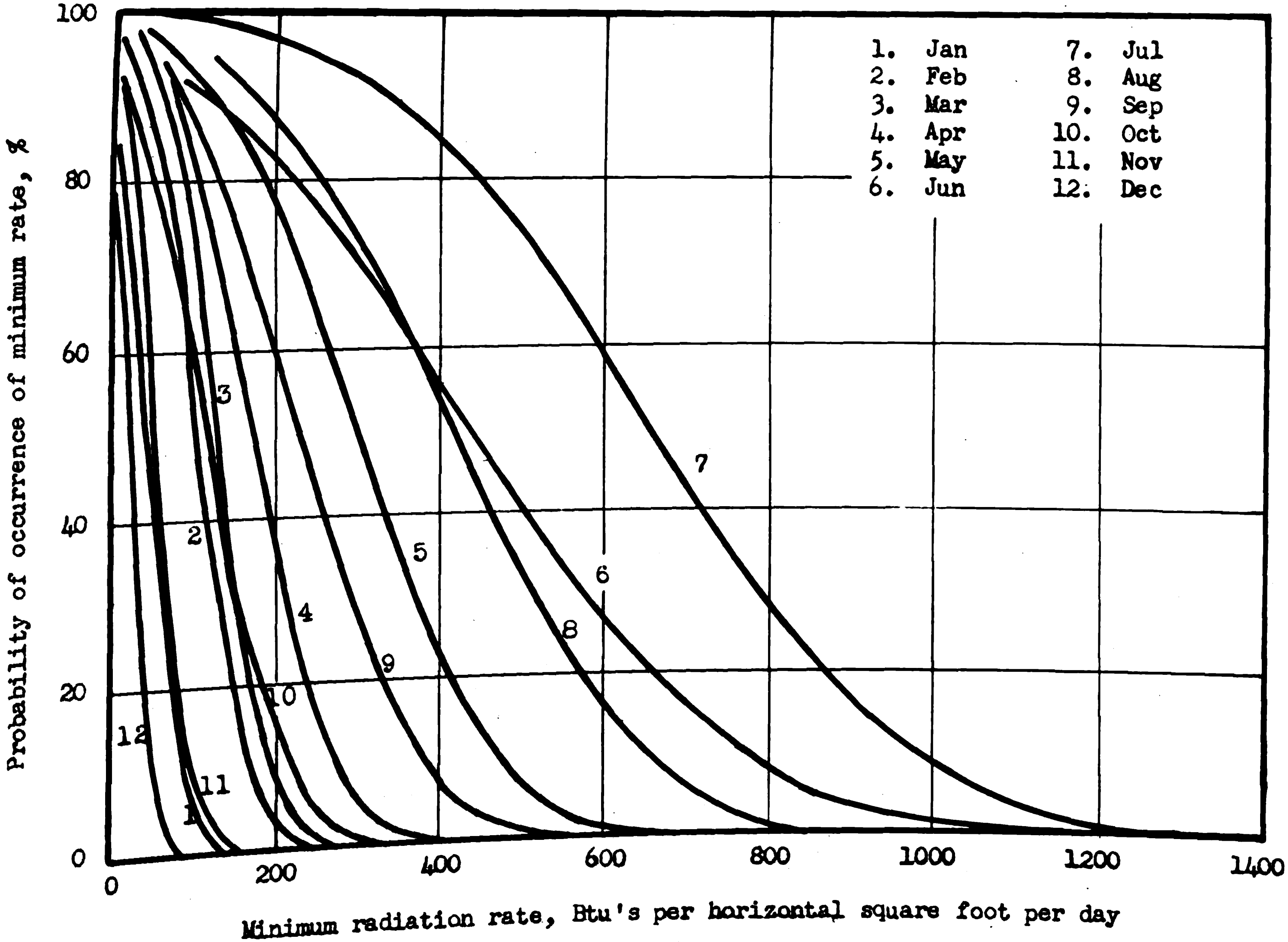


Fig. 4. Cumulative probability curves for minimum daily solar radiation rates calculated about the means for East Lansing, Michigan.

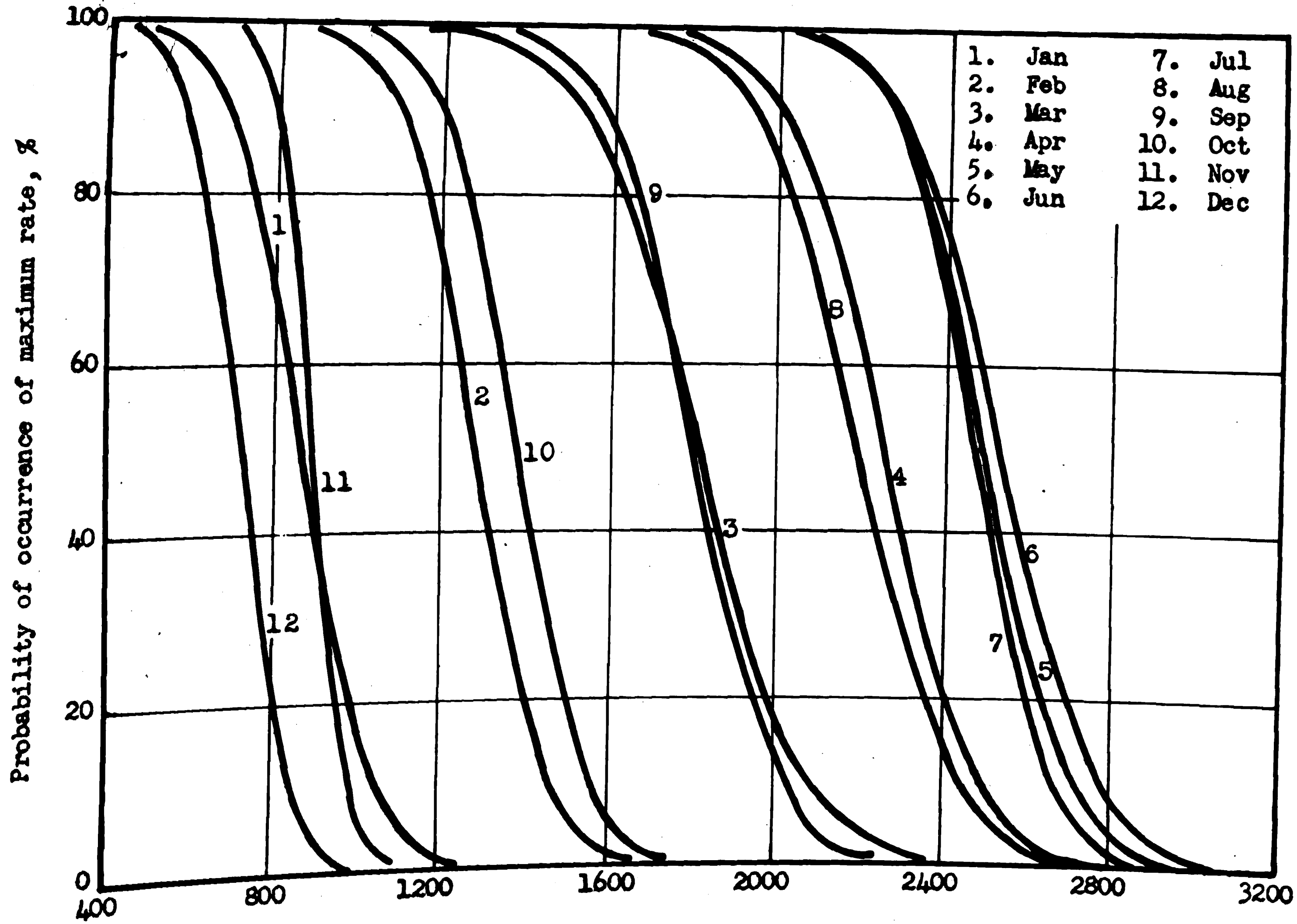


Fig. 5. Cumulative probability curves for maximum daily solar radiation rates calculated about the mean for East Lansing, Michigan.

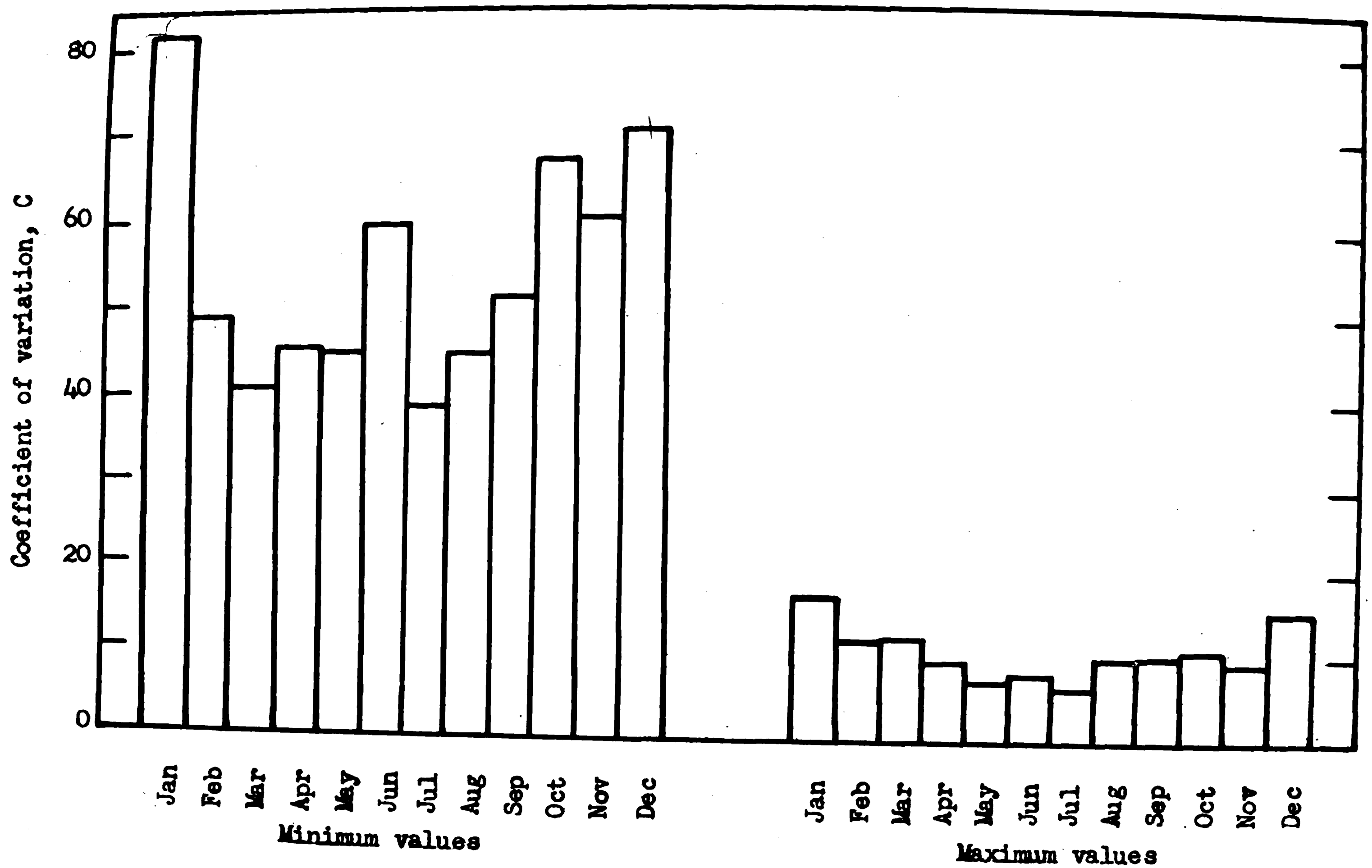


Fig. 6. Coefficients of variation for minimum and maximum daily solar energy rates for East Lansing, Michigan.

maximum daily rate of solar energy in a given month will probably be reached each year, but the minimum rates for a given month will vary widely from year to year.

The limitation of the probability curves lies in the ability of the user to select a proper value of probability. If the need of heat is of a critical nature, a higher probability would be necessary. This will result in dependence on lower radiation rates, but will have assurance of obtaining that particular rate or a greater one a larger percentage of the time. The dependency upon lower rates directly requires larger and more expensive solar equipment to handle the job. The selection of a suitable probability will, therefore, depend upon the judgment of the designer.

The coefficient of variations can aid the user in the selection of the probability. Higher coefficients indicate that the variability of the daily rates will be greater. For closer design work, it would then be desirable to select a higher probability. If a coefficient of variation for a given month in one locality is much lower than in another area, the solar equipment in the former location could be smaller and do the same job when comparing to the latter place. The coefficients of variation can, therefore, help project the experience gained in one place to other areas effectively. This will become more important as more experience is had.

V. DESIGN OF A REGENERATIVE SOLAR STORAGE UNIT

A. Selection of a Storage Medium

Wise selection of a storage means in a solar-energy utilization system requires careful study of the existing methods and some consideration of new ones. The existing ways include (1) biochemical conversion to vegetation by photosynthesis, (2) electrical conversion by thermopiles or semi-conductors and storage in batteries, and (3) storage directly in form of sensible heat or phase-change. For economic reasons, the third method appears more appropriate for application in and around farm structures. The wide choice of materials listed in Table I for this purpose is not the limit of selection. However, it does include those apparently known to be suitable at the present time. Detailed discussion earlier has covered the advantages and disadvantages. For the present study, rock was selected for the storage material in this study for the following reasons:

1. Configuration of material provides a self-contained heat exchanger.
2. Heat loss reduced due to conduction because of point-to-point contact between stones.
3. Large surface area between fluid and solid allows large heat transfer at low temperature differences.
4. Allows low initial cost.
5. Minimizes depreciation, costs of maintenance, and repair of storage material.
6. Installation without skilled labor.

7. No maintenance when not in use.

8. Small energy consumption for forced air movement through heat exchanger.

B. Mathematical Study of Heat Transfer in Spherical Bodies

Although neither field stone nor gravel is found consistently in regular geometrical shapes, they normally approach more nearly the shape of a sphere. This is in comparison to other common geometries, such as the cylinder, plate, or cube, which have been examined for heat transfer purposes. By considering the stones to be spherical, calculation of heat storage ability for various values of surface conductances, of periods of time, and of ranges of size is possible for prediction of optimum requirements.

A spherical rock subjected to heating or cooling can be examined for a variety of conditions. First, consider that it has a high thermal conductivity, k, which would reduce the temperature gradient within the sphere during any heating or cooling process. The heat transfer process would be controlled mainly by the surface resistance and would be called Newtonian heating (or cooling). The temperature history of the sphere could then be expressed as

$$\frac{t - t_f}{t_i - t_f} = e^{-(Ar_o/V)(Nu\,\alpha\,\theta/r_o^2)} \tag{1}$$

or, in reduced form

$$\frac{t - t_f}{t_i - t_f} = e^{-(3Nu\alpha\theta/r_o^2)} \tag{2}$$

as given by Schneider (37). Definitions of symbols used in the above equations and throughout the entire manuscript are given in appendix A. The cumulative heat rate, Q, after time, θ, may be expressed as

$$Q = cwV\left[t_f - t_i\right]\left[1 - e^{-(3Nu\alpha\theta/r_o^2)}\right] \tag{3}$$

The most likely situation that will be encountered in a transient heating and cooling system would be one with finite internal and surface resistances. Schneider (37) and Boelter et al. (10) derive the temperature history of a sphere for this condition,

$$\frac{t - t_f}{t_i - t_f} = 2\sum_{n=1}^{n=\infty} e^{-Mn^2\left[\alpha\theta/r_o^2\right]} \frac{\left[SinMn - MnCosMn\right]\left[Sin(Mnr/r_o)\right]}{\left[Mn - SinMnCosMn\right]\left[Mnr/r_o\right]} \tag{4}$$

where Mn are the roots of the transcendental equation

$$Nu = 1 - MnCotMn. \tag{5}$$

Schneider presents values of the first five roots which apparently are an adequate number for solutions in normal engineering problems.

Boelter et al. go further and derive the cumulative heat, Q, equation for a sphere with finite internal and surface resistance.

$$Q = 8\pi r_o^3 \rho c(t_i - t_f) \sum_{n=1}^{n=\infty} \frac{\left[\text{Sin}Mn - Mn\text{Cos}Mn\right]^2 \left[1 - e^{-(Mn^2 \alpha \theta / r_o^2)}\right]}{Mn^3 \left[Mn - \text{Sin}Mn\text{Cos}Mn\right]} \tag{6}$$

The determination of an optimum size rock for a regenerative storage system was made by Equation 6, using the physical characteristics given in Table I. Several reasonable values for the surface resistance, h, and time of fluid flow, θ, were selected and the results plotted in Figure 7. Note should be made that a temperature difference, $t_i - t_f$, is not specified but would vary and depend on the given condition. Values of the cumulative heat, Q, will naturally vary for various sizes of rocks. To arrive at an optimum rock size, data from Figure 7 were used to calculate the heat stored in a cubic foot of material, considering that the spheres are close packed. The summary of the results is presented in Table II.

Values of the upper limit of heat stored in a cubic foot of rock vary slightly around 25 Btu. This small variation is owing to the vast amount of calculations involved and the limitations of slide rule accuracy. The significant point is that the size of rock at which the heat stored drops off, for a film coefficient of six, is 0.5-, 4-, and 8-inch spheres for periods of 0.1, 1.0, and 10 hours, respectively. Collection periods of less than one hour may not prove feasible for storage. Therefore, the four-inch diameter sphere would have approximately the maximum cumulative heat storage with the least amount of pressure drop of air flow through the system. If higher film coefficients and in turn high velocities of air flow are used, the optimum size rock for maximum heat

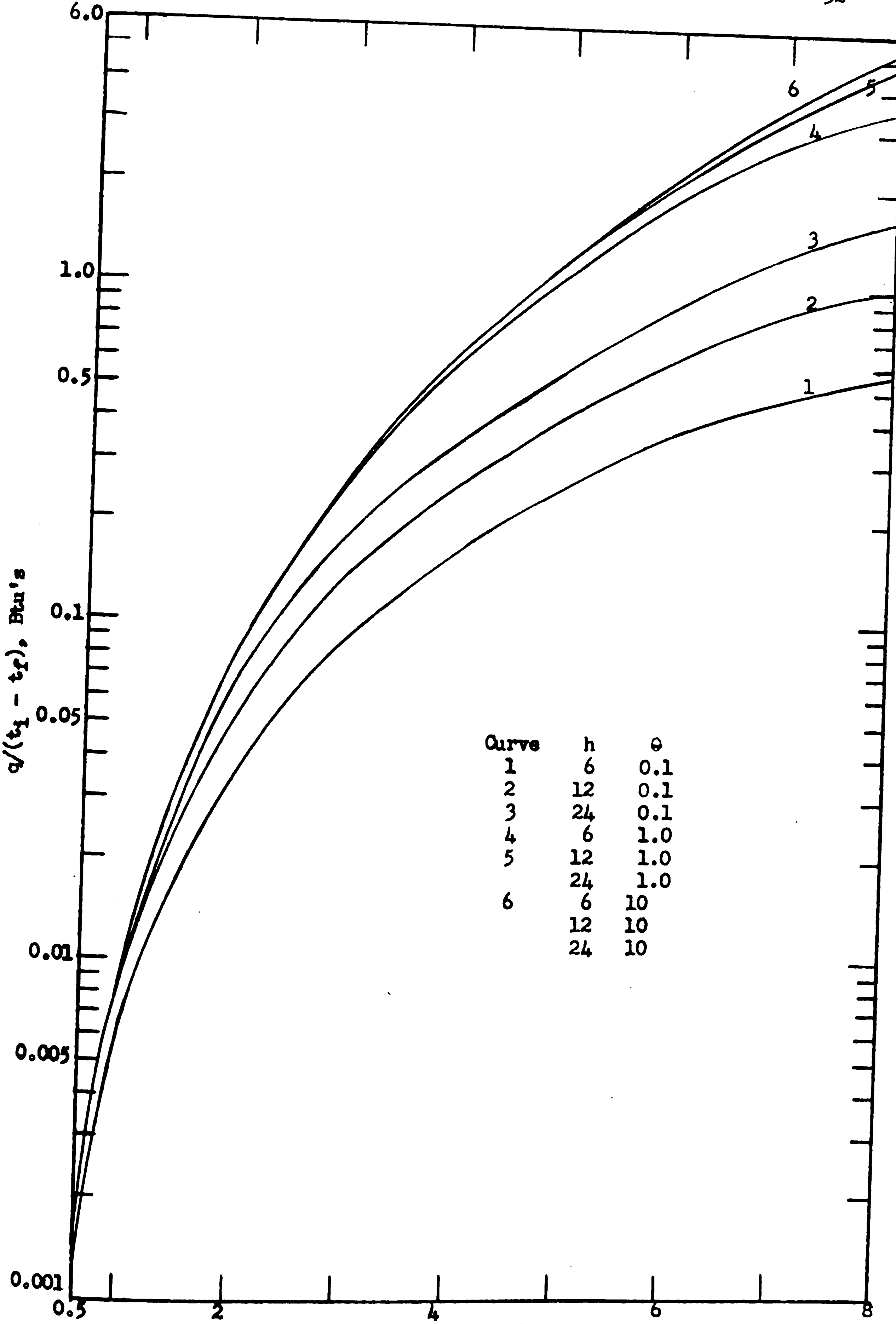


Fig. 7. Theoretical cumulative heat storage in single rocks.

TABLE II

CUMULATIVE HEAT STORAGE IN ONE CUBIC FOOT OF FIELD ROCKS FOR THE SPECIFIED HEATING PERIODS AND SURFACE RESISTANCES

Diameter of Spherical Rock		Rocks per Cubic Foot	Surface Resistance	Time, θ, Hours					
				0.1		1.0		10.0	
				Cumulative Heat Stored $Q/(t_i - t_f)$, Btu					
Inches	Feet			Per Rock	Per Cuft	Per Rock	Per Cuft	Per Rock	Per Cuft
0.5	0.0417	19,584	6	0.001155	22.6	0.001258	25.	0.001258	25.
1.	0.0833	2,448		0.00711	17.4	0.01015	25.	0.01015	25.
2.	0.1667	306		0.0372	11.4	0.0826	25.	0.0826	25.
3.	0.25	90.4		0.0903	8.16	0.273	24.7	0.279	25.
4.	0.3333	38.2		0.163	6.22	0.611	23.3	0.654	25.
6.	0.5	11.3		0.38	4.30	1.84	20.8	2.225	25.
8.	0.667	4.8		0.575	2.76	3.53	16.9	5.240	25.
0.5			12	0.00126	25.	0.00126	25.	0.00126	25.
1.				0.00928	22.7	0.0103	25.	0.0103	25.
2.				0.0519	15.9	0.0827	25.	0.0827	25.
3.				0.1411	12.75	0.277	25.	0.277	25.
4.				0.237	9.05	0.650	24.8	0.659	25.
6.				0.625	7.06	2.19	24.8	2.225	25.
8.				1.038	4.98	5.13	24.6	5.140	25.

(Continued)

TABLE II (Continued)

Diameter of Spherical Rock		Rocks per Cubic Foot	Surface Resistance	Time, Θ, Hours					
				0.1		1.0		10.0	
				Cumulative Heat Stored $Q/(t_i - t_f)$, Btu					
Inches	Feet			Per Rock	Per Cuft	Per Rock	Per Cuft	Per Rock	Per Cuft
0.5	0.0417	19,584	24	0.001292	25.	0.001292	25.	0.001292	25.
1.	0.0833	2,448		0.01018	24.9	0.01018	25.	0.01018	25.
2.	0.1667	306		0.0686	21.	0.0826	25.	0.0826	25.
3.	0.25	90.4		0.206	18.6	0.277	25.	0.278	25.
4.	0.3333	38.2		0.332	12.7	0.654	25.	0.654	25.
6.	0.5	11.3		0.906	10.25	2.200	25.	2.225	25.
8.	0.667	4.8		1.657	7.95	4.93	23.7	5.23	25.

transfer will, of course, increase. It was concluded, therefore, to use a four-inch diameter rock for the heat-storage bed.

C. Determination of Thermal and Physical Characteristics for Storage Material

A sufficient number of four-inch diameter field stones approaching spheres would be difficult to accumulate for the test. Also, the exact thermal conductivity of these stones could not be measured with available equipment. These variabilities in the storage system were eliminated by molding the spheres from a very dry mixture of a water: cement: sand ratio of 1:1.92:6.68 by weight. This enabled an exact determination of the thermal conductivity to be made for this material. The guarded hot plate modified and calibrated by Anderson (5) and operated under the specification of the American Society for Testing Materials (3) was used to determine the thermal conductivity of two 1-inch by 12-inch by 12-inch mortar plates made of the same mix as that of the spheres. The value obtained was 0.372 Btu per (hr)(sq. ft)(^{o}F) per (ft). A picture of this testing equipment in operation is shown in Figure 8.

Determination of the specific heat was made by use of a calorimeter, with the resulting value of 0.210 (Btu) per (lb)(^{o}F). An average density of 123.3 pounds per cubic foot was determined by taking the weight and dimensions of several geometrical shaped figures made of this mixture.

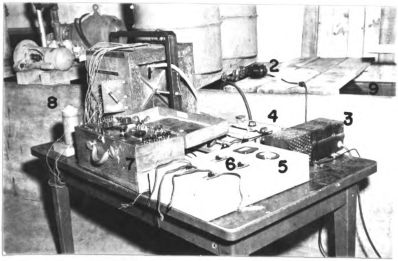


Fig. 8. A view of the equipment used in determining the thermal conductivity of the heat storage material by the guarded hot plate method.

(1) Guarded hot plate with specimen
(2) Water circulation pump
(3) Constant voltage transformer
(4) Rheostats
(5) Voltmeter and ammeter
(6) Switches
(7) Potentiometer
(8) Ice bath (ref. thermocouple junction)
(9) Water source

With the necessary physical characteristics determined, the cumulative heat in one sphere for one degree temperature difference was calculated by Equation 6 for time periods of the range 0.1 to 2 hours and film coefficients of 6, 12, 18, and 24. The resulting data were plotted in Figure 9. Study of the curves in Figure 9 discloses that the rate of heating is not increased in the same proportion as the increase in the surface conductance value. Higher velocities of air, with accompanying increased operational cost, are not necessarily justified for obtaining high h-values. For surface conductance values of 6 or more, 93 percent of the maximum storable heat is already in the sphere after one hour's operation. Again, note should be made that the difference between the initial temperature of the sphere and that of the air or fluid does not affect the rate of heating or cooling, but only the quantity of heat stored. The apparent maximum heat which can be stored per °F temperature difference is 0.5 Btu, which is approached asymptotically for all values of the surface conductance coefficient.

D. Design of the Control Sphere

For actual study of the heat received by the sphere, four thermocouples were placed in 16 control spheres. Placement of the thermocouples was made at the boundary layers of three concentric, equal-volume shells about the center of the sphere during the molding process. Exact positions are denoted in Figure 10 and a photographic cut-away view of the spheres in Figure 11. The

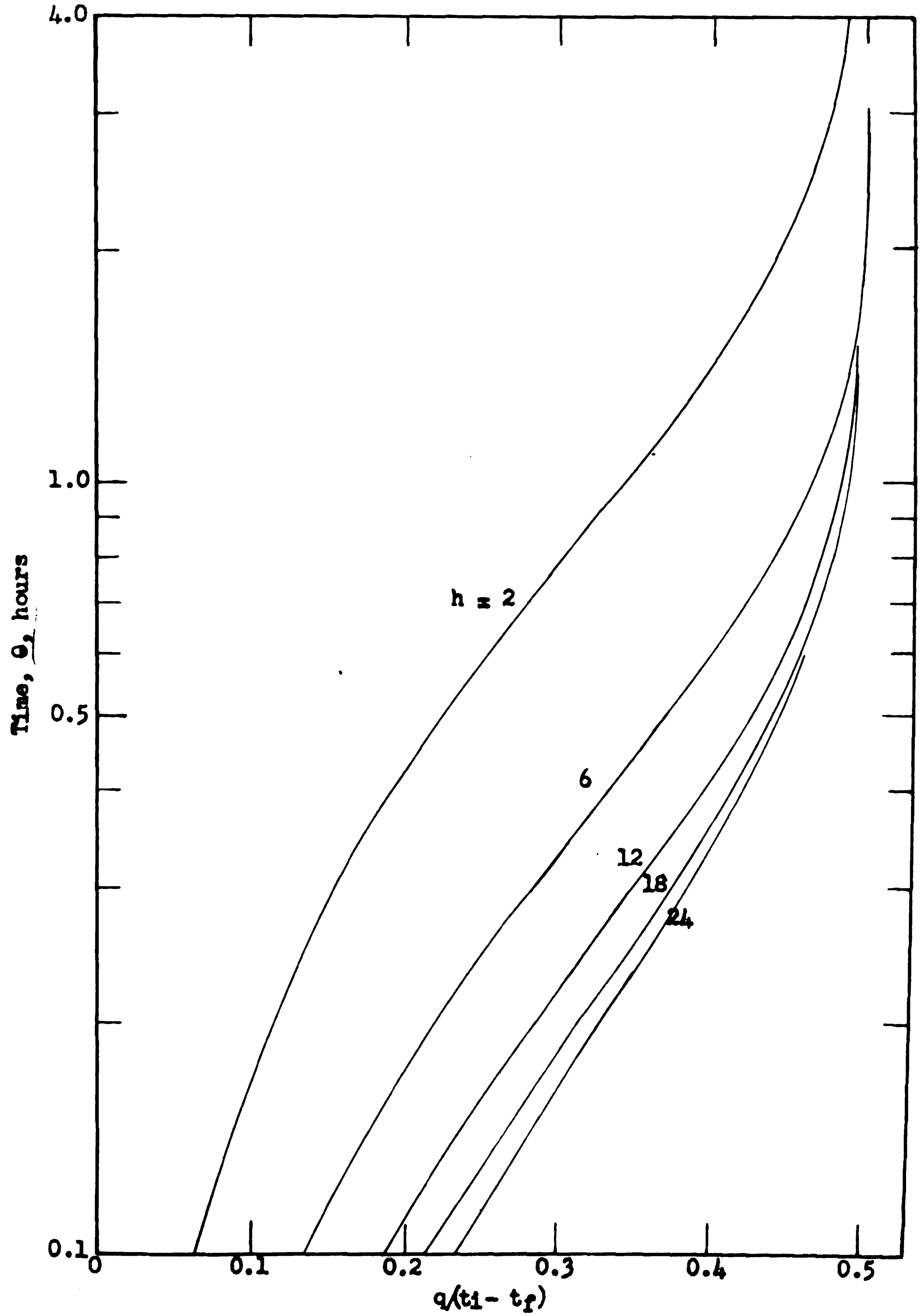
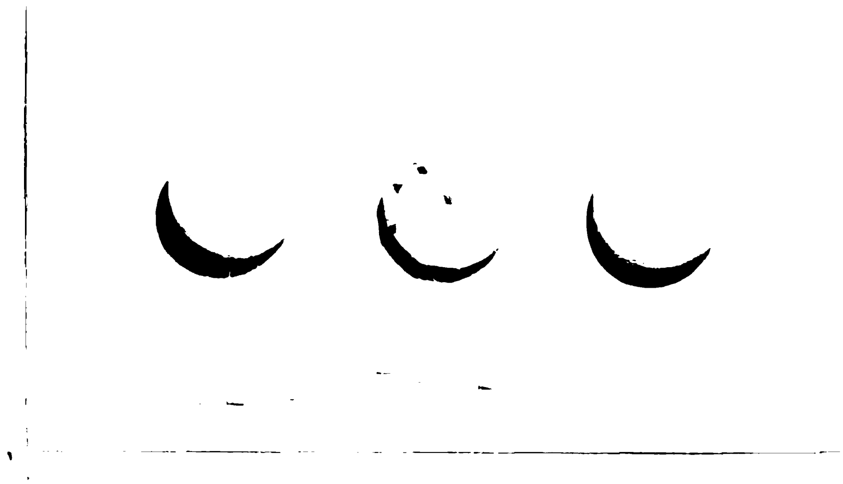


Fig. 9. Theoretical accumulated heat in a single sphere.



a b c

Fig. 10. A photograph of concrete spheres used in tests.
(a) Control sphere with thermocouples
(b) Cut-away view of control sphere with thermocouple placement
(c) Plain sphere

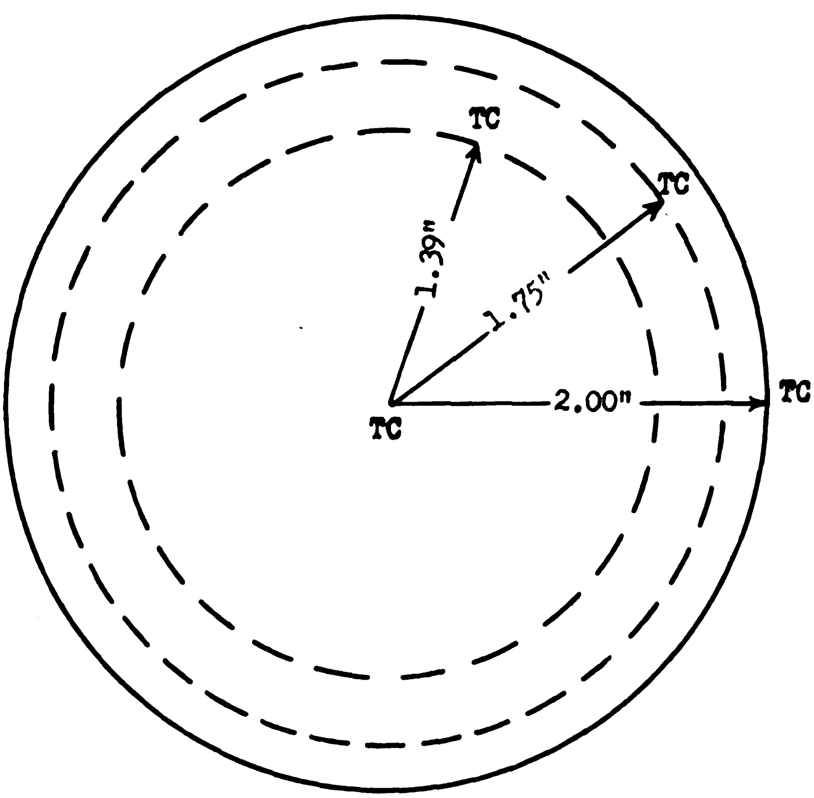


Fig. 11. Positions of thermocouples in 15 control spheres.

The copper-constantan thermocouples were made of 30-gauge wire to give a small junction point and reduce heat loss or gain through the wire.

Theoretical calculations with Equation 4 were made to study the temperature history of the thermocouple points within the sphere and plotted in Figure 12. For these calculations, an h-value of 1.65 was used for comparison with oven conditions. However, an h-value of 1.76 is produced by the following empirical equation from Brown and Marco (11) for natural convection about a sphere:

$$h_c = 0.63 \frac{k}{r} (ar^3 \Delta t)^{\frac{1}{4}} \tag{7}$$

Definitions of symbols are given in Appendix A.

A similar study was made by placing the control spheres in an oven of an average temperature of 208°F. Two recording potentiometers were used to make the temperature history. Average values of the corresponding points for the 16 spheres at specified times were used to plot the actual temperature histories in Figure 13.

Two distinct differences are to be noted between the actual and the theoretical temperature histories. First, the temperature gradient appears to be considerably smaller under actual tests. Secondly, the equilibrium temperatures are reached more quickly under the actual situation than the equation predicts. Several factors could be responsible for these discrepancies. First, it is apparent that the film conductance chosen for natural convection was slightly low in comparison with the one given by the formula. Other variables entering the formula have been measured with reasonable accuracy;

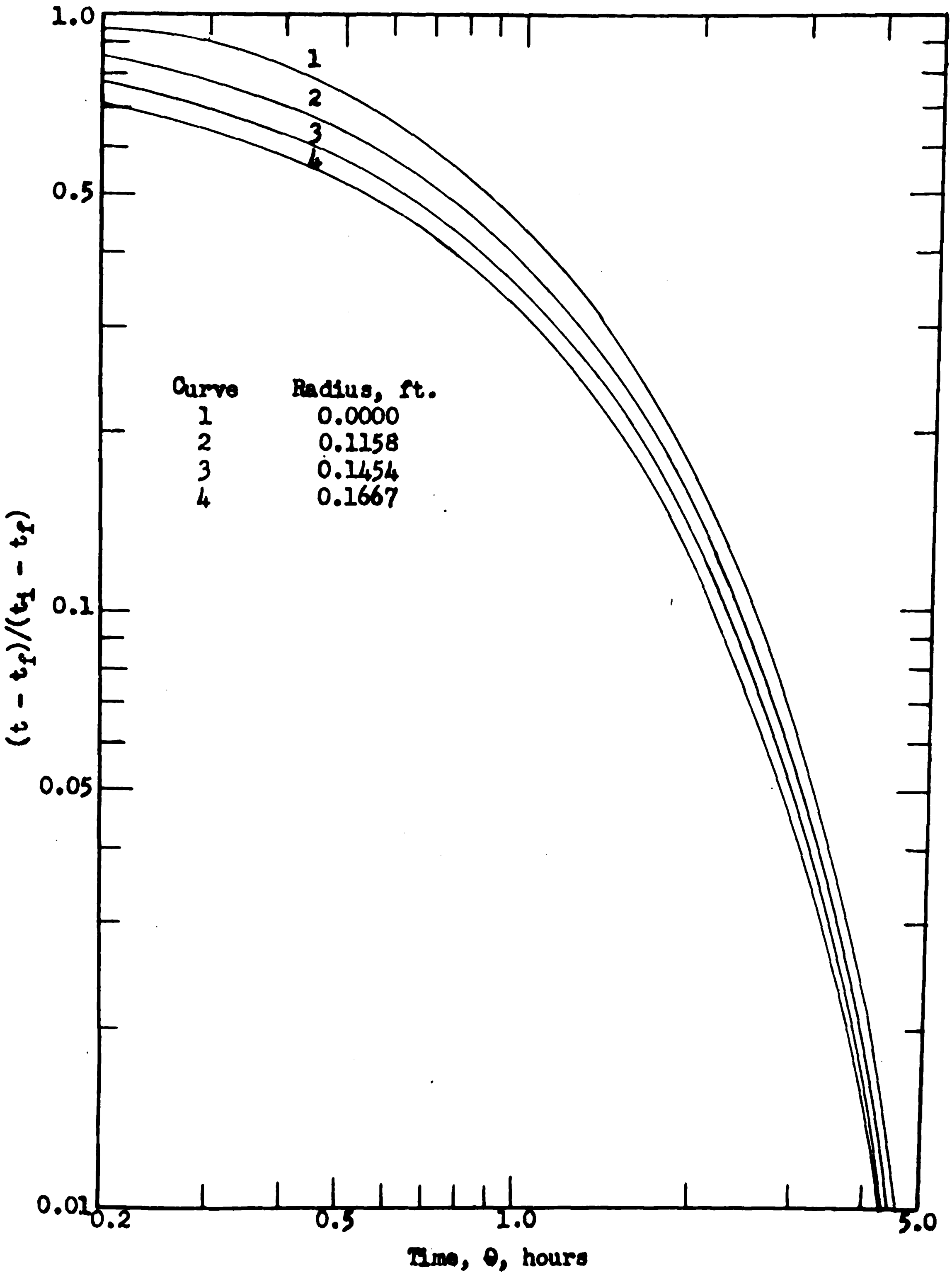


Fig. 12. Theoretical temperature history of points at the thermocouple locations within 4-inch diameter concrete spheres.

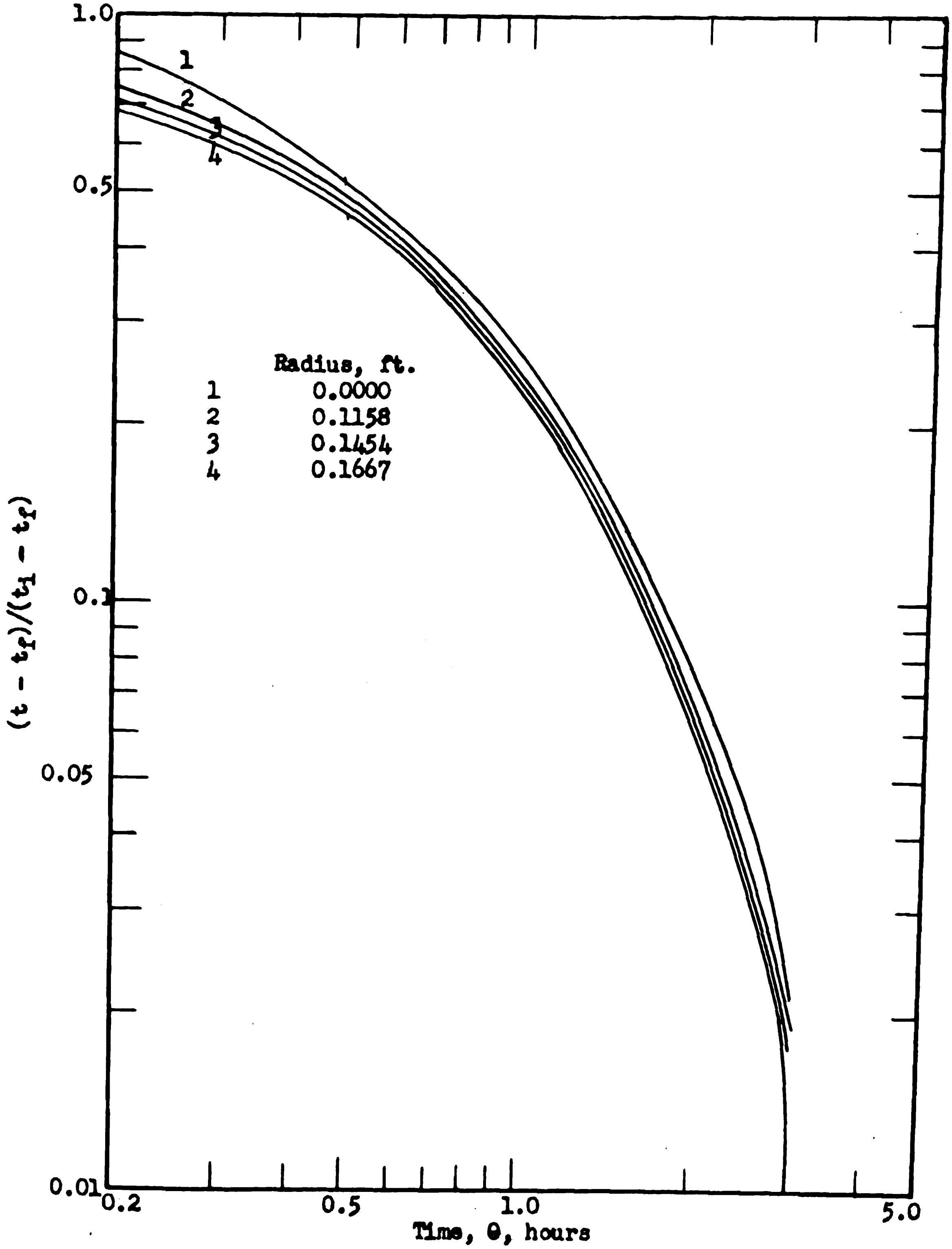


Fig. 13. Observed temperature history of point at the thermocouple locations within a 4-inch diameter concrete sphere.

however, variations owing to time lag in thermostat control of heater, conduction from plates supporting the spheres, and radiation probably are the contributors to these discrepancies. Percentage of error varies from approximately zero within the first 12 minutes to 53.9 percent of the theoretical at the time two hours. The high percentage discrepancy at the later time is due to the sharp drop in the curves.

E. Design of Heat Storage Unit

A storage unit for the initial investigations was designed to be small and compact so that closer control could be had over the variables. A cross-sectional view of the cylindrical-shaped container and its supplementary components is shown in Figure 14. The cylindrical container provides the minimum exposed surface to volume ratio of common geometrical shaped bodies with the exception of a sphere. This allows a minimum heat loss for a defined amount of insulation. The one-inch rock wool insulation covers both the outside and inside of the main body. Only one layer of insulation was used on the approach and exhaust frustums and the pipe leading to the fan. The three-layer sheet asbestos covering over the pipe containing the heating element was primarily a safety feature to eliminate exposure of a hot pipe to nearby surroundings in the building.

An electrical heating source was selected in order that constant control over the incoming heat was obtained. Variation of solar energy through a collector during tests would make accurate measurement of heat input to the storage unit more difficult. Input heat was regulated by the carbon pile rheostat in series with the heating element. A vane anemometer was used to measure the air velocities in the six-inch pipe on the exhaust. This anemometer was found to have a considerable error, owing to the fact that it was calibrated in open air rather than on a pipe. A $2\frac{1}{2}$-inch orifice plate was later constructed to fit in a nominal three-inch tube, according to

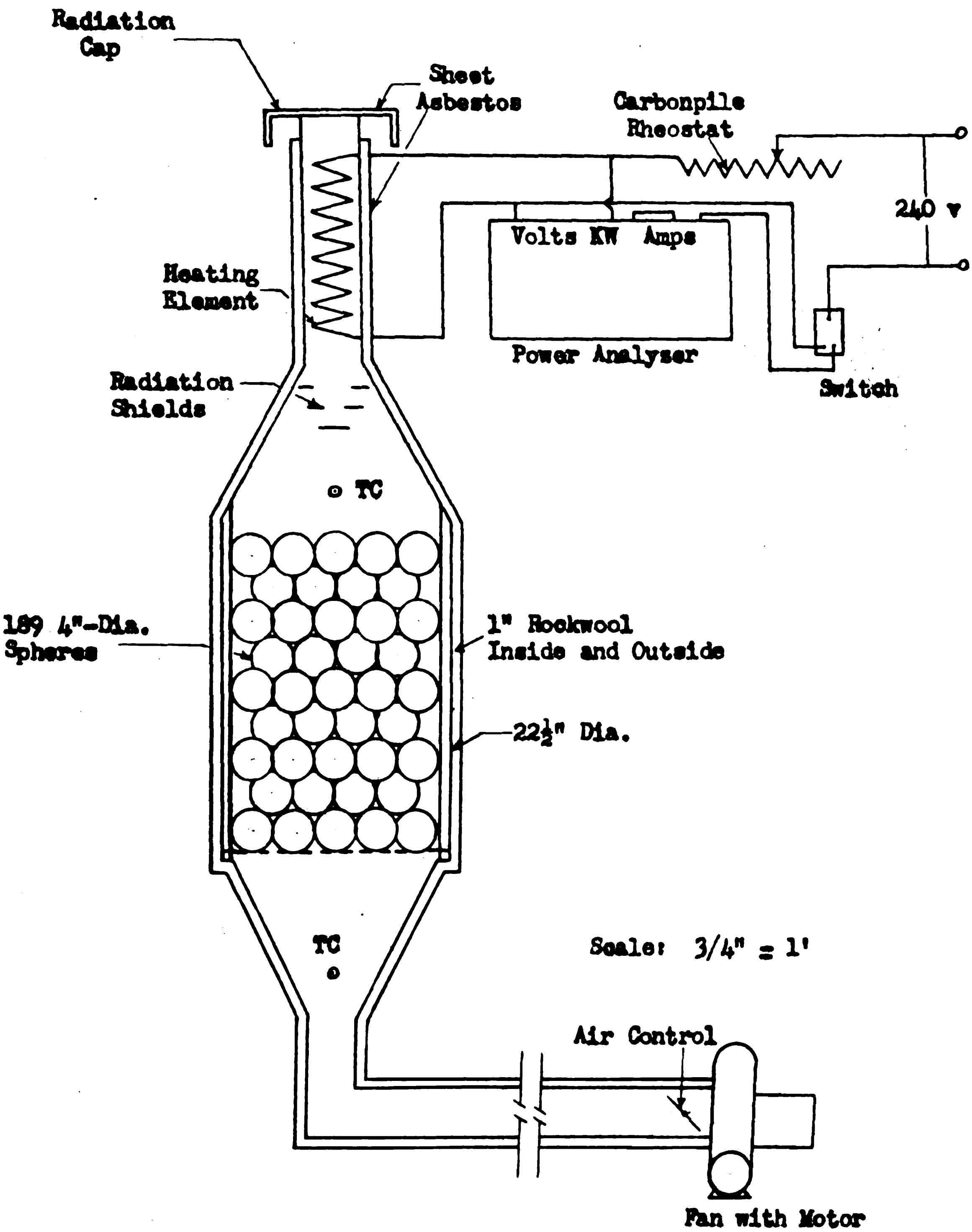


Fig. 14. Cross-sectional view of the storage unit and its supplementary components.

specifications of Madison (33). The orifice plate with the micromanometer was used to calibrate the vane anemometer. Flow rate values by the orifice were found to be 61.5 to 64 percent of those recorded by the anemometer. The previously recorded readings of the anemometer were corrected accordingly.

The unit was constructed so as to have the warm air directed into the top layer of spheres, passing through the entire unit, and exiting at the bottom after releasing the heat. The heating process will progress on through toward the bottom layers. While in storage, any convection currents set up will have a tendency to move the heat from the lower layers to the top ones. When the heat is recovered, the air flow is reversed, bringing the cooler air over the lower temperature spheres first and progressively heating as it moves to the top.

To reduce radiation losses from the heater as much as possible, a cap was placed over the six-inch diameter intake pipe. It was necessary to install the radiation shields between the heating elements and the top layer of spheres after preliminary tests indicated that the top layer of spheres was heating to a higher temperature than that of the air passing over them. It was evident that the spheres could "see" the higher temperature heating element and were receiving radiant energy from the element. The shields proved to be valuable also in that they provided sufficient turbulence of the incoming air to give approximate even distribution of the velocity and temperature pattern over the top layer of spheres.

Pitot tube and thermocouple probes were made to determine this distribution.

A metal grate at the bottom of the unit provides support of the spheres and a minimum drop in static pressure. The static pressure drop across the main body was found to range from 0.004 to 0.09 inches of water for the mass velocities used in the tests. The spheres were placed in layers as shown in the photographic view of the top layer in Figure 15. There were nine layers with 21 spheres per layer, making a total of 189 spheres in the unit. Originally 16 control spheres with thermocouples were made. However, No. 9 was damaged in the preliminary test and was not used in the storage unit. Placement of the other control spheres in the unit was made according to Table III. Description of the instruments used in the tests are listed in Appendix B.

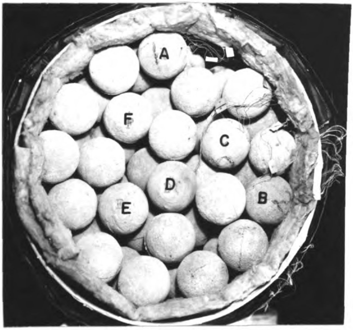


Fig. 15. Orientation of spheres in top layer of storage unit with notation of control sphere placement.

TABLE III

PLACEMENT OF CONTROL SPHERES IN SYSTEM

Layer	A	B	C	D	E	F
1	1	2	3	4		
4	5	6	7	8		
7	10	11	12	13		
9			15		14	16

F. Plan of Tests and Measurements

Basically, the desired information from the proposed tests was the effectiveness of the stone spheres as a heat storage unit. Measurement of this effectiveness must be accomplished by measurement and calculation of several heat transfer characteristics. For immediate application, optimum values or description of the following characteristics would be desirable:

1. Surface conductance, h.
2. Mass velocity, G.
3. Effectiveness of unit during heating period.
4. Effectiveness as a storage unit.
5. Quantity of recoverable heat.
6. Dimensional ratio of a prototype bed.
7. Economical aspects of heat storage.

To accomplish the above, tests were designed to have variations over the range of the equipment. The surface conductance variation was obtained by changing the mass velocity of the air within the limits of the fan. Different temperature rises of the incoming air were made possible through the variable heat input.

Measurements of air temperature by thermocouples were made at the following points: (1) outside ambient, (2) approach to spheres, (3) exit from spheres, (4) at anemometer. Temperatures were measured in the control spheres, which were placed according to Table III. For heat-loss determination, thermocouples were placed on the surface of (1) heater cylinder, (2) top frustum, (3) storage cylinder, (4) bottom frustum. The wet and dry bulb temperatures of the outside air were obtained from a sling psychrometer and the air velocity with the anemometer every 30 minutes. Also, input electrical power was measured at the same time interval.

The 12-point recording potentiometer provided a reading on each point every minute. Readings of the air outside, incoming to spheres, exiting from spheres, and at the anemometer were made every minute. All other temperature readings were taken every eight minutes through the use of the switching mechanism shown in Figure 16. Barometric pressures were obtained from a mercury barometer in a nearby building.

The general procedure used involved (1) a heating period, (2) a period for holding the heat or storage period, and (3) a cooling or recovery period. The length of a heating and recovery periods depended on the mass velocity of the air. At lower velocities, the period was extended to as much as five hours, while higher velocities reduced it to as little as one hour. The fan and heater were shut off when the difference between the incoming and outgoing air temperatures was 10 to 20°F during heating. The fan was stopped during cooling or recovery when the air temperature difference was 5 to 10°F.

The main storage period during the tests was 24 hours. During this period, checks were made at intervals to determine the heat retained. Shorter storage periods of 4, 9, and 12 hours were used to provide possible comparative studies with the longer duration.

Fig. 16. General view of the storage unit.

(1) Storage unit
(2) Thermocouple switches
(3) Recording potentiometer
(4) Sling psychrometer
(5) Vane anemometer and stop watch
(6) Power analyzer
(7) Carbon pile rheostat
(8) Fan

A total of nine tests was completed. However, the first four were eliminated from use, because the radiation shields were not in place. The latter five tests provided data for a wide range of air flow rates and quantities of heat stored.

VI. RESULTS AND DISCUSSION

A. Determination of the Surface Conductance Coefficient

Several methods are available for determining the surface conductance, h, under forced convection conditions. Löf and Hawley (31) recommend the relation

$$h = 0.79\,(G_o/d)^{0.7} \tag{8}$$

for the determination of the surface conductance coefficient in "builder's gravel." Application of Equation 8 to the proposed heat storage system appeared impractical as extremely high surface conductance values are obtained when calculating for the 4-inch diameter. As the equation was developed for a small size rock, an error for use with a larger rock is quite possible.

McAdams (32) recommends for a single sphere a relationship which was derived from data of several investigators in the form of

$$\frac{h_m\, D_s}{k_f} = 0.37\left(\frac{D_s\, G}{\mu_f}\right)^{0.6} \tag{9}$$

that holds true in the range of $D_s\, G/\mu_f$ from 17 to 70,000. However, the spheres in the storage unit act more like a bank of staggered tubes with an effective diameter being determined by

$$\frac{1}{D_e} = \frac{1}{D_s} + \frac{1}{D_s}\,. \tag{10}$$

The recommended formula for nine layers of spheres then becomes

$$\frac{h_m D_e}{k_f} = 0.492 \left(\frac{D_e G}{\mu_f}\right)^{0.553} \tag{11}$$

where $\frac{D_e G}{\mu_f}$ ranges from 2,000 to 40,000.

The h-values predicted by Equations 9 and 11 for the five tests are presented in Table IV for both heating and cooling conditions. The values given by the latter equation are 14 to 25 percent higher, which would be expected for staggered banks of spheres as compared to a single. It is, therefore, expected that the surface conductance coefficients calculated by Equation 11 would best predict the conditions actually occurring around the spheres. The values ranged from 2.51 to 6.09 for the heating phases of the tests. This range approaches the one found by the subsequent method.

An alternative method for determining the surface conductance coefficients is by constructing curves similar to the theoretical ones in Figure 9 from data obtained in tests. Data for No. 3 sphere in the top layer were used for this determination, as the temperature of the incoming air was known and was approximately constant. Resulting curves are presented in Figure 17.

Relative positions of these curves are according to the mass velocity, with the higher velocities allowing the sphere to reach maximum possible heat absorption first. Qualitative characteristics of the curves are very much the same as the theoretical ones in Figure 9. The curves plotted from observed data approach asymptotically a value of 0.55 Btu per °F difference; whereas, the

TABLE IV

PREDICTED VALUES OF THE
SURFACE CONDUCTANCE COEFFICIENT
BY KNOWN RELATIONSHIPS

Test	Process	Mean t_f °F	w, #/hr	G, #/(hr)(ft^2)	h_m Eq. 9	h_m Eq. 11
A	Heat	122.5	147	320	1.87	2.51
	Cool	80.0	238	518	2.39	2.97
B	Heat	120.1	309	674	2.87	3.54
	Cool	73.1	532	1160	3.83	4.56
C	Heat	95.5	394	859	3.41	3.96
	Cool	79.0	474	1033	3.62	4.32
D	Heat	100.8	559	1220	4.06	4.83
	Cool	73.3	705	1538	4.83	5.36
E	Heat	87.5	863	1883	5.22	6.09
	Cool	85.7	965	2105	5.55	6.41

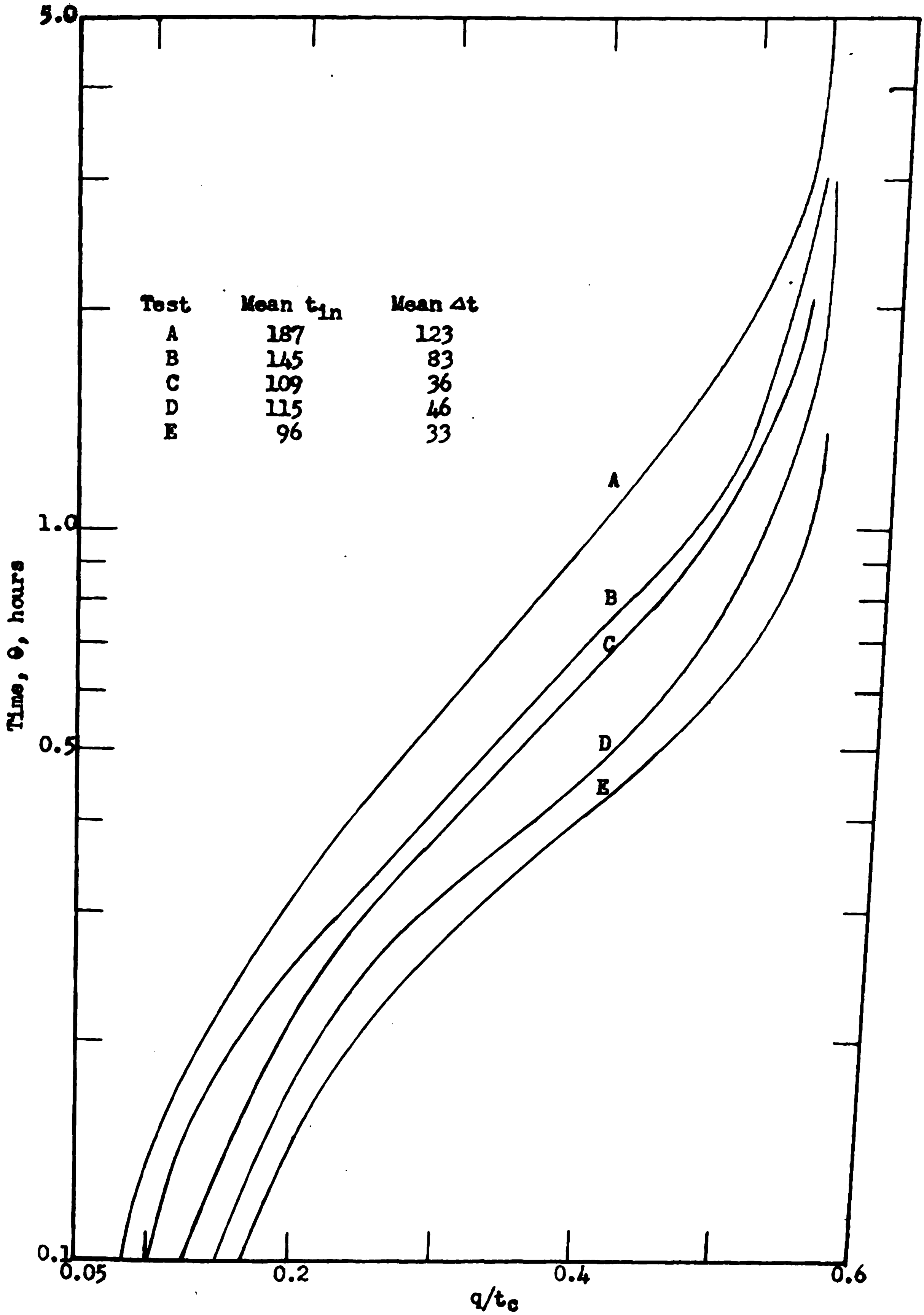


Fig. 17. Observed cumulated heat per degree temperature difference in single sphere.

theoretical value approaches 0.5 in the same manner. Several factors could be responsible for this discrepancy. (1) Radiation from the heating coils could be reflected by the shields and frustum. (2) Measurement of the thermal and physical characteristics, such as specific heat, density, or thermal conductivity, may not be of sufficient accuracy, although the same corresponding values appeared in both the observed and theoretical data.

By superimposing the observed results in Figure 17 over the theoretical in Figure 9, it could be noted that the entire set of observed data would fall approximately between the curves having h-values of two to slightly beyond six. This range of coefficients fits very closely to the one obtained by Equation 11. Basically, the two methods produced comparatively close results, which give confidence in the validity of the data.

The importance of these findings is that only a slight advantage is obtained by producing high surface conductance coefficients by means of high air velocities. The lowest mass-velocity rate allowed the spheres to receive 72.8 percent of the asymptotic value in one hour. On the other hand, the highest velocity rate, 5.9 times greater than that of the lowest rate, almost reached the asymptotic value at the end of the same period. This vast increase in power requirement for moving the air is not justified when considering that the lower velocity system will have 92 percent of the maximum possible heat stored in one additional hour.

It should be noted from these curves that the effectiveness as a heat exchanger does not depend on the temperature difference between the incoming air and initial temperature of the sphere. The mean temperature difference varied from 33°F at the highest velocity to 123°F for the lowest velocity. The time required for the sphere to reach asymptotic condition depends on the surface conductance and not Δt. Such characteristic would allow the solar collector to operate at the lower air temperature rise which gives a higher operating efficiency (13). Compatibility between the storage unit and the collector is obtained in this respect.

B. Effectiveness during Heating

Although the mass velocity and surface conductance study produced a qualitative view of the unit's effectiveness, the quantitative aspect is of equal importance. Ultimately, the quantity of heat which is later recoverable for use is the primary objective.

Control spheres were placed in the unit at various locations in order that reliable observations could be made on heat absorption and release throughout the unit and within a sphere. Temperature gradients within the spheres proved to be important only at the beginning of a cooling or heating period. This gradient was as high as 26°F difference between the surface and center in top layer spheres, but always less than 10°F difference in the lower layers. However, the larger difference occurred only when higher incoming air temperatures were used. Maximum temperature gradients observed during all the five tests are plotted in Figure 18. This family of gradients was plotted from sphere 3 data during Test A when the

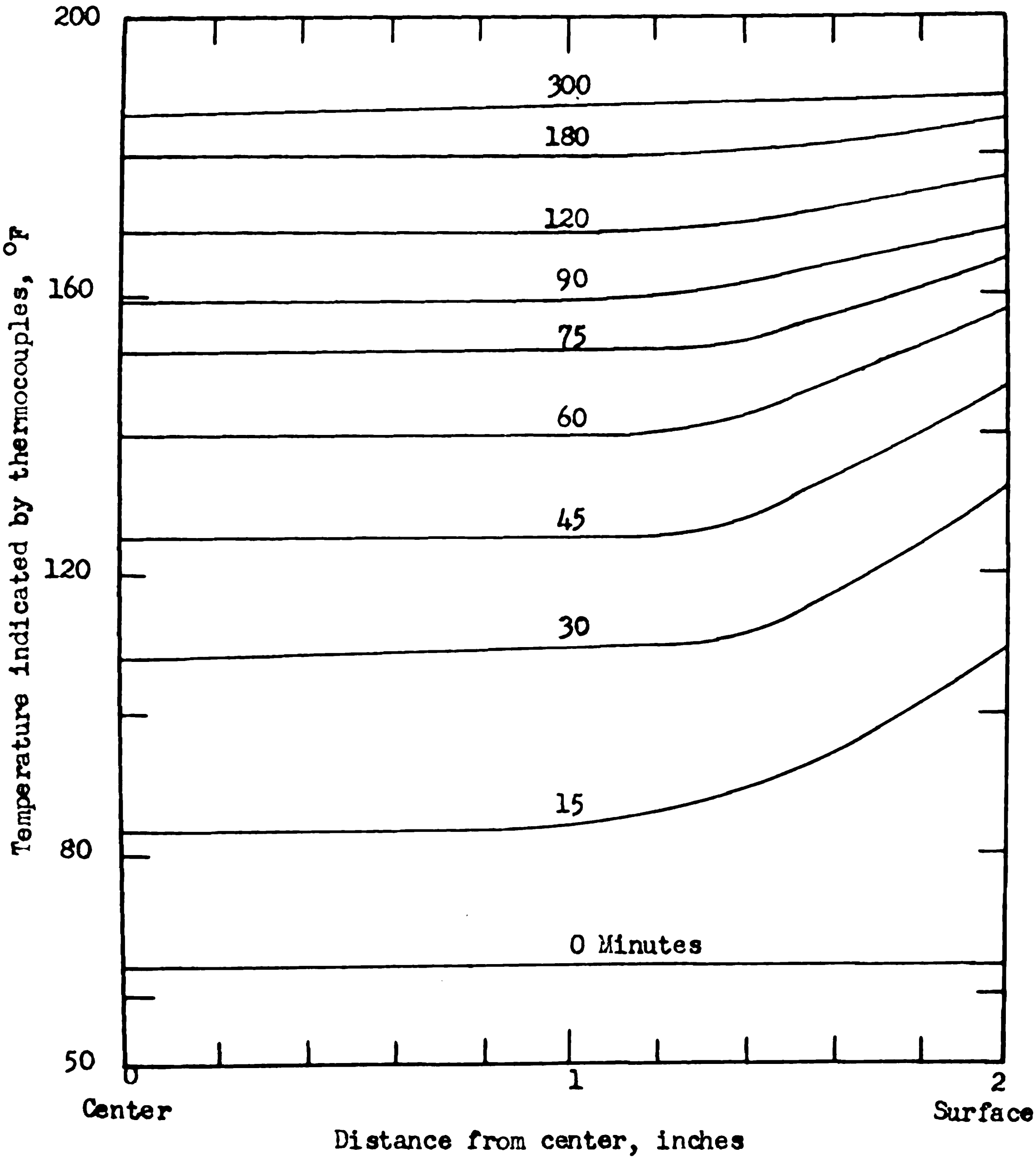


Fig. 18. Maximum temperature differences observed within sphere 3, Test A, with 198°F incoming air temperature.

incoming air temperature was 198°F. Note should be made of the gradient being maximum at 15-minutes and reducing on to practically zero after 300 minutes.

A mathematical check was made to determine the value of using all four thermocouple readings within a sphere, as compared with only the center and surface ones, for calculating the heat absorbed. At most, the error, when using only the two, was less than 5%. The curves in Figure 18 show that averaging will always give values too high. The gradient within the sphere when making oven tests was generally larger; however, this was probably because of radiation it received from the insulated walls.

Determination of total heat absorbed in the storage unit was made by projecting the quantity absorbed by control spheres into the other spheres according to their location. Figure 15 shows that the spheres can be thought of being arranged roughly in two concentric circles around a central sphere. The outside ring has 13 spheres, while the inner circle is made of seven spheres. At least one control sphere was located in the two rings and at the center for Layers 1, 4, and 7. Layer 9 has three control spheres located in the inner ring.

After calculating the quantity of heat each control sphere absorbed for each 15-minute period of heating, heat absorbed by plain spheres in a given layer was found by assuming that they receive the same amount of heat as a control sphere in that corresponding ring. Summation of individual heat quantities gives the total energy received in one layer. Values for Layers 1, 4, 7, and 9 were plotted at 15-minute intervals in Figures 19 through 23 for the five heating tests, respectively. Curves were then drawn through the known points, which made it possible to pick off the quantity of heat absorbed at other layers. It is noted that Layer 1 is the top section in the storage unit.

The initial temperatures on the graphs are for Layers 1, 4, 7, and 9 in each case. Temperatures of the incoming air over the top layer and the mass velocities are also listed on each of the five graphs.

Operational characteristics, such as incoming air temperature, mass velocity, and initial temperature of the spheres, are different for each tests. Although this prevents making direct comparisons between graphs to a certain extent, it provides a means of predicting the reaction within a storage unit. Results of Test A are presented in Figure 19. The mean incoming air temperature of 187°F was the highest for all tests, and the mass velocity was the lowest. Initial temperatures of the spheres varied only 5°F from the top to the bottom layer. The quantity of heat stored was higher for this test because of high temperature of incoming air.

Qualitative results derived from Figure 19 are as follows: (1) The top layers heated at the most rapid rate during the first part of the tests, with an increase of heating rate for the other layers after about 90 minutes. The heating rate can be distinguished by the spacing between the time lines, with the wider spacing representing higher heating rates. (2) Only a small quantity of heat was added during the last two hours (180 to 300 minutes)

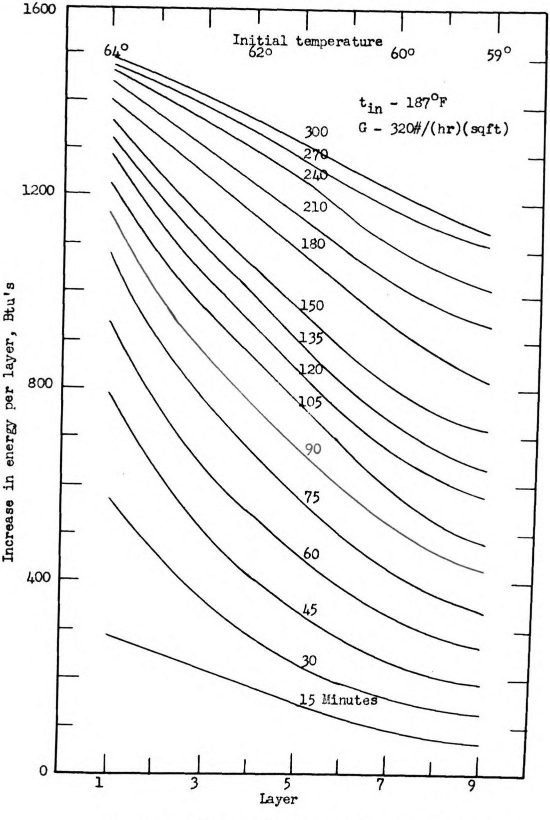


Fig. 19. Accumulated energy during heating, Test A.

as compared to the quantity received during the first three hours. This shows a marked drop in the efficiency of heat transfer which will be explained more in detail later. (3) The low mass velocity allowed the upper layers to continue to have an increase in heat level even after five hours. This is demonstrated by the substantial negative slope of the heat level line even after 300 minutes.

In Test B, Figure 20, the mean incoming temperature has been reduced and the mass velocity increased. Initial temperature variation of the spheres is somewhat reversed with the higher temperatures in the lower layers. A similar type of heating process was had as in Test A, with the exception that the heating rate drops off after only two hours. Only about 10 percent additional heat was added during the third hour as compared with that which had been absorbed during the first two hours. Note should be made also that the heat level line began to approach more of a horizontal line than did the curves in Figure 19. This is primarily because of the higher velocities.

The distinguishing change seen in the curves for Test C, Figure 21, is the flattening of the heat level curves at a very rapid rate. This characteristic is due to two factors. First, the heat level line at any one layer is based on the initial temperature as a reference point. Note that the initial temperature of the upper layer is $10^{\circ}F$ higher than the lower layer. It is evident that the lower layers have more storage potential for a given incoming air temperature as compared to the upper layers. This is the reason that

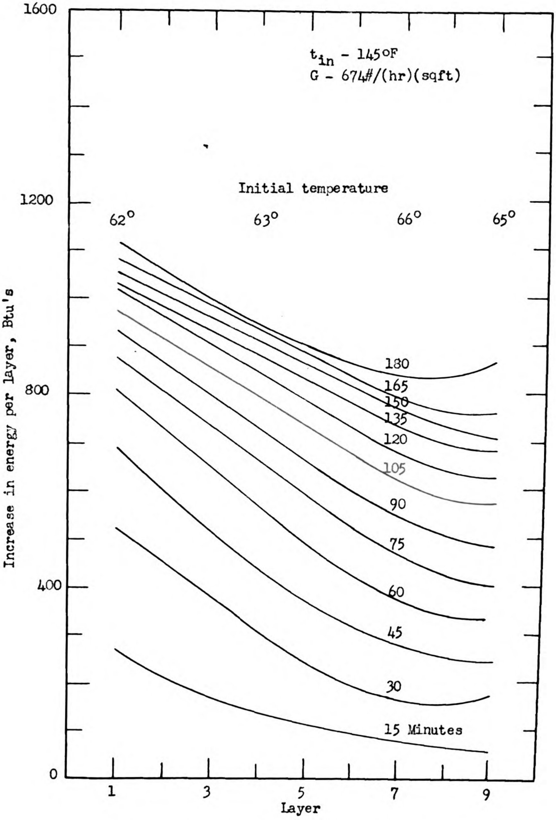


Fig. 20. Accumulated energy during heating, Test B.

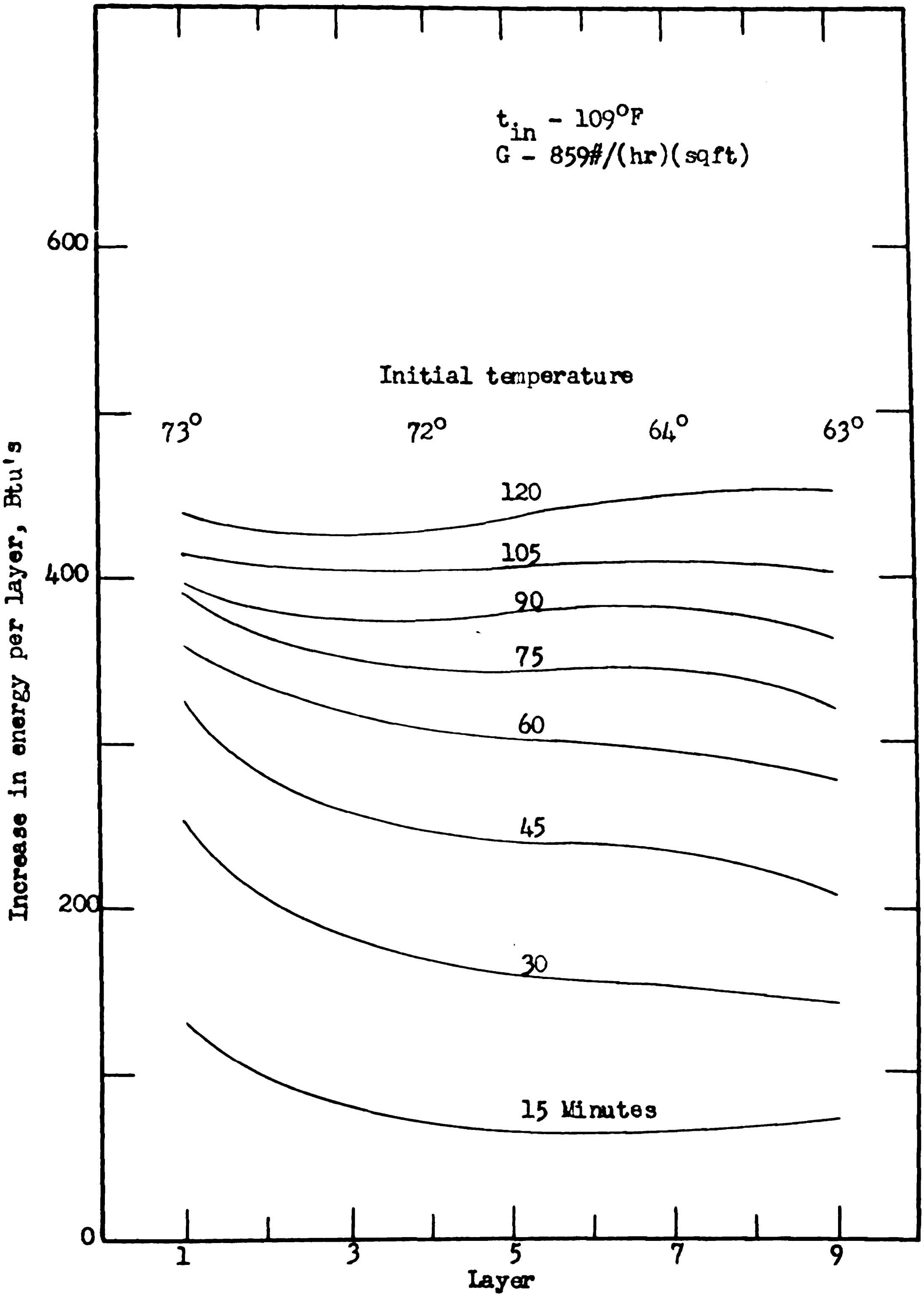


Fig. 21. Accumulated energy during heating, Test C.

heat level lines can be higher for the lower section after a period of receiving heat.

The second reason for the lines flattening out quickly is the higher mass velocity, which gives the heated air a shorter period of time to release the heat in the upper layers. Air will remain at a higher temperature while going through the storage unit. This allows the lower spheres access to higher temperature air and a possibility of receiving more heat. Whereas, at lower velocities, only lower temperature air reached the bottom spheres.

Test D, Figure 22, had a very pronounced heat level increase for Layers 6, 7, 8, and 9. This resulted from the high initial temperature gradient of 15^{o}F, which was greater than for any other test. Test E in Figure 23 approached the same qualitative results found in Test C. The initial temperature gradient was not as pronounced,and the heat level lines had a tendency to become horizontal after 90 minutes of operation.

The effectiveness of the regenerative storage unit during heating periods was determined quantitatively on the basis of the percent of available heat the air releases to spheres during the progress of storage. The available heat was based on the difference between the incoming air and initial sphere temperatures, and the mass velocity of air. Two sets of plots were made of this study: (1) percent of available heat released during any 15-minute period, and (2) the cumulative percentage of available heat released up to any time during the operation. These results are presented in Figures 24 and 25, respectively.

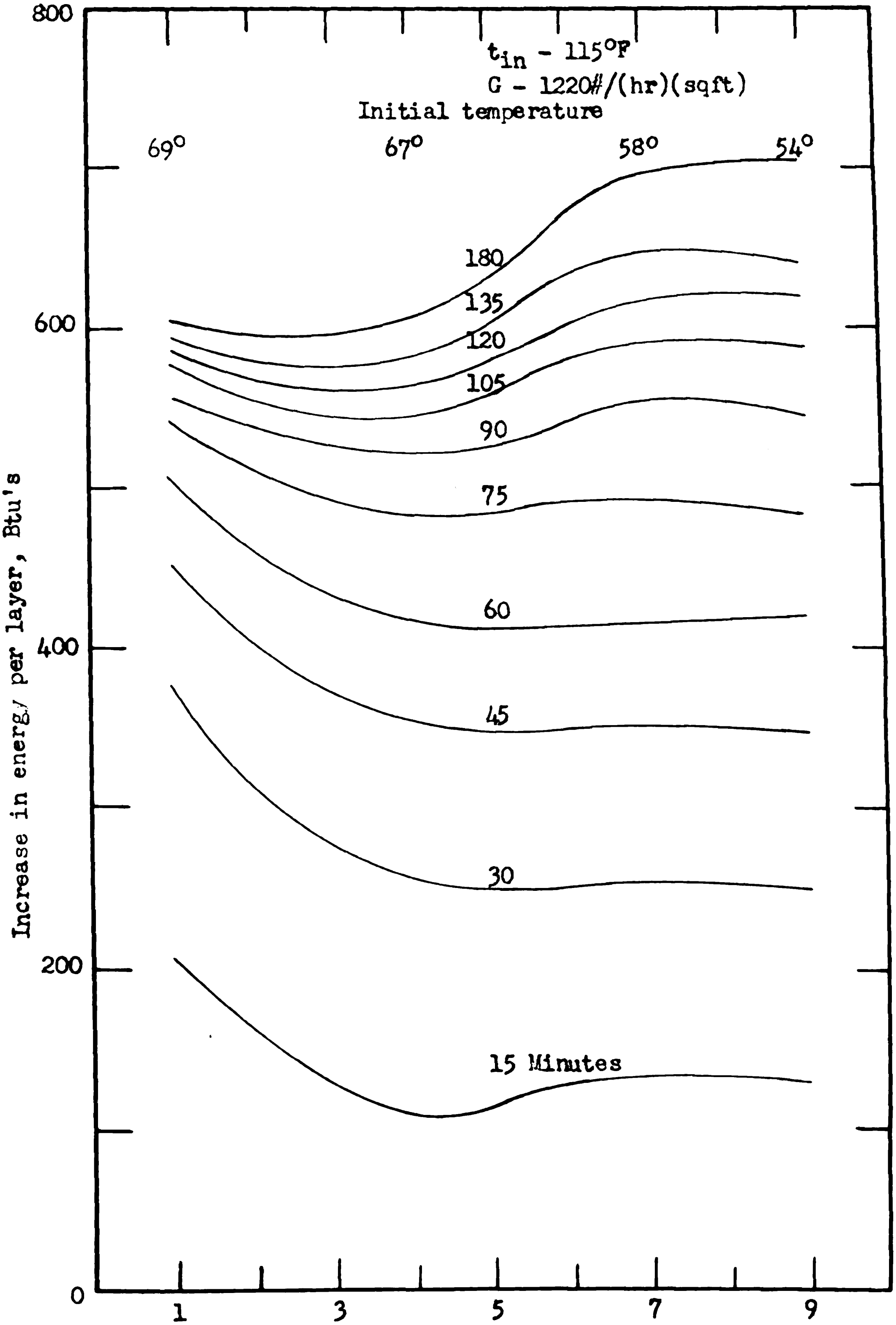


Fig. 22. Accumulated energy during heating, Test D.

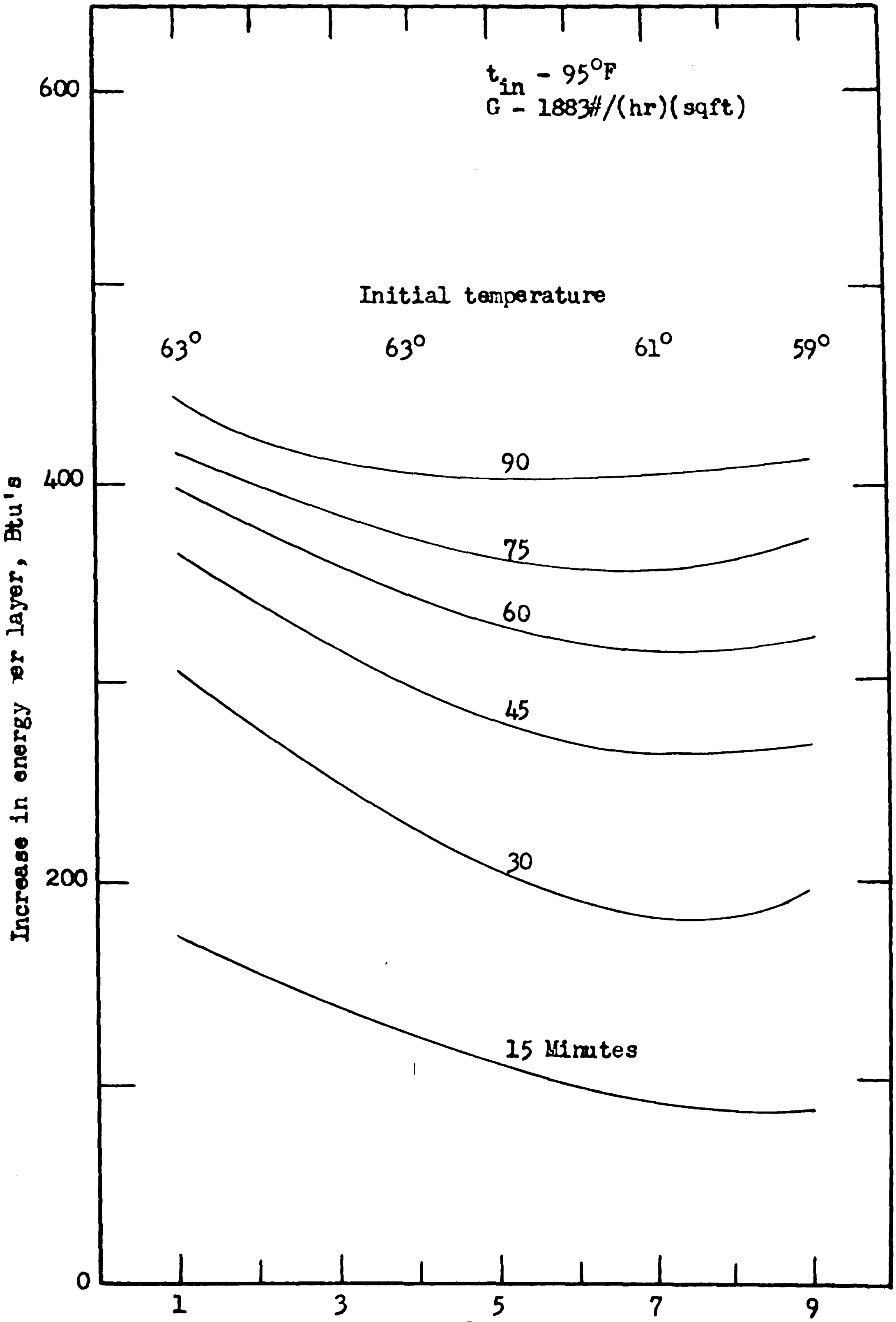


Fig. 23. Accumulated energy during heating, Test E.

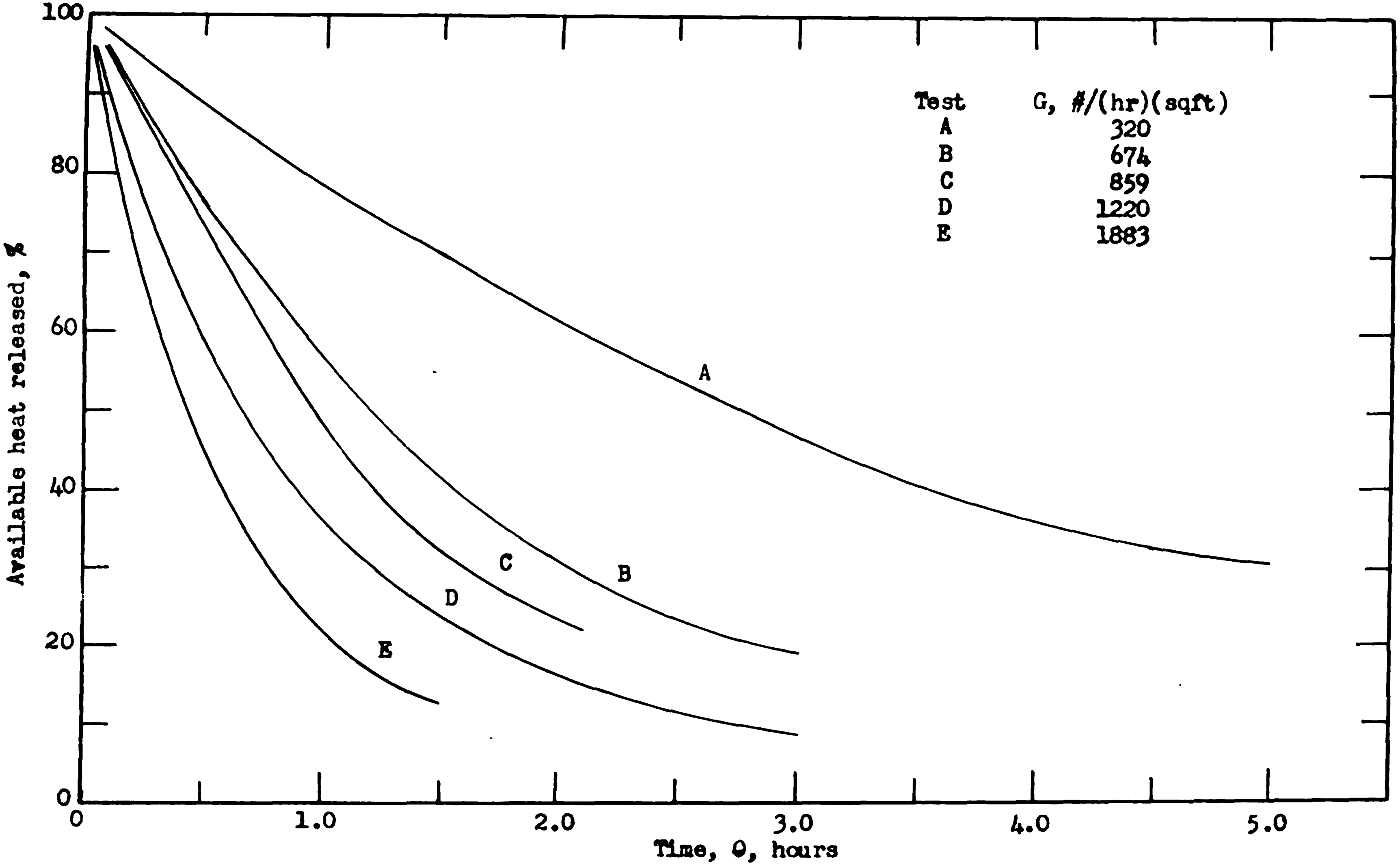


Fig. 24. Heat released in cylindrical storage unit during 15-minute period.

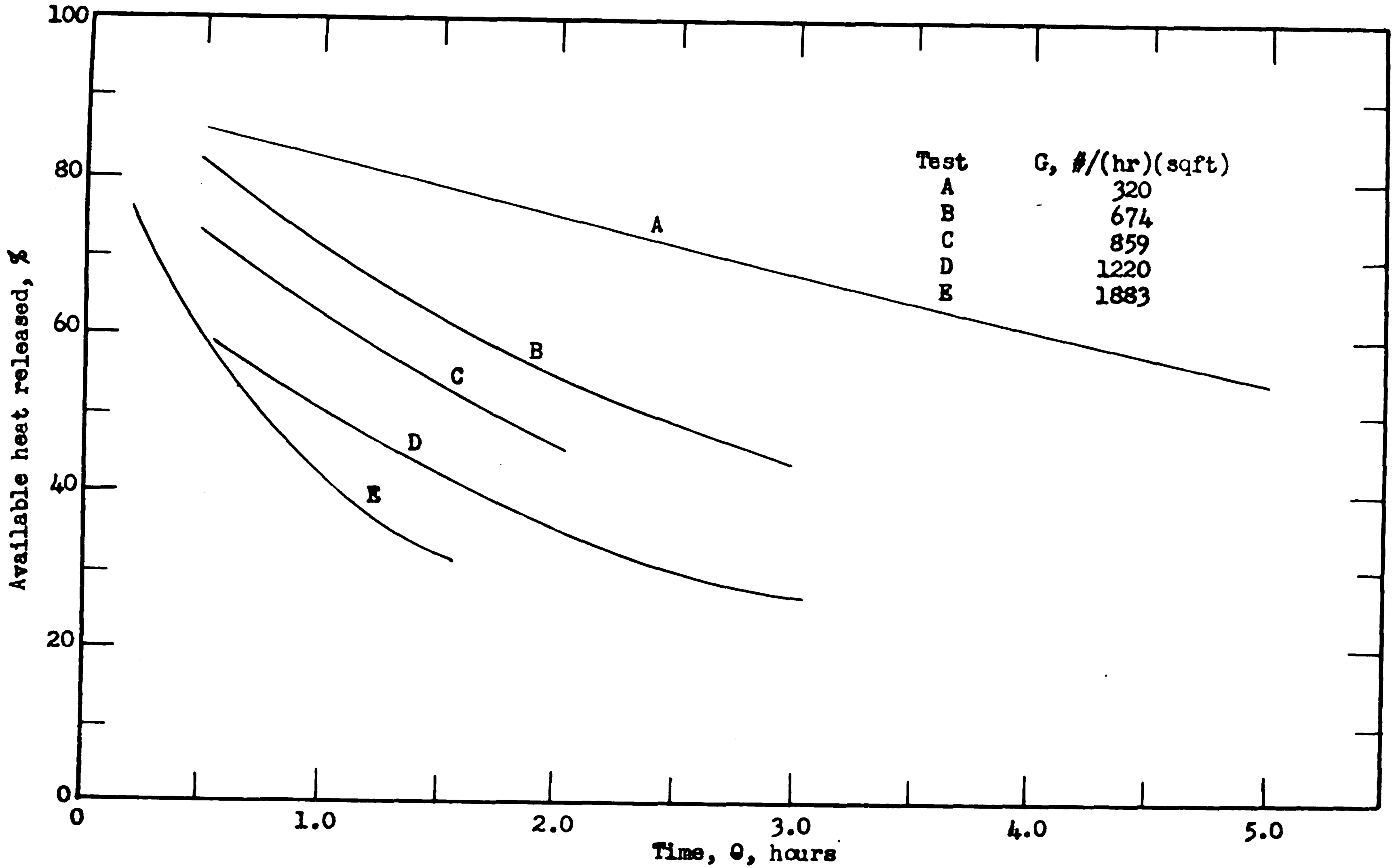


Fig. 25. Cumulative percentage of available heat released to spheres.

The points at the end of the first 15-minute period were erratic because of the transient heating conditions. However, the succeeding points plotted into relatively smooth curves. The most outstanding feature these curves present is that higher efficiencies of heat absorption were obtained with low mass velocities. The instantaneous efficiency (Figure 24) for the air mass velocity of 320 lb per (hr)(sqft) was 70 percent at the end of $1\frac{1}{2}$ hours. For the velocity of 1883 lb per (hr)(sqft), it was only 13 percent at the end of the same period. The values of efficiency for the intermediate tests range accordingly between these two extremes.

Cumulative efficiencies of available heat released present a similar picture in Figure 25, but the sharpness of drop is not so pronounced, as would be expected. The heat released after $1\frac{1}{2}$ hours operation is 79 percent of available heat for the lowest velocity, as compared to 33 percent for the highest velocity. Lower velocities, therefore, appear to be more suitable from the standpoint of higher efficiency of heat release and lower cost for moving the air through the system.

C. <u>Effectiveness as a Storage Unit</u>

Heat retention of a storage unit during the holding period will depend essentially on four factors: (1) temperature gradient between the spheres and the surrounding medium, (2) the insulation effectiveness of the material enclosing the storage unit, (3) the length of holding period, and (4) the ratio of the storage material bordering the sides of the container to the total volume.

In the storage tests presented in Figure 26, the temperature gradient did not appear to have the effect on the heat retention characteristics as might be expected. The gradient was much higher in Test B; however, the percent of heat retained was slightly greater during the same period of 12 hours. Length of holding period has a considerable effect on the percent of heat retained in storage. For this small unit, the percent retained dropped to 55 after 12 hours, and then to 33 after 24 hours of storage. This low percentage of retention after 24 hours lies to the fact that 81 percent of the spheres in this small unit make up the outside boundary, which allows heat loss to be at a relatively higher rate than in a larger unit. This factor will be projected into the larger unit in the section devoted to application.

Additional study of the storage was made of the layer profile with regard to the heat retained in Figures 27 through 30. The distinguishing change of layer profile during storage was a more rapid cooling of top and bottom layers as compared with center layers. A hump or higher heat level appears after about four hours of storage as a result of convection cooling in the top and bottom of the unit. In each of the four storage periods, heat loss was more pronounced on the lower side. These curves indicate that convection at the areas where air enters or exits must be kept at a minimum to reduce storage losses.

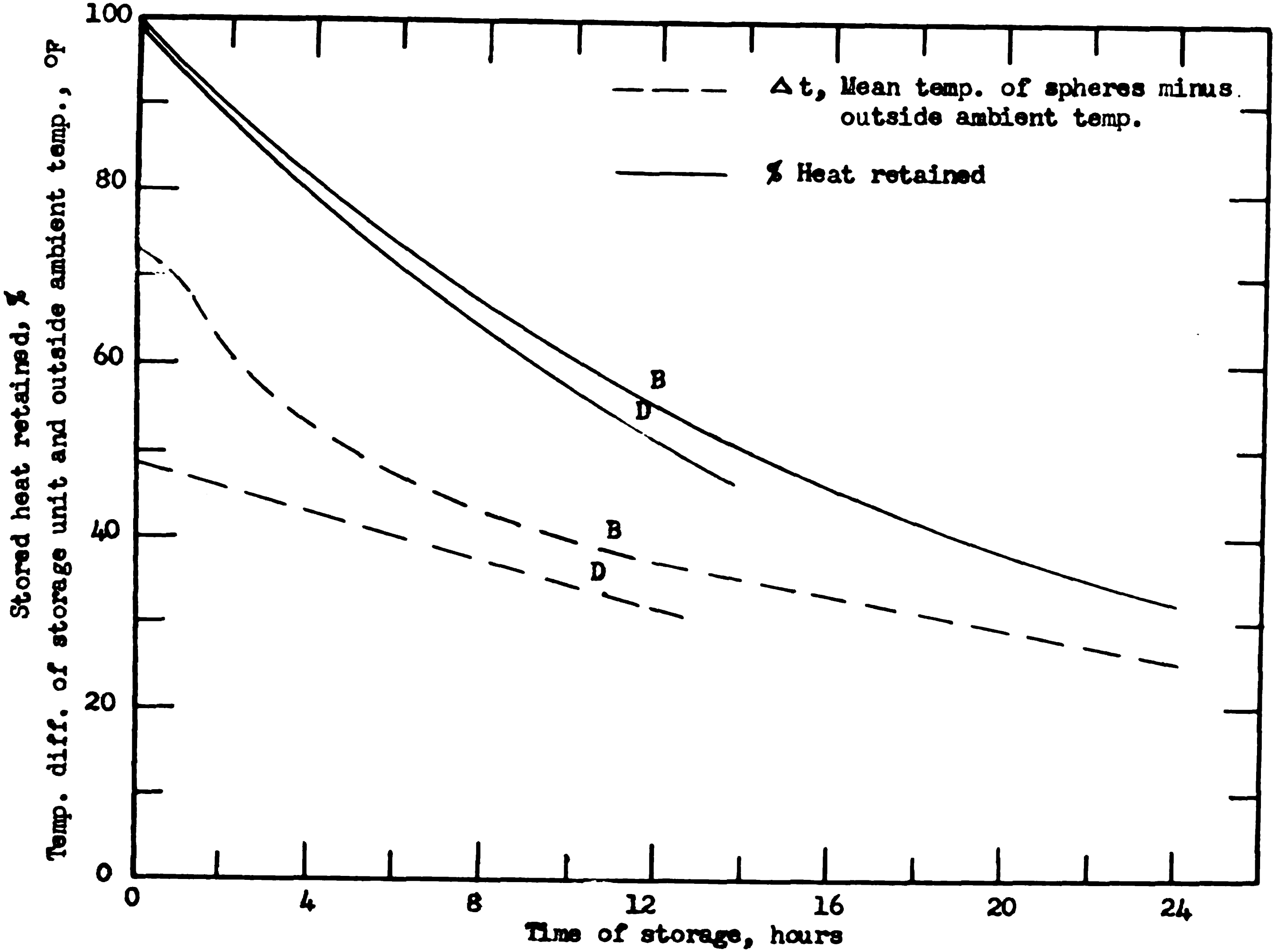


Fig. 26. Heat retention characteristics of storage unit.

D. Heat Recovery Characteristics

The quantity of heat recovered after storage is the most important factor governing justification for a unit. The cooling or heat recovering curves are given in Figures 27 through 31 for the five tests. These curves present qualitative characteristics during heat recovery. As the air flow has been reversed and cold air is coming up through the bottom layers first, it is expected that this section will cool at a faster rate. The disappearance of the storage "hump" in the heat level curves takes place at a rapid rate. The curves then approach a straight line, which at the end of each test has only a small slope.

Summation of heat levels for each layer at a specified time in Figures 27 through 31 will give the total quantity of heat in the system. These total energy levels are listed in Table V for the beginning of the storage period, end of storage period, end of cooling period, and total energy recovered.

Note should be made that the Btu level in the unit depends on some reference. All calculations of the cumulative energy were made on the basis of initial temperature of spheres, which had to be adjusted to the incoming air temperature at time of cooling to give a true picture of recovery. Comparisons between tests can more easily be made by heat recovery characteristics which are summarized in Table VI. On the basis of available heat in the unit at end of

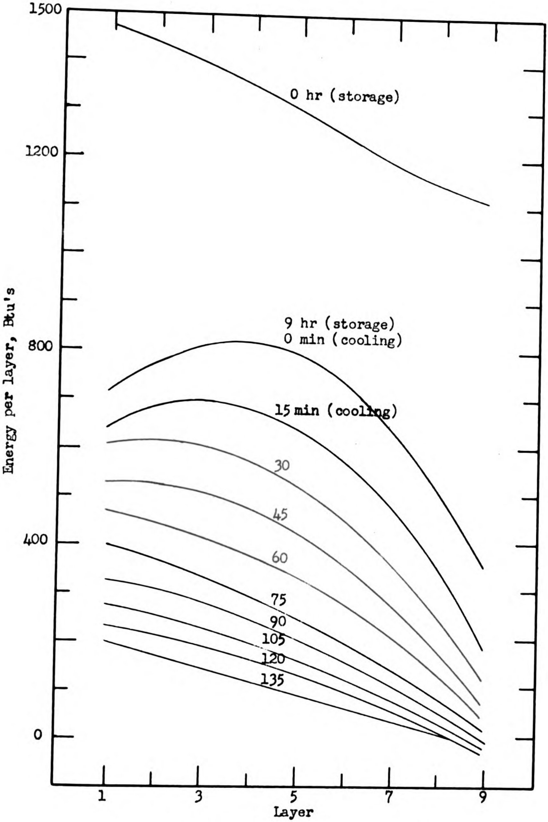


Fig. 27. Heat level during storage and cooling, Test A.

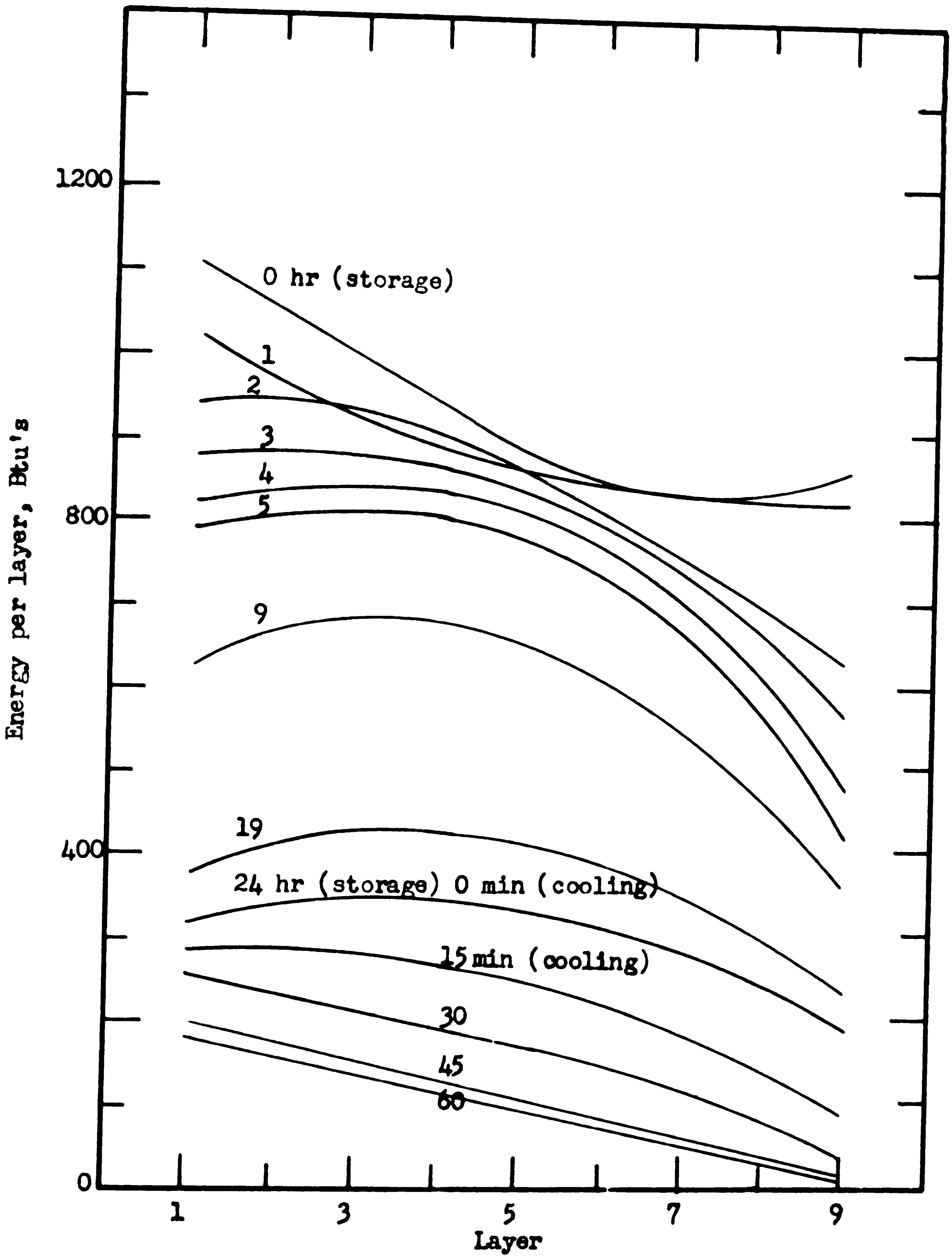


Fig. 28. Heat level during storage and cooling, Test B.

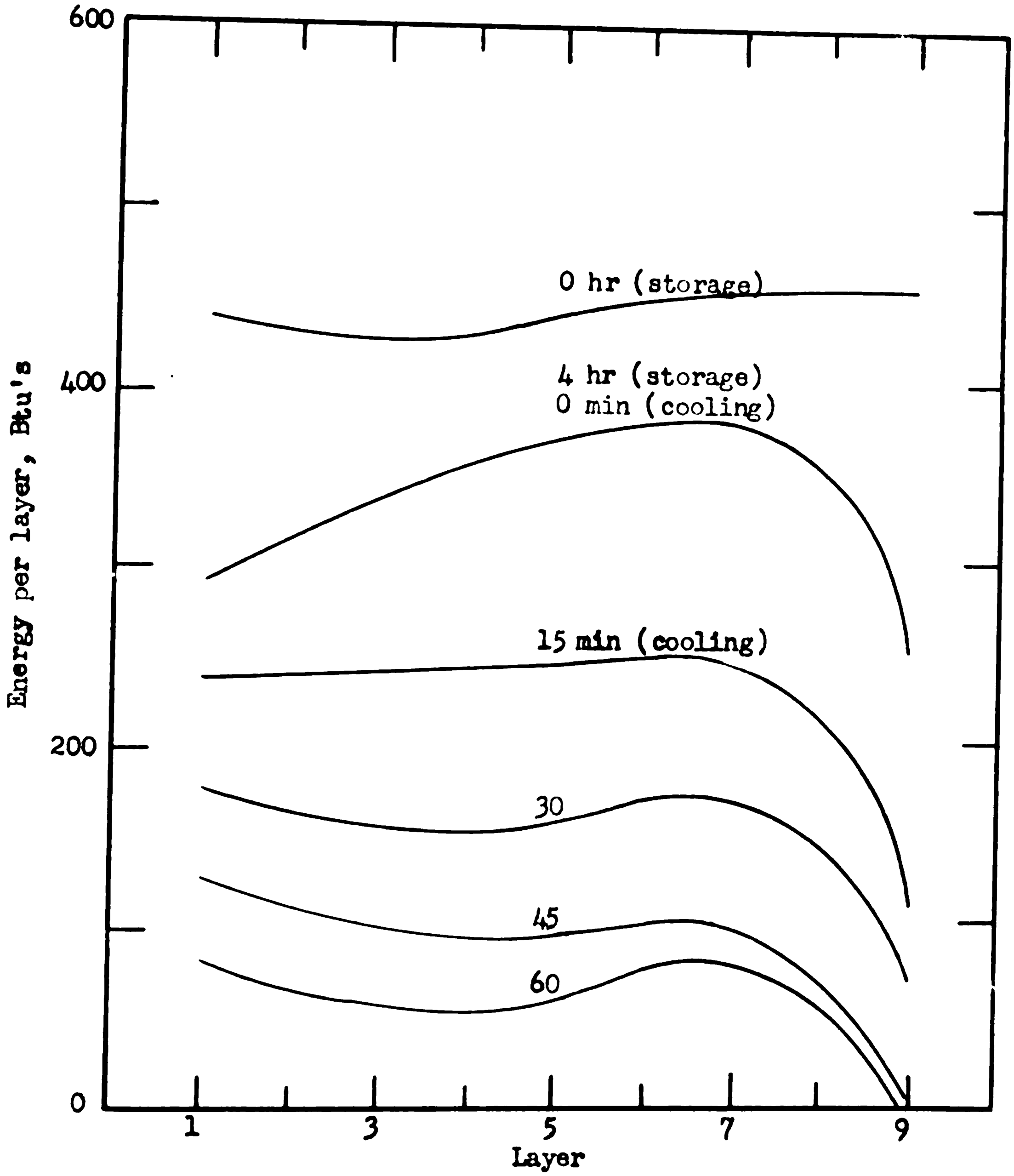


Fig. 29. Heat level during storage and cooling, Test C.

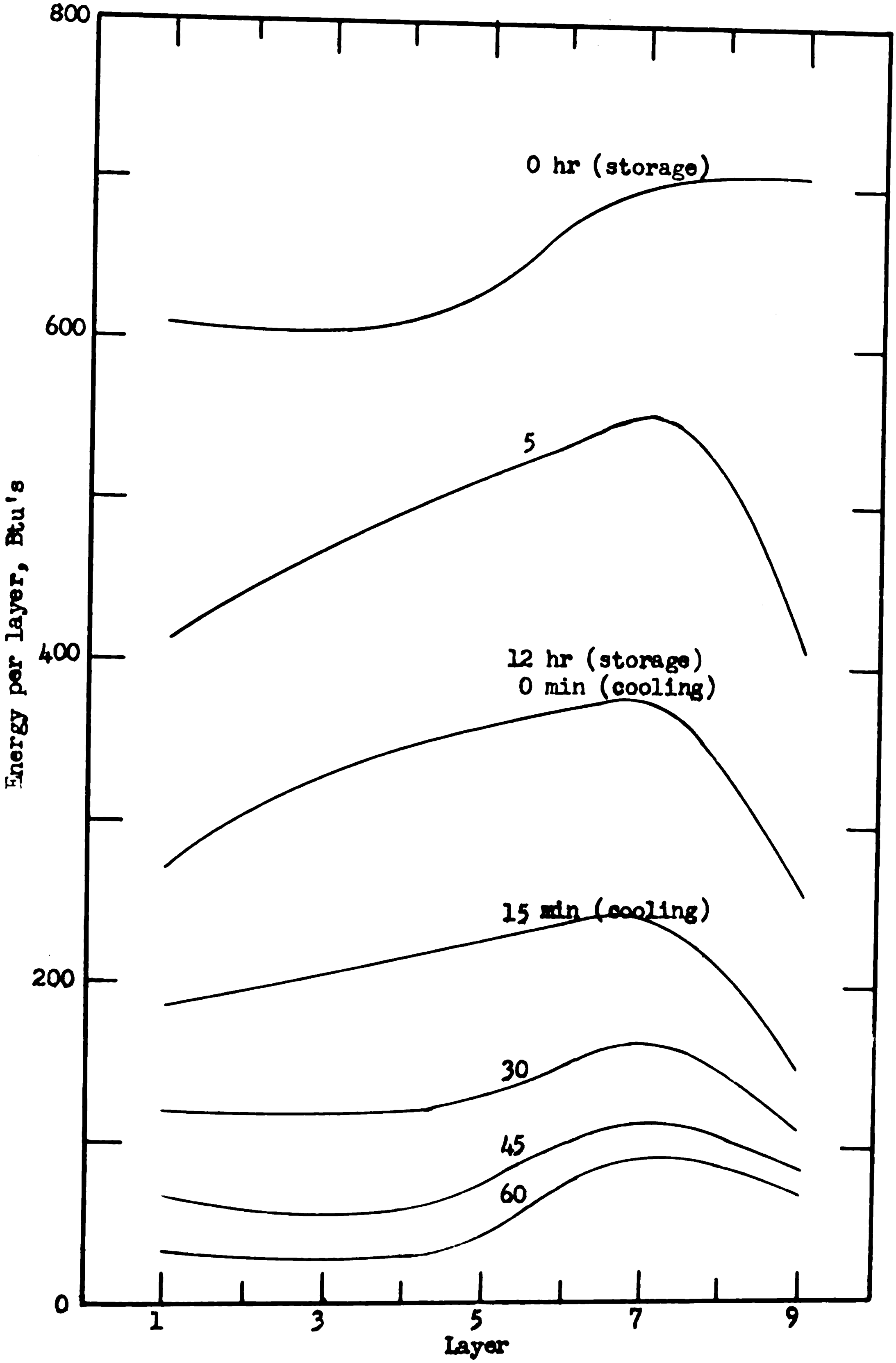


Fig. 30. Heat level during storage and cooling, Test D.

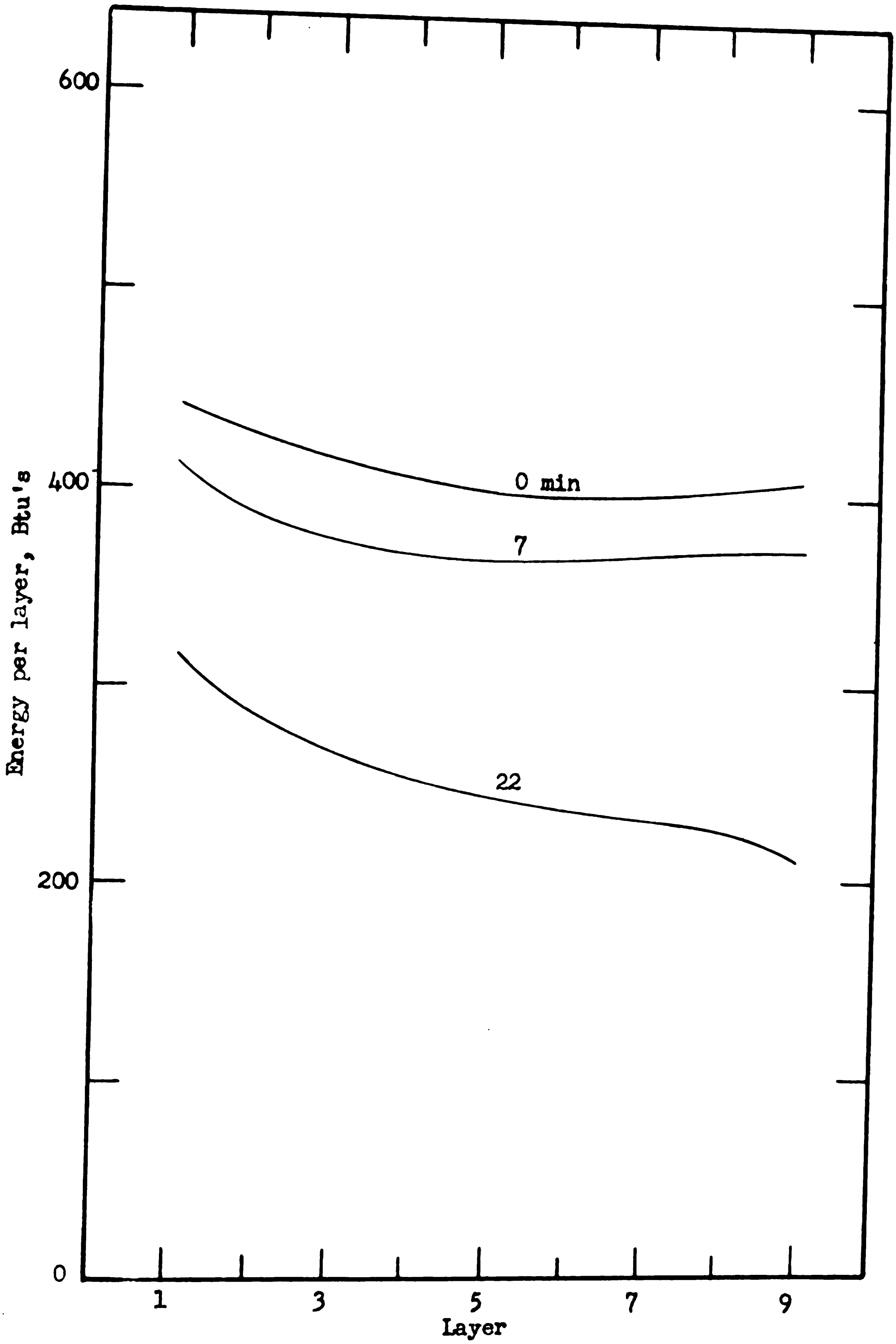


Fig. 31. Heat level during cooling, Test E.

TABLE V

TOTAL ENERGY LEVEL OF
STORAGE UNIT DURING TESTS

Test	Energy Stored, Btu	Storage Period, Hrs	Energy after Storage, Btu	Energy after Cooling, Btu	Recovered Energy, Btu	Adjusted* Recovered Energy, Btu
A	11,821	9	6,193	1,013	5,180	4,863
B	8,394	24	2,723	885	1,838	2,240
C	3,987	4	3,067	513	2,554	2,026
D	5,796	12	2,953	471	2,482	2,320
E	3,704	0	3,704	2,278	1,426	1,715

* Reference temperature of energy level adjusted due to difference between initial sphere temperature prior to heating and incoming air temperature upon cooling.

storage period, the lower mass velocity will have a more efficient rate recovery than at higher velocities.

Table VI shows that Tests A and E had an air temperature rise of 10°F or more. This means that the system was turned off when the temperature rise of air passing through the storage unit fell below 10°F. Fractions listed for recoverable heat should be representative in both cases. The other three tests, B, C, and D, have temperature rises of less than 10°F at the end of the tests. The fraction of available heat recovered will be higher than if the operation had been stopped when the 10°F temperature rise was reached. The values, 0.822, 0.660, and 0.787, are all too high in this respect.

Comparison between Test A and E proves that lower velocities are most suitable in heat recovery. At the lower velocity, 78.6 percent of the available heat was recovered, but only 46.2 percent was recovered for the highest velocity.

TABLE VI

SUMMARY OF HEAT RECOVERY CHARACTERISTICS

Test	Storage Period, Hrs	G, #/(hr)(sqft)	Air Temp. Rise Range, °F	Time of Recovery, Hrs	$\frac{\text{Btu Recovered}}{\text{Btu Stored}}$	$\frac{\text{Btu Recovered}}{\text{Btu Available}}$
A	9	518	61-10	2.5	0.412	0.786
B	24	1,160	30-6	1.0	0.267	0.822
C	4	1,033	33-9	1.0	0.508	0.660
D	12	1,538	28-5	1.0	0.390	0.787
E	0	2,105	31-10	0.37	0.462	0.462

VII. APPLICATION

A. Cost of Operation

Although the economical feasibility of the storage unit will depend on a number of factors, comparisons will be made here of operational cost, assuming that the equipment for heating with fuel oil would cost approximately the same as the storage unit. It is also considered that the collector will be available for other use and is not primarily for the storage unit. Utilization of energy from a storage system requires the movement of air one additional time over normal direct use of heating with oil. The charge for operation is based on the power required to move the air. Comparison between cost of oil and storage of solar energy is to be made on the following assumptions and calculations.

No. 1 Fuel Oil

Heat value, 136,000 Btu per gallon
Cost, 17 cents per gallon
Combustion efficiency, 70%
Unit cost, 1.785 cents per 10,000 Btu

Stored Solar Heat
Total heat input, 20,550 Btu
Incoming heated air, 50^{o}F above initial sphere temperature
Air flow rate, 200 cfm [320 lb/(hr)(sqft)]
Total fan pressure, 4 inches water
Total fan efficiency, 50%
Power requirement, 0.251 hp
Heating period, 2 hours (selected arbitrarily)
Heating efficiency, 76% (Fig. 25)
Storage period, 72 hours
Retention during storage, 67.2% (calculated below)
Recovery efficiency, 78.6% (Table VI)
Heat stored, 15,600 Btu
Heat recovered, 8,240 Btu

Total cost at 2.5 cents per Kwhr, 1.25 cents
Unit cost, 1.518 cents per 10,000 Btu

The fan horsepower requirement was calculated from formula by Madison (33),

$$hp = \frac{(0.000157)(CFM)(\text{Total pressure in inches water})}{\text{Total efficiency}} .$$

Cost of operating the electric motor was made on the basis of 1000 watts required per horsepower. The retention of heat during storage is based on the surface-volume ratio of a twelve-foot cube, in the same relation that was had with the small laboratory unit.

This larger unit has a ratio of spheres at the surface to total number of spheres of about 15 percent and has a heat loss of 67 percent at end of 24 hours storage (Figure 26). Then the ratio between this large unit and the laboratory model with a surface-to-volume ratio of 81% is

$$\frac{67}{81} = \frac{x}{15}$$

or $$x = 12.4\%,$$

the loss of heat at the end of 24 hours. Projecting it to 72 hours, the storage retention would be 67.2 percent.

The heating period is a critical factor in the economic balance. The optimum operation period which will make the solar energy equal to that of fuel oil under the assumptions above is about 3.1 hours. Periods of heating longer than this will make the solar energy cost more, and, in like manner, shorter periods will cost less than the fuel oil as a heat source.

B. Storage Material Used

For a practical operation, the concrete spheres would not be used, but instead field stones for several reasons. (1) The process of molding spheres is entirely unnecessary when field rocks of comparable size are available in many areas right on the farm. (2) The field stones are superior in the quantity of heat retained for a given temperature difference. Although the specific heat is virtually the same for both, the density is 37.2 percent higher for the stones. This gives the field stones a heat retaining capacity of 1.38 times that for the concrete spheres. (3) Thermal conductivity of the natural occurring rock ranges from 1.0 to 1.5 Btu per (hr)(sqft)(^{o}F) per ft, as compared to 0.37 for the test spheres. This factor would increase the speed of heating, making the theory of Newtonian heating or cooling more pronounced. An advantage of the higher k-value is the reduction of heating time. However, a disadvantage is the possibility of higher heat loss rates. (4) The factor of roughness of the field stones would have an effect on the surface conductance coefficient. Increased roughness would give higher h-values for equal mass velocities.

C. Container for Storage Material

Maximum value can be derived from the solar storage unit only when it is constructed within the structure utilizing its heat. It is obvious that any heat loss from the unit would go directly into the structure, thereby, increasing its effectiveness. Two possible locations within the structure are (1) a compartment located on the ground floor, or (2) an excavated hole.

The first type of structure would take up usable floor space that would normally be used for production or storage of the farm commodities. The side walls would require considerable bracing for retaining the rocks. The second proposal would involve the expense of excavation, but the only building material required, other than the necessary duct work, would be a vapor barrier to keep moisture out of the system. In wetter areas, drain tile should be laid around the perimeter to help reduce moisture troubles. Good vapor barriers are aluminum foil with building paper padding to reduce damage and protected plastic sheeting.

D. Shape

Basically, the most effective shape for storage is a container with the greatest volume to surface ratio. The normal shapes in order of decreasing value are the sphere, cylinder, and cube. For ease of construction and air passage, the cylinder and cube are preferable. The longer air flow path will provide a more effective heat exchanger. Within reasonable pressure drops across the unit, large L/d ratios would be desirable. The directional movement of air in and out should follow the same paths as the laboratory unit.

E. Size

The size of a storage unit will naturally depend upon the heat load of the utilizing activity. For comparative purposes, an actual problem of removing moisture from a poultry house has been set up in the Appendix D, based on work of Esmay and Moore (22). The sizes of collector and storage units were calculated with the assumption that the energy required would be obtained on the probabilities of 80, 60, and 40 percentages. From the data it can be noted that higher probabilities of receiving a given amount of radiation require the largest collector and the smallest storage unit. The storage unit is smaller because of the higher air temperatures obtained by the collector. The largest storage unit required to store the 484,000 Btu at a 35.5°F temperature rise would be about 541 cubic feet. This is equal to a cube 8.15 feet on a side.

Fig. 32. Storage unit in operation.

VIII. CONCLUSIONS

Charts, developed on the availability of daily solar radiation rates at East Lansing, provide a designer of solar-energy equipment with basic design information on quantities of heat expected (Figures 1 through 6). It was concluded that the cumulative probability curves were highly reliable since the plots from 14 years of raw data matched very closely normal curves (Figures 1 and 2). Selection of the probability, for finding the daily normal, maximum, and minimum radiation rates, must be made by the designer on the basis of experience and good judgment. Balance must be made between importance of consistent energy rates and the allowable investment. Months having small coefficients of variation will have less variability of available energy. These coefficients will aid the designer in selecting the probability and adapting equipment to other localities.

The optimum size rock for solar-energy storage was determined to be four inches diameter (Table II). This provides maximum rate of heat storage at minimum pressure drop across the system. Lower mass air velocities of 320 lb per (hr)(sqft) provide the greatest heat transfer effectiveness and most economical operation. This velocity will produce a surface conductance coefficient of about two for the spheres. Except at high temperature differences, the concrete spheres could be considered Newtonian heating or cooling.

The effectiveness of the spheres in heat absorption is reduced considerably after receiving heat for three hours with a constant incoming temperature. About 68 percent of the available heat will be absorbed during this period (Figure 25). The top layers heat rapidly at first and then the heat front moves on to the lower layers.

If the storage unit is within the building where the heat is utilized, up to 78.6 percent of the stored heat can be recovered. Large units, such as a 12-foot cube, will retain about 68 percent of the stored heat after three days (page 84). Smaller units will lose the heat during storage at a higher rate because of a higher percentage of the spheres bordering a surface. During storage the center layers retain the largest percentage of heat. Means should be provided to prevent convection currents around the rocks during storage (page 72).

When heating a given quantity of air in a prototype unit, it was found that the larger the collector used the smaller the storage unit necessary for storing a given amount of energy.

IX. SUMMARY

The objectives of this project were to (1) statistically analyze daily radiation data for East Lansing in the determination of frequency of various rate levels for each month, as would affect utilization and storage of solar energy, (2) mathematically design and construct a solar storage unit, and (3) perform operational tests on the solar energy storage unit under laboratory conditions.

Quantity of available solar radiation at any locality is the determining criterion in the design of solar utilization equipment. Daily solar-energy data for 14 years at East Lansing were analyzed. Charts developed were monthly probability curves and monthly coefficients of variation for normal, maximum, and minimum daily rates. The probability of a given job is selected by balancing between the importance of consistent energy rates and the allowable investment in equipment. Once this selection has been made, the probability curves give quantitative rates expected. The coefficients of variation aid in selecting a probability and adapting solar-energy equipment to other localities.

A study of heat storage methods proved that rocks would be the best material for agricultural use. Analysis by heat transfer principles indicated that the 4-inch diameter rock would provide the maximum rate of heat storage at minimum pressure drop across the system. Thermal conductivity, specific heat, and density of a special concrete mixture were determined. The 4-inch diameter spheres of this mixture were used in a laboratory storage unit.

Copper-constantan thermocouples in 15 control spheres provided information on rate of heating, retention of heat, and rate of heat recovery from the spheres. Observations proved the spheres to react very closely to theoretical solutions. Lower mass air velocities of 320 lb per (hr)(sqft) provided the greatest heat transfer effectiveness and the most economical operation. This velocity provided a surface conductance coefficient of two for the spheres. The heating and cooling of the spheres could be considered essentially Newtonian.

Tests showed that the effectiveness of the spheres in heat absorption was reduced considerably after subjected to heated air for three hours. About 68 percent of the available heat was absorbed during this period, with the top layers heating rapidly at first and then the lower layers.

Up to 78.6 percent of the stored heat was calculated to be recoverable, when using a 12-foot cube storage unit within the building where the heat was utilized. It was also found that 68 percent of the heat stored could be remaining in the storage unit after three days. Faster heat losses at ends of storage unit during storage indicated that convection currents must be reduced to conserve heat.

In a prototype unit, the storage material would be of well sized and selected 4-inch diameter field rocks. Placement of the storage unit within the building where the heat is utilized will increase efficiencies. The shape should be cylindrical or cubical.

Calculations showed that a 7-ft cube storage unit with stones could furnish a poultry house 25,700 Btu/hr for drying 16 hours a day. This was based on an 80 percent probability in January.

X. RECOMMENDATIONS FOR FUTURE RESEARCH ON STORAGE

Storage means, other than sensible-heat methods, should constantly be examined for possible use with solar energy in agricultural work. Further study of sensible heat storage in rocks should be carried out in a proto-type unit to determine the heating, storage, and recovery characteristics. A larger unit in series with a solar collector could determine the compatibility of the two under conditions of maximum operation efficiencies for each. Such study would enable an accurate cost analysis and determination of the feasibility of incorporating a unit in farm buildings for the purpose of reduction of humidity, crop drying, and other uses.

In any further study with small laboratory units, it is suggested that the container be square to enable perfect close packing of the spheres. Also, it is recommended that the inside walls of the container have a smooth uniform surface, such as wood. These factors will favor more uniform air flow and heating across any one layer.

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XII. APPENDIX

APPENDIX A

Nomenclature

a - $(g\beta\rho^2 c)(\mu k)$

A - Surface area of heat transfer, sqft

c - Specific heat, Btu/(lb)(^{o}F)

C - Coefficient of variation, dimensionless

d - Diameter, ft

D_s - Diameter of sphere, ft

D_e - Effective diameter of sphere when compared to tubes, $1/D_e = 1/D_s + 1/D_s$

e - Base of Napierian logarithms

g - Acceleration due to gravity; 4.17×10^8 ft/hr^2

G - Mass velocity, lbs/(hr)(sqft of cross-section)

G_o - Superficial mass velocity, lbs/(hr)(sqft of cross-section without particles)

h - Surface resistance, Btu/(hr)(^{o}F)(sqft)

h_c - Surface resistance in natural convection, Btu/(hr)(^{o}F)(sqft)

h_m - Mean surface resistance

k - Thermal conductivity, Btu/(hr)(sqft)(^{o}F per ft)

k_f - Thermal conductivity at film temperature, t_f

L - Length of heat exchanger, ft

M_n - Roots of the transcendental equation, Nu $= 1 - M_n$ Cot M_n

Nu - Nusselt number, $(h\ 4r_h/k)$

q - Instantaneous heat rate, Btu/hr, or Btu/day

APPENDIX A (Cont.)

Nomenclature

Q - Cumulative heat, Btu

r - Radius of sphere or any point within, ft

r_h - Hydraulic radius, ($4r_h$ = hydraulic diameter)

r_o - Surface radius, ft

s - Standard deviation

t - Temperature at specified time, ^{o}F

t_c - $t_i - t_f$

t_f - Film temperature, $(t_s + t)/(2)$, ^{o}F

t_i - Initial temperature, ^{o}F

t_s - Surface temperature, ^{o}F

V - Volume of sphere, cuft

$\bar{x}$ - Mean of a group of data

α - Thermal diffusivity, $(k/c\rho)$, sqft/hr

β - Coefficient of thermal expansion of a fluid; reciprocal degrees Fahrenheit

θ - Time of fluid flow, hr

μ - Absolute viscosity, lb/(ft)(hr)

μ_f - Viscosity at film temperature, t_f, lb/(ft)(hr)

ρ - Density of material, lb/cuft (also denoted w)

APPENDIX B

Instruments Used in Tests

1. Ammeter, Simpson Electric Co., Chicago. Measurement: Milliamperes.

2. Barometer, Central Scientific Co., Chicago. Measurement: Inches of Mercury.

3. Guarded hot plate complete with constant voltage transformer, coil rheostats, water pump, and thermocouples. Mechanical Engineering Department, Michigan State University. Measurement: Thermal Conductivity (by calculations).

4. Inclined Oil Micro-manometer. E. Vernon Hill & Co., Lake Geneva, Wisc. Measurement: Inches of Water.

5. Industrial Analyzer, Model 639, Type 2, No. 4161. Weston Electrical Instrument Corp., Newark, N. J. Measurement: Volts, Amperes, and Kilowatts.

6. Orifice, 2.5 in. dia. in 3 in. tube. Special made. Measurement: Used with Inclined Manometer.

7. Potentiometer, No. 300083, A. E. No. 1776. Leeds & Northrup Co., Philadelphia. Measurement: Millivolts.

8. Potentiometer, Recording, Serial No. 860990, 12 point, Minneapolis-Honeywell. Measurement: ^{o}F, Range 50 to 350^{o}F.

9. Potentiometer, Recording, Serial No. 327582, 12 point, Minneapolis-Honeywell. Measurement: ^{o}F, Range -100 to 250^{o}F.

10. Sling Psychrometer. Tycos, Rochester, N. Y. Measurement: ^{o}F, wet and dry bulbs.

11. Vane Anemometer, No. 5963, Serial No. 13947, 6 in. dia., Keuffel & Esser. Measurement: Feet, with stopwatch feet per minute.

12. Voltmeter, Type A025, Model VAX8M. General Electric. Measurement: Volts.

APPENDIX C

Summary of Pertinent Data

	Tests				
	A	B	C	D	E
Heating:					
Period, hrs.	5	3	2	3	1.5
Dry bulb temp., °F	62.2	64.5	67.2	68.5	65
Wet bulb temp., °F	50.6	54.4	55.0	56.1	52
Barometric pressure, in. Hg.	29.62	29.3	29.6	29.47	29.45
Wattage input	1,497	2,099	1,302	2,004	2,153
Total heat input, Btu	25,550	21,500	8,900	20,500	11,020
Humidity, lb. H_2O/lb.dry air	0.00514	0.00672	0.00642	0.00686	0.00543
Correction for Air (28):					
Specific heat	1.004	1.005	1.005	1.005	1.004
Density	0.997	0.9965	0.9965	0.996	0.997
Sp. vol. at B.P. ft^3/lb	14.21	14.52	13.91	14.36	13.96
Sp. vol. at 29.92" Hg. ft^3/lb	14.10	14.23	13.83	14.16	13.75
Flow rate, lbs/min	2.44	5.15	6.56	9.32	14.39
Flow rate, cfm	34.6	118.7	91.2	133.8	200.6
Mass velocity, lbs/(hr)(sqft)	320	674	859	1220	1883
Storage: Period, hrs.	9	24	4	12	0
Cooling:					
Period, hrs.	2.5	1	1	1	0.367
Dry bulb temp., °F	57.1	63.3	63.0	61.8	65.0
Wet bulb temp. °F	47.6	54.0	51.7	51.7	52.0
Humidity, lbs H_2O/lb dry air	0.00486	0.00687	0.00543	0.00586	0.00543
Sp. vol. at B.P., ft^3/lb	13.19	13.59	13.21	13.33	13.57
Sp. vol at 29.92" Hg. ft^3/lb	13.02	13.32	13.13	13.16	13.38
Flow rate, lb/min	3.96	8.85	7.90	11.65	16.08
Flow rate, cfm	52.0	103.8	103.8	155.5	218.0
Mass velocity, lbs/(hr)(sqft)	518	1160	1033	1538	2105

APPENDIX D

Example Problem of Storage Unit Application

The following example is given to explain the use of the data developed in the main body of the thesis. Data for the problem was based on a mechanized poultry house designed and studied by Esmay and Moore (22). Description of this building is as follows:

Location: Lapeer Co., Michigan
Size: 36 ft. by 108 ft.
Capacity: 2200 laying hens
Ventilation: Forced air
Lowest inside temperature allowable: 42^oF
Heat production by hens: 72,600 Btu/hr
Moisture production by hens: 60.5 lb/hr
Moisture removed by cleaner: 19.6 lb/hr
Moisture to be removed by fans: 40.9 lb/hr
Inside temperature: 42^oF
Inside relative humidity: 80%
Outside temperature: 12^oF
Outside relative humidity: 70%
Heat loss through side walls and ceiling for 30^o temperature difference: 24,000 Btu/hr
Month: January

<u>Calculations</u>:

1. Pounds of air to move:

$$M = \frac{W_e}{(RH_2)(W_{s2}) - (RH_1)(W_{s1})} \qquad \text{Ref. (6)}$$

$M =$ air flow per hour, lb

$W_e =$ moisture to be exchanged per hour, lb.

RH_1, $RH_2 =$ relative humidity of incoming and outgoing air respectively

W_{s1}, $W_{s2} =$ water in saturated air vapor mixtures at temperature of incoming and outgoing air, respectively, lb/lb dry air.

$$M = \frac{40.9}{(0.8)(0.00566) - (0.7)(0.00187)} = 11{,}700 \text{ lbs/hr}$$

2. Total heat required:

$$Q = MS\,(t_2 - t_1)$$

s = specific heat of air

t_1, t_2 = temperatures of incoming and outgoing air, respectively, ^{o}F

$$Q = (11{,}700)(0.24)(30)$$

$$= 84{,}300 \text{ Btu/hr}$$

$$Q_t = 84{,}300 + 24{,}000 = 108{,}300 \text{ Btu/hr}$$

3. Net heat required:

$$Q_n = 108{,}300 - 72{,}600 = 25{,}700 \text{ Btu/hr}$$

4. Size of collector:

$$\text{Area} = \frac{(24 \text{ hr})(25{,}700 \text{ Btu/hr})}{(0.85 \text{ Eff.})(380 \text{ Btu/sqft})} = 1908 \text{ sqft}$$

5. Temperature rise of heated air from collector:

The temperature rises were calculated from equation by Buelow and Boyd (13) assuming collector efficiency of 85%. Results are listed in table below.

6. Radiation rates expected in January:

Solar radiation rates were taken from Figure 1 and corrected by a factor 1.9 [Becker and Boyd (8)] upon assumption that collector is at a 30° angle with the horizontal. Rates for three different probabilities are given to show effect on collector and storage sizes.

7. Size of Storage Unit:

It is assumed that the air will be blown through the collector, storage unit (within building) and then .

through the building. Assumption is made that one-third of the heat is utilized during daytime and two-thirds (for 16 hours operation) is stored for night-time use. Sample calculations for storage capacity size are given below making the following assumptions:

Rock density = 170 lbs/cuft

Rock sp.ht. = 0.2 Btu/lb^{o}F

Size 4-in. dia. rock = 33.5 cuin

Wt. 4-in. dia. rock = 3.3 lbs

Rocks per cuft: 38.2

Wt. of rocks $= \frac{(16)(25,700)}{(0.2)(55.1)} =$ 43,900 lbs

Volume Occupied $= \frac{(43,900)}{(3.3)(38.2)} =$ 348 cuft

SUMMARY OF DATA ON COLLECTOR AND STORAGE UNITS

Probability, %	Btu/day Hor. ft^2	Btu/day 30^o ft^2	Collector Size, ft^2	Temp. Rise ^{o}F	Size Storage, ft^3
80	200	380	1908	55.1	348
60	350	665	1090	41.6	462
40	450	855	847	35.5	541

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