THE EFFECT OF VARIOUS PARAMETERS ON THE DESIGN OF SOLAR ENERGY AIR HEATERS

Thesis for the Degree of Ph. D.

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Frederick Henry Buelow

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This is to certify that the

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The Effect of Various Parameters on the Design of Solar Energy Air Heaters.

presented by

Frederick H. Buelow

has been accepted towards fulfillment
of the requirements for

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James S. Boyd
Major professor

Date May 14, 1956

THE EFFECT OF VARIOUS PARAMETERS ON THE DESIGN OF SOLAR ENERGY AIR HEATERS

By

Frederick Henry Buelow

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

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Approved James S. Boyd

ABSTRACT

The purpose of the investigation was to find a design for a solar energy air heater that can be used for drying hay and grain, and for heating buildings on the farm; and to determine the operating characteristics of the unit.

A review of literature showed that solar heaters had been designed for heating water and air for household uses. The air heaters were designed to give a temperature rise of at least 110 degrees F., which is higher than is required for either drying or heating of farm buildings. On the basis of investigations performed by other researchers, a basic design was prepared. The collector consisted of an absorber plate with an air space and insulation below, and an air space and one sheet of glass above the plate. The air to be heated was drawn through the two air spaces and thus passed over both sides of the absorber plate.

An application of the formulas for the convection coefficient and pressure drop to the collector design showed that a shorter air passage is preferable to a longer one, and that the spacing between the absorber plate and the glass or insulation should be reduced until the pressure drop is the maximum that can be tolerated. The air flow rate considered optimum was one giving a Reynolds Number near the critical value between laminar and turbulent flow.

The type of absorber plate judged to be best consisted of sheet metal coated with a mixture of lampblack and asphalt paint on the side exposed to solar radiation. The tests were performed using only asphalt paint.

Ordinary window glass was chosen for the covering because of its transmissivity and resistance to weather. It was found that more than one sheet of glass over the absorber plate would usually not increase the heat gain and temperature rise enough to justify the additional cost. It was also found that the air temperature rise could be increased by reducing the air flow rate per unit area of collector and increasing the total area of the collector to maintain the same total air flow rate.

Equations were developed to describe the operating characteristics of the collector. The validity of the equations was shown by comparing the air temperature rises predicted by the equations with those obtained with the collector which had been constructed and instrumented for testing. The maximum difference between the experimental and calculated values was 15 percent and the average was 7.3 percent. The range of temperature rises used was from 27 to 88 degrees F.

The equations indicate that the efficiency drops rapidly when air temperature rises approach their maximum values and air flow rates become low. At the smaller temperature rises and higher flow rates, the efficiency change is not so pronounced with air flow rate and air temperature rise change. The equations also indicate that the efficiency of the collector and the air temperature rise drop as the entering air temperature increases and the outside air temperature remains the same.

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INTRODUCTION

The scientific interest in the use of solar energy has increased markedly during the past decade. The most important factor contributing to the interest has been that the world's sources of fuels are no longer abundant and that a shortage of inexpensive usable fuels is possible. A report published by Putnam (27) for the Atomic Energy Commission gives the following picture of the world supply of energy. Defining one Q of energy as 10^{18} BTU, we have at present 0.6 Q of usable energy in oil, six Q of usable energy in coal, plus an estimated 100 Q available as atomic energy. The generation of power by atomic means has some limitations, as the disposal of wastes is a problem and the process of conversion is so expensive. Then, practically, we have for use approximately seven Q of fossil fuel for energy. The present rate of expenditure is 0.4 Q per year but by present rate of increase, the rate of use will be one Q per year by the year 2000. This means that the supply of usable fuel has a limited life of from 10 to 12 years plus the time added by atomic energy. Nuclear fuels may replace a considerable portion of the coal, oil, and gas requirements; but no single fuel or power source today can economically furnish all power for all purposes in all parts of the world.

Solar energy is a potential source of power that offers much promise in the near future. Already it is being used on a large scale for drying fruits, heating water, and for attaining high temperatures for research. Work is also being done to perfect a system for converting solar energy into electrical energy directly. By means of photochemical processes, attempts are being made to combine a solar energy collector and an energy storage into a single unit. Some attempts are being made to capture solar energy through photosynthesis, in which the solar energy is used to grow algae, which in turn are dried and used for fuel or food.

Very little research has been reported concerning the possibilities of using a solar energy heating system on the farm, even though it appears to be a logical place to begin since large areas are available for placing collectors and storage units. The energy requirements for the farm as compared with the energy falling on the available area are low. The temperature rises required for drying products and for ventilation of buildings are lower than for house heating. There is a year-round requirement for warmed air because a solar energy system that will heat air efficiently and economically could be used on the farm for drying hay and grain in the summer and fall, and for supplying heat to farm buildings in the winter and spring.

Heating air with solar energy is accomplished by first absorbing the solar energy and then transferring the energy to the air. The absorber may be heated either by focussing the sun's rays on it with a parabolic mirror or by allowing the rays to fall on it directly. The latter system is the more logical for low temperatures because the cost of constructing a large parabolic mirror that will follow the sun throughout the day would be prohibitive for a farmer. In addition, the efficiency of a concentrating collector is lower than the non-concentrating or flat-plate type because of the higher temperatures of the absorber. Transferring the heat to air is also difficult when the absorber is small. Therefore a flat plate type of solar energy air heater would appear to be the most practical for farm use.

The investigation of solar energy air heaters and recommendations for their design would not be complete without determining how the temperature rise of the air passing through the heater and the efficiency of the unit are affected by the rate of air flow, the intensity of the solar radiation, the materials used in construction, and the size and shape of the heater. Once the relationships of these parameters have been defined, the design of solar energy heaters is greatly simplified.

Therefore the problem chosen is to find a design for a solar energy air heater that can be used for drying hay and grain, and for heating buildings on the farm; and to determine the operating characteristics of the design.

The approach to the problem is first to study the designs which have been proposed by other researchers for heating air. The majority of these designs are intended for house heating. On the basis of these designs, keeping in mind that lower temperature rises are needed and that cost must be reasonable, one may arrive at a basic design for a farm solar air heater. The details of the design must then be filled in before constructing a model for testing. The testing of the model constructed can be simplified greatly by analyzing the operating characteristics mathematically if possible and then comparing the experimental and mathematical results.

BACKGROUND INFORMATION

Research in Solar Energy

Nature and Availability of Solar Energy

The energy which the earth receives from the sun consists of radiation at wavelengths between 0.29 and 2.20 microns. The solar spectrum outside the earth's atmosphere has wavelengths between 0.22 and 7.0 microns. However, when this radiation passes through the earth's atmosphere, nearly all of the energy with wavelengths less than 0.29 microns is absorbed by the ozone in the atmosphere. The energy at the longer wavelengths is partially absorbed by the water vapor in the air, and some energy at all wavelengths is absorbed by the air itself and by the dust particles in the air. The spectral distribution of solar energy outside the atmosphere and after passing through an atmospheric mass twice the thickness of the atmosphere perpendicular to the earth's surface is shown in Figure 1. The distribution curve for the spectrum outside the atmosphere was plotted from data given by Johnson (18), and the distribution at the earth's surface is from data by Moon (25). The curve P_{λ} in Figure 1 is the fraction of the total solar energy below wavelength λ, and was also obtained from Johnson (18).

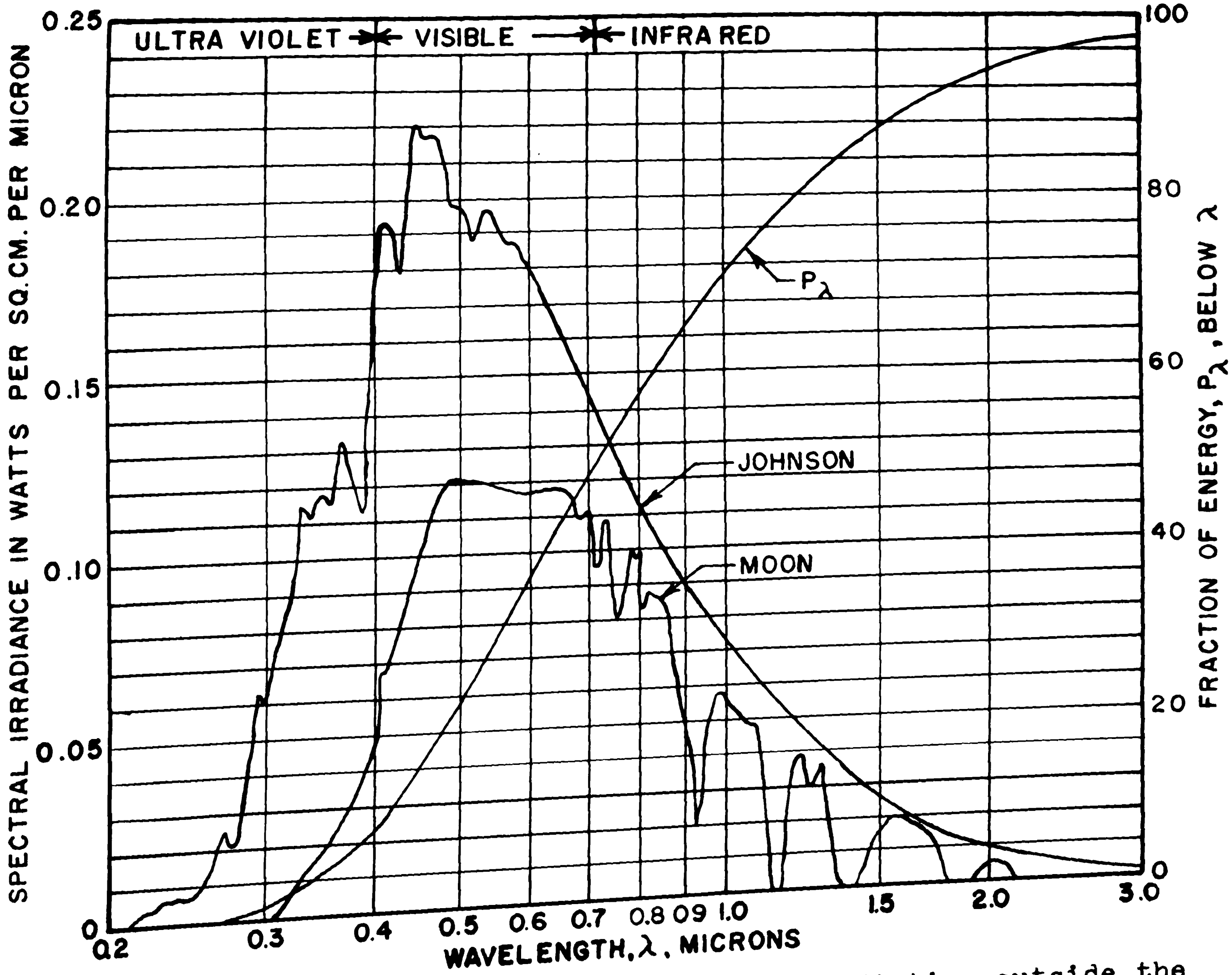

Fig. 1. Spectral irradiance of solar radiation outside the atmosphere and after passing through two atmospheres, and the summation of energy to a certain wavelength.

It is noted from the P_λ curve that the energy in the ultraviolet below the 0.4 micron wavelength is about nine percent of the total energy. The visible spectrum contains about 41 percent of the total energy, and the infrared region beyond wavelengths of 0.72 microns has about 50 percent of the total energy.

As the radiation passes through the atmosphere, the air molecules, the dust, and the water vapor reflect a portion. A spot on the earth's surface therefore receives most of the energy from the direct beam on a sunny day, and the rest from the scattered energy and from the reradiation of the particles which absorbed a portion of the incoming energy. The scattered and reradiated portion of the incoming radiation is known as diffuse radiation.

Johnson (18) has determined that the rate at which solar energy falls on a surface normal to the rays of the sun at the earth's surface without losing any energy to the atmosphere is 2.00 cal. min.$^{-1}$ cm.$^{-2}$, or 442 BTU hr.$^{-1}$ ft.$^{-2}$ at mean distance of the earth from the sun. Owing to the atmospheric absorption and to cloudiness, only about 70 percent of the energy reaches the earth's surface as direct radiation, together with about 15 percent more as diffuse radiation. However, these percentages vary considerably, even on clear days.

Flat-plate Solar Energy Collectors

Apparatus for collecting solar energy in a form that is useful may be classified into the concentrating type and the non-concentrating, or flat-plate type. The concentrating type has reflectors or mirrors to increase the flux density of the incoming energy, and is used where higher temperatures are desired than can be obtained with the flat-plate types. The flat-plate type of collector may be divided into devices for collection by heating, by photoelectricity, by photochemistry, and by photosynthesis. The final breakdown of the heating collectors can be made by dividing the group into liquid heaters and gas heaters.

The first solar heater was patented in 1877 (10). Since that time over 70 patents have been issued for flat-plate collector designs. Nearly all of the collectors shown in the patents were for liquids (10).

Three basic types of flat-plate air heaters are shown in literature: the metal plate, the black gauze, and the glass shingle. Many variations of each of the basic types are possible, but the differences in operating characteristics are minor.

The metal plate type of solar energy air heater consists of a flat metal plate blackened on the side exposed to the sun. The air to be heated passes across the front and/or back surface of the blackened plate. The back side of the plate is usually insulated to minimize heat loss. If air passes behind the plate, the insulation is used to form one side of the air passage. Glass may be used to cover the metal plate. To make the glass covering effective, an air space between the glass and the metal plate is necessary, and this air space may be used as a passage for the air to be heated. The number of glass cover plates is determined by the temperature required and the economy of using additional layers of glass to obtain higher temperatures. Typical of the metal plate design is the collector used in the solar heated house at Dover, Massachusetts (30). The collector was built into the vertical south-facing wall. The collector plate was made of thin gage galvanized iron painted a flat black. The air space between the two sheets of glass was one inch, and between the glass and metal plate it was 3 5/8 inches. The air duct was behind the metal plate and was 3 5/8 inches deep. The back side of the air duct was insulated.

The black gauze type of solar energy air heater consists of several layers of black gauze or screen instead of a metal plate to absorb the incident solar energy (4). The air to be heated is passed through the gauze or screen from the front to the back. The collector may have no glass cover, one cover plate, or several cover plates.

The glass shingle type of solar energy heater consists of overlapped glass plates with air spaces between the plates.

Each plate is coated with a black heat absorbing surface the entire width of the plate and one-third the length of the plate. A number of the plates supported at the edges are installed in the collector frame so that each black surface is below two clear surfaces. A single or double sheet of glass is used to cover the system of plates (22). This collector could be used without cover plates. The air to be heated passes between the overlapped glass plates. As it does, it first comes in contact with a relatively cool glass surface, then a warmer section, and finally the black surface of one of the glass plates.

The solar water heaters in general consist of copper tubing, with or without fins, laid on a bed of insulation and covered by one or more glass cover plates. The pipes and fins are painted black for maximum absorption of energy (5).

Solar Collector Design Formulas

Formulas for the performance characteristics of solar energy collectors which help to determine the most economical design for a particular application are appearing in literature.

An equation was proposed by Hottel and Woertz (16) for water heating collectors in which the temperature rise of the water as it passes through the collector is not more than 15 degrees F. The collector consisted of copper water tubes soldered to a blackened copper absorber plate. Five and one-half inches of insulation was placed below the absorber plate so that the heat loss through the bottom was rendered negligible. Air-spaced layers of glass were placed above the absorber plate. The equation is as follows:

$$\frac{q_L}{A} = \frac{T - T_a}{\dfrac{n}{c\left(\dfrac{T - T_a}{n + f}\right)^{1/4}} + \dfrac{1}{h_w}} + \frac{\sigma\,(T^4 - T_a{}^4)}{\dfrac{1}{\epsilon_c} + \dfrac{2n + f - 1}{\epsilon_G} - n}$$

where q_L is the rate of total heat loss

A is the exposed area of collector

T is the absolute temperature of the absorber plate

T_a is the absolute temperature of the outer air

n is the number of glass plates

c is the convection coefficient for transfer between parallel tilted plates. Suggested values are

Angle of tilt from horizontal	c
0°	0.19
30°	0.17
60°	0.15
90°	0.13

f is the ratio of thermal resistance of the outer glass plate to that of an average inner plate

h_w is the coefficient of convection due to wind. Suggested values for f and h_w are

Wind velocity mph.	h_w	f
0	1	0.76
10	4.07	0.36
20	7	0.24

σ is the Stefan-Boltzmann constant, 0.1723×10^{-8} BTU ft.$^{-2}$ hr.$^{-1}$ R^4 .

ϵ_c is emissivity of absorbing surface for low-temperature radiation.

ϵ_G is emissivity of glass surface for low-temperature radiation.

It is noted that the above equation is based on a uniform temperature T, but may be applied to collector temperature varying continuously from water inlet to outlet if T is replaced by the arithmetic-mean water temperature.

Hottel and Whillier (15) have proposed equations which are intended for both water and air. The equations are

$$\frac{q_u}{A} = F_R \left[H\ R\ (\tau\alpha) - U_L\ (t_1 - t_o) \right]$$

where

$$F_R = F' \left[1 - e^{-\frac{1}{F'}} \ \frac{G\ C_p}{U_L} \right]$$

In these equations

q_u is the rate of useful energy collection

A is the collector area

F_R is the heat removal efficiency

H is the total insolation rate on a horizontal surface

R is the orientation factor to convert horizontal incidence to incidence on the tilted collector surface

τ is the effective transmittance of the cover glass

α is the absorptivity of the absorber plate

U_L is the heat loss coefficient of the collector

t_1 is the entering fluid temperature

t_o is the outdoor ambient air temperature

F' is a factor determined by collector design

G is the mass flow rate of transport fluid per unit collector area

C_p is the specific heat of the transport fluid.

For the case where an air stream bathes the entire rear surface of the blackened absorber plate, the following values are suggested:

No. of glass cover plates	Typical value of U_L	h for laminar flow	F'	h for highly turbulent flow	F'
1	1.2	2	0.63	6	0.83
2	0.7	2	0.74	6	0.90
3	0.5	2	0.80	6	0.92

Masson (23) has developed an equation for water heaters in which the water passes through the collector between metal plates or in pipes. His equation is as follows:

$$\Delta t = (t_m - t_1) \quad (1 - e^{-\frac{\lambda S}{D}})$$

where

Δt is the increase in temperature of the water as it passes through the apparatus

t_m is the temperature of a radiation sensing element

t_1 is the initial temperature of the water

λ is the overall coefficient of heat transfer

S is the area of the collector surface

D is the mass flow rate of the water.

It is noted that the temperature t_m is the maximum temperature which the absorbing surface will reach when the flow rate of the water is zero.

Historical Uses of Solar Energy

The magazine, Chemical and Engineering News, in a staff report (3), gives several uses and some history of the use of solar energy. Part of the report reads as follows:

> One of the oldest applications of solar energy is in the production of high temperatures. This use dates back even before the time of Archimedes, who was himself an early solar energy pioneer. In 1774, Antoine Lavosier used solar energy to melt iron. Several years ago, William M. Conn, at Rockhurst College in Kansas City, Mo., erected a 10-foot parabolic aluminum mirror that was able to produce temperatures at 3000° C. in as little as 10 seconds. Conn's solar furnace, which created a 0.3-inch spot of heat, was used primarily in the study of metals and refractories - their melting points, high-temperature modifications, and potential uses as structural components in jet engines. Last year Felix Trombe, at work in the French Pyrenees, constructed a 40-foot solar furnace, the largest in the world.
>
> The Russians have been active in the field of solar power. In 1941, F. F. Molero of the Power Institute of the USSR Academy of Sciences constructed an experimental solar energy plant at Stalingrad. To focus the sun's rays, Molero used 33-foot parabolic mirrors made of ordinary window glass. In 1946, in another Russian project, a solar converter, also employing parabolic mirrors, began supplying power to a cannery at Tashkent in Central Asia, where a solar research institute is now located.
>
> During the past two years, the National Physical Laboratory in Pusa, India, has been investigating the use of solar energy for the production of steam for driving low-horsepower engines that might

be used to operate water pumps and small looms. They have also been studying use of solar energy for household cooking.

Heywood (14) has noted several interesting and historical uses for solar energy. His statements are as follows:

> The period of 1870 to 1915 was one of great activity in the development of such systems [mirrors], and almost every possible arrangement of mirrors was devised and tested.
>
> Carl Gunter (Austria, 1854 to 1873 approx.) experimented with solar boilers and mirror systems of the tilting type....but gave no records of large-scale results.
>
> A. Mouchot (France, 1860 to 1880) constructed conical mirrors....to generate steam....which operated engines and water pumps.
>
> On September 29, 1878, steam was used to operate an ammonia-absorption refrigerator which had been invented by Carre' about 1860, and this must have been the first occasion on which ice was produced by direct solar radiation.

A solar energy power plant was used at Cairo, Egypt, for pumping irrigation water from the Nile but was apparently abandoned during the World War of 1914-1918 (1).

Storage of Energy for Later Use

The effective use of solar energy on the farm depends not only on the economical design for collectors, but also on facilities for storage of surplus energy collected while the sun is shining, for use when there is no incoming solar energy. The storage of the energy when using flat-plate solar energy collectors presents a problem because of the small temperature working range.

Two types of thermal energy storage have been proposed in literature for use with flat-plate solar energy collectors.

One is the use of the sensible heat characteristics of a material, and the other is the use of the latent heat of fusion of certain materials. The sensible heat characteristics of small rocks and of water seemed the most promising in economic terms (2). Small rocks were employed where air was the transport medium between the collector and storage, and water was used when water was the transport medium.

The experimental work for a system using an air heater and rocks for storage was done by Löf and Hawley (21). The air was heated by the collector and was then passed through the bin of rocks. When energy was required during the night, cool air was passed through the warm rock bed and then blown to the point where it was needed. The rock bed has the characteristic that a high degree of temperature stratification is possible and therefore the entire bed need not be warm before air can be reheated to a high temperature. The technical advantages of a rock bed storage are given by Löf (20) as follows:

(1) Very large heat-transfer area in the pebble bed, requiring only small temperature driving forces for substantially complete transfer of heat;
(2) the combination of a heat exchanger and a heat storage system in one unit;
(3) a counter-current heat-exchange system wherein advantageous temperature stratification is realized;
(4) comparatively low energy requirements for circulating air through the heat-storage bed.

A water storage system has the advantage that less space is required for storage of a given quantity of energy than with a rock bed. However, the cost will probably be higher because of the container required for the water. If the end use of the solar energy is to furnish the energy for a hot water supply, the water storage will be the more economical. Another disadvantage of water storage is that there is little, if any, temperature stratification in the water and therefore the temperature of the medium used to carry the energy out of the water storage will vary more than with rock or latent heat types of storage.

The latent heat characteristics of substances that have melting points at or near the temperatures at which the energy is to be stored may be used for storage of thermal energy. Telkes (29) has used chemicals for storing energy for house heating and has listed the merits of heat of fusion storage as follows:

(1) Heat storage as the heat of fusion, which is around 10,000 BTU per cu. ft. at the melting point, generally within a rather narrow temperature range.
(2) Higher solar heat collection efficiency due to the relatively low, nearly constant, storage temperature.
(3) Additional heat storage as specific heat, which is about the same as that of water on an equal volume basis.

The chemicals tested thus far have had shortcomings because of undercooling; temperature stratification or segregation and therefore poorer heat transfer characteristics in the container; corrosion of the container; volume changes during phase changes; and cost of the chemicals and the containers.

A different method of using the heat of fusion principle to store energy is the method proposed by Yanagimachi (31) in which the heat of fusion of water is used in combination with a heat pump. The heat pump evaporator is in a tank of water and as it removes energy from the water, ice is formed. The ice is changed back to water by running the liquid water through the solar collector. The advantages of the system are that the collector will operate at a high efficiency because of the low temperature; and it is not necessary to use chemicals which have the undesirable properties mentioned previously.

Collector Design Considerations

Several problems in the design of a solar energy air heater for experimental work and for actual large scale operation may be resolved from information given in literature. The problems of air passage shape and size, and the characteristics of the air movement have been studied. The characteristics of surfaces for absorbing solar energy have also been studied so that a good energy absorbing surface can be selected. The covering for the collector can be selected from information given on glass and other transparent materials. The instrumentation required to obtain data from an experimental collector is available from commercial sources and does not require additional research to select the proper instruments to obtain the desired data.

Air Passage and Movement

The notation that is used in this section is as follows:

C is specific heat at constant pressure; for air it is equal to about 0.24 BTU per lb. °F.

D is diameter in ft.; for a rectangular duct, four times the hydraulic radius divided by the wetted perimeter may be substituted.

F is friction loss in ft. lb. force per lb. mass and is equal to $\Delta p/\rho$.

G is mass velocity in lb. mass per (sq. ft. sec.).

L is length of duct in ft.

Re is Reynolds Number.

V is average velocity in ft. per sec.

a is width of the rectangular duct in ft.

b is depth of rectangular duct in ft.

f F is the Fanning friction factor.

g is 32.1740 lb. mass ft. per (lb. force sec.^{2}).

h is the coefficient of heat transfer by convection in BTU per (hr. ft. °F.).

k is the coefficient of heat transfer by conduction in BTU per (hr. ft. °F).

w is mass rate of flow through the duct in lb. per sec.

p is pressure drop in lb. force per sq. ft.

μ is dynamic viscosity; for air at 100° F. it is equal to 0.0468 lb. mass per (hr. ft.) or 1.3×10^{-5} lb. mass per (sec. ft.).

μ_s is the dynamic viscosity at surface temperature.

ρ is density; for air at room conditions it is approximately 0.075 lb. mass per cu. ft.

The shape of a collector and the air passages in it will influence the efficiency with which the air passing through it is heated. The shape of the air passages will also determine the pressure drops that will occur as the air passes through the collector. The velocity of the air and the width of the duct will influence the convection coefficients for heat transfer from the absorber plate to the air and from the air to the transparent covering of the collector. The way in which these characteristics vary with different rates of air flow and different spacings between the covering and the absorber plate may be shown by using equations given in literature to give a graphical picture of the variations.

In order to show the variations in pressure drop and convection coefficient with collector shape and air velocity, several assumptions will be made. The cross-sectional shape of the individual air passage is rectangular. The absorber plate will form one side of the duct and the transparent covering the other. The width of the duct is assumed to be 1.5 feet because a glass covering requires support at spacing no greater to withstand moderate stresses without breakage. Since the roughness and entrance conditions of the duct may vary with different collector designs, it will be assumed that there is laminar flow when Reynolds Number is less than 2100 and that there is turbulent flow when Reynolds Number is greater than 3000.

The maximum pressure drop that can be tolerated in the collector is determined by (a) the pressure which the fan moving the air can produce and (b) by the pressure drop necessary to move the air through the hay or grain to be dried, or through the building to be heated. The fans usually used for grain drying and hay drying have a maximum static pressure head of about four inches of water. Most of this available pressure is required to move the air through the hay or grain and through the ducts. Therefore a collector can only use a small portion of the total pressure drop of the system. It will be assumed that about five percent, or 0.2 inch of water pressure drop can be tolerated in the collector. If entrance and exit pressure drops are subtracted, the pressure drop in the heating duct will be less and is assumed to be about 0.1 inch of water.

The coefficient of heat transfer by convection, h, should be a maximum because the higher it is, the lower will be the temperature of the absorbing plate. A low absorbing plate temperature is desirable because radiant heat transfer from the absorbing plate to the covering glass will be less and therefore the overall heat loss from the collector will be less.

The relationship of Reynolds Number, depth of the air space, and rate of air flow may be determined from the definition of Reynolds Number, $Re = DG/\mu$. Four times the hydraulic radius may be substituted for the diameter D in the equation. The hydraulic radius for a cross-section of width a and depth b is $ab/2(a + b)$. Total flow rate may be substituted for the mass velocity G . The relationship is $G = w/ab$. The equation which results is

$$Re = \frac{2\,w}{\mu\,(a + b)}\,.$$

Then if μ is 1.3×10^{-5} lb. mass per (ft. sec.), a is 1.5 feet, and Reynolds Number is either 2100 or 3000, the equations which result are

$$b = 73.26\,w - 1.5 \qquad \text{for } Re = 2100$$

$$b = 51.28\,w - 1.5 \qquad \text{for } Re = 3000$$

where b is in ft. and w is in lb. mass per sec. of air. The curves of the two equations are shown in Figure 2.

The relationship of pressure drop in the turbulent region, depth of the air space in the collector, and rate of air flow may be determined from the Fanning equation for steady flow (8)

$$F = \frac{4\,f\,L\,G^2}{2\,g\,D\,\rho^2}\,.$$

The desired form of the equation can be obtained by substituting $\Delta p/\rho$ for F, $2ab/(a+b)$ for D, and w/ab for G. The Fanning friction factor f is equal to $0.046(Re)^{-0.2}$ (24). The pressure drop per foot of duct length is desired. Therefore L is one foot. The constants $a = 1.5$ ft., $\mu = 1.3 \times 10^{-5}$ lb. mass per (ft. sec.), $g = 32.17$ lb. mass

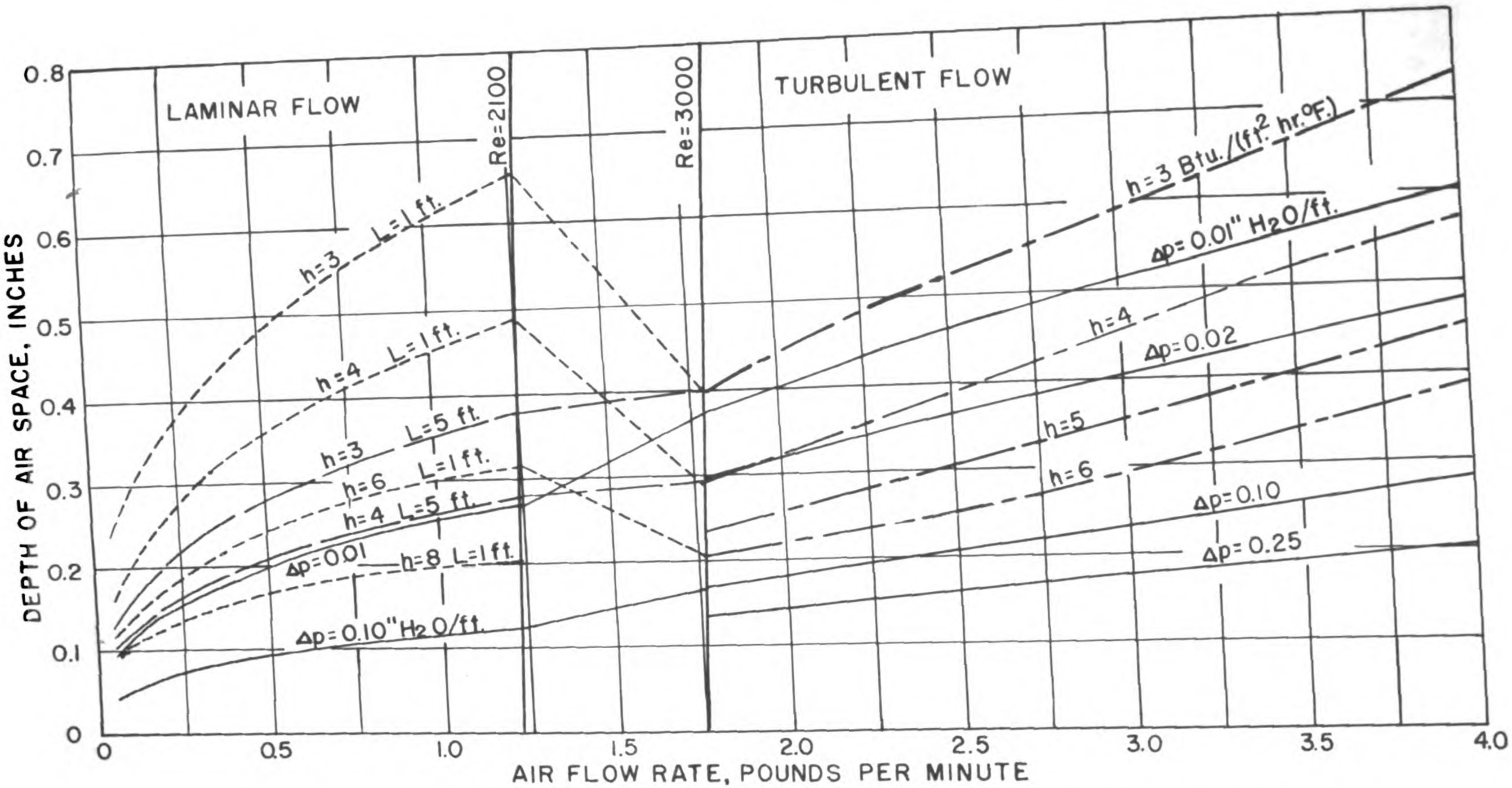

Fig. 2. Calculated convection coefficients and pressure drops for various air flow rates through a duct 1.5 feet wide and of various depths.

ft. per (lb. force sec.^{2}), and $\rho = 0.075$ lb. mass per cu. ft. are also substituted into the equation to give

$$w = \frac{66.93\ (b)^{1.667} (\Delta p)^{0.556}}{(1.5 + b)^{0.667}}\ .$$

Plots with pressure drops equal to 0.01, 0.02, 0.10 and 0.25 inch of water per foot of length are shown in Figure 2.

The relationship of pressure drop in the laminar region, depth of the air space, and rate of air flow may be determined from the equation given in the Chemical Engineers' Handbook (9) for a streamline flow between broad parallel plates with a spacing of 2B.

$$w = \frac{2\ a\ B^3\ \rho\ g\ p}{3\ \mu\ L}$$

Substituting $B = b/2$, $\rho = 0.075$ lb. mass per cu. ft., $a = 1.5$ ft., $g = 32.17$ lb. mass ft. per (lb. force sec.^{2}), $\mu = 1.3 \times 10^{-5}$ lb. mass per (ft. sec.) and $L = 1$ ft. gives

$$w = 185{,}596\ b^3\ \Delta p$$

Plots with pressure drop equal to 0.01 and 0.10 inch of water per foot of collecter length are shown in Figure 2.

The relationship of coefficient of convection in the turbulent region, depth of air space in the collector, and rate of air flow may be determined from the following equation given by Colburn (7) where the coefficient is based on the difference in temperature between a surface and the mean cup temperature.

$$\frac{h}{CG}\left(\frac{C\mu}{k}\right)^{2/3} = 0.023\ (Re)^{-0.2}.$$

In this equation G is in lb. per (hr. sq. ft.). The equation may be simplified for air passing through the collector by substituting the known values and identities into the equation. Thus $C = 0.24$ BTU per (lb. °F.), $\rho = 0.075$ lb. per cu. ft., $\mu = 0.0468$ lb. mass per (ft. hr.), $k = 0.0154$ BTU per (hr. ft. °F.), $D = 2(1.5b)/(1.5 + b)$, and $G = 3600\ w/15b$. The equation which results is

$$w = 0.6013\ h^{1.25}\left(\frac{b^5}{1.5 + b}\right)^{\frac{1}{4}}$$

The curves for constant h values are shown in Figure 2.

The relationship of coefficient of convection in the laminar region, depth of air space in the collector, and rate of air flow may be determined from the equation given by Sieder and Tate (28).

$$\frac{hD}{k} = 2.0\left(\frac{wC}{kL}\right)^{1/3}\left(\frac{\mu}{\mu_s}\right)^{0.14},$$

in which the convection coefficient h is based on the difference between surface temperature and the mean cup temperature. The equation may be simplified for air passing through the collector by substituting the known values into the equation. Thus, $C = 0.24$ BTU per (lb. °F.), $\mu = 0.0468$ lb. mass per (ft. hr.), $k = 0.0154$ BTU per (hr. ft. °F.), and $D = 2(1.5b)/(1.5 + b)$. The value of μ_s is estimated to be 0.0532 lb. mass per (ft. hr.), so that the last term of the equation is 0.98. Two values for L selected are 1 ft. and 5 ft. The equations which result are

$$h = 0.3851 \left(\frac{1.5 + b}{b}\right) w^{1/3} \qquad \text{for } L = 1 \text{ ft.}$$

$$h = 0.2252 \left(\frac{1.5 + b}{b}\right) w^{1/3} \qquad \text{for } L = 5 \text{ ft.}$$

Curves of constant h values are shown in Figure 2.

The combination of all the curves mentioned in this section in Figure 2 gives considerable information concerning the design characteristics of a solar energy air heater. It is apparent from the curves that the length of the duct or the distance the air moves between the plates should be as short as practical design will permit, because the average coefficient of convection is higher for a short section with a shallow air space than for a longer one with a wider air space and the same pressure drop. The depth of the air space should be as narrow as the maximum pressure drop will permit because the narrower spacing has a higher convection coefficient for the same air flow rate.

The gradients of the pressure drop curves do not change sharply between the laminar and turbulent regions. Nor are the convection coefficient gradients greatly affected unless the duct length is very short. If the duct length is very short the optimum air flow rate is one with a Reynolds Number in the laminar region but near the critical value for turbulence. For longer ducts the air flow rates are not so critical.

The rates of change of the pressure drop and convection coefficient are approximately equal when air flow rate changes.

Absorber Plate

The purpose of the absorber plate is to absorb the incoming solar energy and transfer the heat to the air passing over the surface of the plate. The temperature at any one point of the absorber plate is constant under operating conditions and therefore the specific heat of the plate is of no concern in the selection of a plate. The desired absorbing properties of the surface exposed to the sun can most easily be obtained by coating the surface of the plate with a good solar energy absorber. Therefore the plate can be thin and of any material that has the required mechanical strength and will hold the coating on its surface. Sheet metal meets all of these requirements, is easy to handle, is not expensive, and therefore seems to be the best material.

The coating for the absorber plate must have a high absorptivity in the solar energy spectrum. Some materials with this property are

Material	Absorptivity	Reference
Asphalt pavement, dust free	0.93	12
Black asphalt saturated paper	0.93	13
Slate, dark gray	0.89	12
Green paint	0.93	13
Lampblack	0.98-0.97	11
Galvanized iron, very dirty	0.91	11

None of the above materials are satisfactory coatings in themselves but a combination of materials is possible. Carnes (6) tested coatings for use with solar water heaters and concluded that a paint made of lampblack, asphalt paint, turpentine, and gasoline gave the best absorbing characteristics. The basic ingredient was asphalt paint. Enough turpentine and lampblack were added to prevent gloss. The gasoline served as a thinner.

The quantity of heat transferred from the absorbing plate to the air passing over it is equal to the energy absorbed minus that lost by radiation. Therefore a fixed quantity is transferred at a given set of conditions, equal to the product of the area of the absorber plate in contact with the air, the coefficient of convection, and the temperature difference between the air and the absorber plate. A low absorber-plate temperature is desirable because the radiation losses will then be less. Therefore a large convection coefficient is desirable. Methods of increasing the convection coefficient were discussed in the previous section on air passage and movement. It is possible to increase the area and thereby decrease the plate temperature by allowing the air to move along both sides of the absorber plate instead of along the surface exposed to the sun only. Thus the area is twice as large. It is possible to use an absorber plate with finned surfaces to increase the area further but it may not be economical to do so.

Glass Covering

The functions of a covering for a collector are (1) to form one side of the duct for the air that is passing over the absorber plate; (2) to allow solar energy to fall on the absorber plate with little energy loss, and at the same time prevent the long wavelength energy from the absorber plate from escaping from the collector; (3) to form a protective covering over the absorber plate; and (4) to keep the heat loss from the air in the collector to the atmosphere to a minimum.

The first function can be satisfied by any material. However, the second function requires that the material used for the covering have a high transmissivity in the major portion of the solar energy spectrum but be opaque to long wavelength energy coming from radiation sources at several hundred degrees F. or less. This requirement is met by ordinary plate glass and window glass. Two typical spectral transmittance curves as determined by Parmelee et al. (26) are shown in Figure 3. A comparison of these curves with the energy spectrum of solar radiation at the earth's surface indicates that none of the infrared and ultraviolet energy is lost beyond the limits of the curve. Fishenden and Saunders (12) have found that glass will absorb 90 to 95 percent of the long wavelength energy falling on it. The remainder is reflected. However, there is considerable variation in transmittance values with glass from different

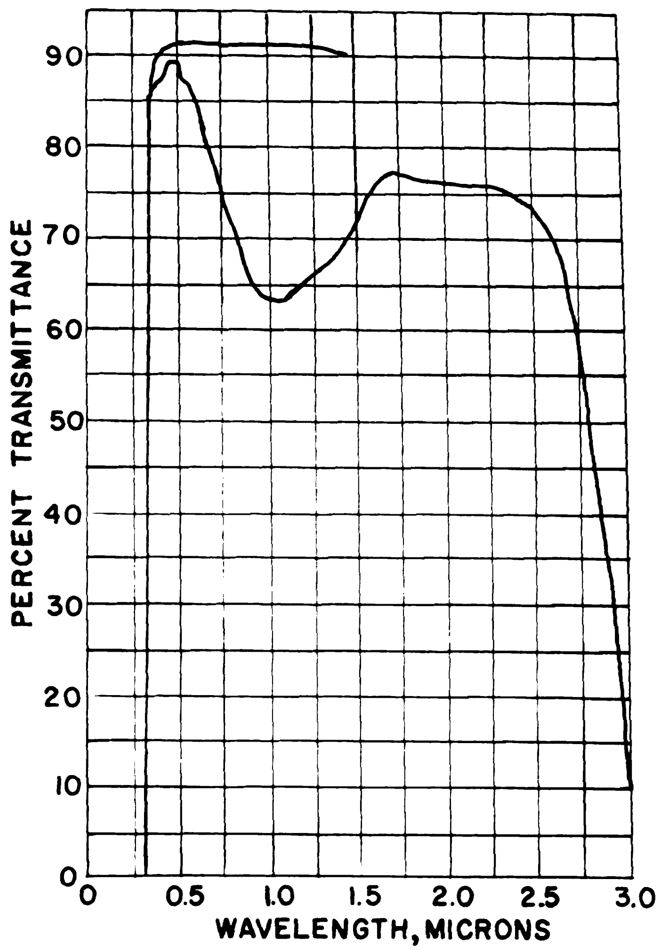

Fig. 3. Experimental spectral transmittance curves of two different polished plate glasses 1/4 inch thick, as determined by Parmelee _et al._ (26).

sources, due probably to the differences in iron content which imparts a greenish tinge and results in energy absorption in the near infrared portions of the spectrum. Hottel and Woertz (16) have determined the normal transmittances of several glasses and plastics from different sources. Portions of their results are given below. The transmittance is for four parallel plates unless otherwise indicated. The thickness given is for a single plate.

Cellulose-acetate from various sources:	
(0.010 in. thick)	61.4%
(0.103 in. thick)	41.4%
(0.010 in. thick)	67.4%
(0.060 in. thick)	53.1%
(0.006 in. thick)	64.1%
Cellulose nitrate (0.0275 in. thick)	45.0%
Acrylic type resins:	
(0.122 in. thick)	53.2%
(0.265 in. thick)	57.5%
Glasses:	
Greenish window glass (0.085 in. thick)	41.9%
Belgian plate (0.239 in. thick)	52.4%
S. S. window	55.2%
Grade A window (0.126 in. thick)	63.9%
Grade A window (0.089 in. thick)	65.8%
Grade A window (0.089 in. thick)	61.3%
Water-white plate (0.242 in. thick)	67.8%
Water-white plate (0.247 in. thick)	68.0%
Water-white plate (0.117 in. thick)	68.8%
Water-white plate (0.117 in. thick)	69.9%
Water-white plate (0.117 in. thick), three layers	75.5%
Pyrex (three layers only)	77.5%

A comparison of the glasses and the plastics shows that glass generally is more transparent than plastic. In addition, glass is more resistant to damage by weather, other than hail. It is also noted that window glass is a good transmitter, being exceeded only by water-white plate glass and by Pyrex glass. Therefore at present the most satisfactory covering is ordinary window glass.

The transmittance of glass also varies with the angle of incidence of the solar radiation. The relationship was determined by Parmelee et al. (26) for two different glasses and is shown in Figure 4. The curves show that the transmittance is decreased very little until the angle of incidence of the incoming energy is greater than 30 degrees.

The radiation coming from the absorber plate is absorbed or reflected by the glass covering. This energy is dissipated by convection and radiation to the outside. It is possible to decrease this loss by adding one or more sheets of glass above the first one with an air space between each pair. The more sheets added, the less will be the heat loss. But at the same time more of the incoming solar energy will be absorbed or reflected by the additional glass plates. Thus a point is reached at which the addition of another sheet of glass will actually decrease the efficiency of the collector. The number of glass cover plates is determined by the operating temperature of the absorber plate, and also by considering that the addition of another sheet of glass to the collector will not furnish enough additional energy to be economical.

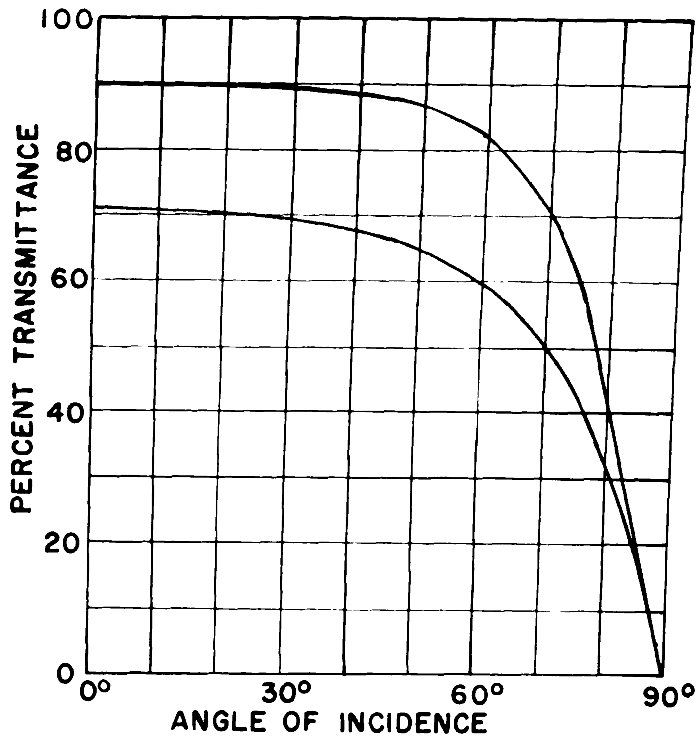

Fig. 4. Experimental transmittances at different angles of incidence for two different polished plate glasses 1/4 inch thick using direct solar radiation, as determined by Parmelee _et_ _al_. (26).

The transmissivity of glass is based on direct solar radiation. Diffuse solar radiation has about the same characteristics except that it hits the glass cover plate at all angles of incidence and therefore more of it is reflected than direct solar energy at a normal angle of incidence.

EXPERIMENTAL INVESTIGATION

Construction of Experimental Air Heater

The construction of an experimental air heater was based on a design possibly suited for agricultural purposes such as drying hay or grain, and heating buildings. The equipment therefore must be low in cost, and give a temperature rise of about 30 to 60 degrees F. to the air passing through it. The pressure drop of the air passing through the unit must be small. It is assumed that a collector on a farm will be installed on the roof of a building and thus have an orientation which will allow the incoming radiation to have a normal angle of incidence at solar noon during the time of the year when the demand for energy is the greatest.

Paint Investigation

In order to determine which paint would be the most satisfactory for use on the surface of the absorber plate, a set of four boxes, each five inches square and one inch deep, was constructed of 1/4 inch plywood and mounted on a board. The paints to be tested were used to coat a set of four five-inch square plates of 22-gage sheet steel. After the paint had dried, a thermocouple was soldered to the center of the back side of each of the plates. By means of spacer blocks, the plates were placed about 1/2 inch from the bottom of the box. Additional spacer blocks held a sheet of single strength window glass in place at the top of the box. The thermocouple leads were connected to a Leeds and Northrup Portable Precision potentiometer (Serial No. 300083) through a rotary switch and an ice bath. The apparatus is shown in Figure 5.

The paints tested were as follows:

Paint no. 1 - Great Lakes Dead Flat Black

Paint no. 2 - Sherwin-Williams Enameloid, 730 Flat Black

Paint no. 3 - Rutland Black Asphalt Paint (has glossy finish)

Paint no. 4 - Pittsburgh Black Waterspar Enamel (has glossy finish).

The test units were placed in the sun and moved at short intervals of time so as to keep the plate surface normal to the incoming solar radiation. The first time the plates were exposed, paints numbers 1 and 2 lost some of their oils by evaporation when the plate temperatures increased. These oils condensed on the glass covers and as a result more of the solar energy was absorbed and reflected by the glass and oil with a resulting drop in temperature. To remedy this trouble, the two plates were placed into an oven at 320 degrees F. for 40 minutes to remove the volatile oils.

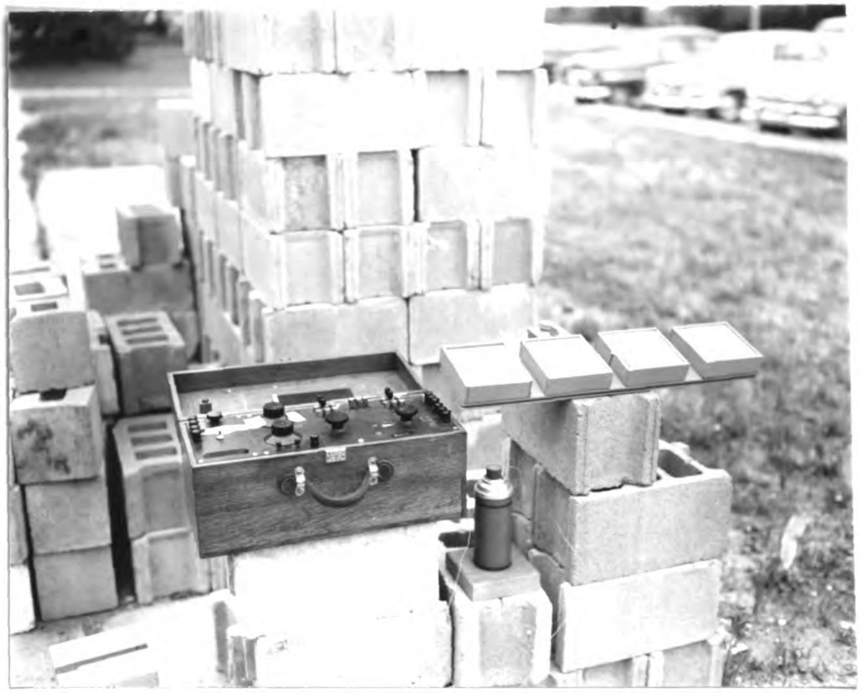

Fig. 5. Paint testing units with potentiometer, ice bath and rotary switch.

Readings of the plate temperatures were taken when the rate of incoming solar energy was constant and the plate temperatures had reached constant values. The results are shown in Table I.

TABLE I

PLATE TEMPERATURES IN DEGREES F.

	Trial 1	Trial 2	Trial 3	Trial 4	Mean
Paint no. 1	167	172	172	170	170.25
Paint no. 2	173	178	178	176	176.25
Paint no. 3	175	178	181	178	178.00
Paint no. 4	166	171	173	170	170.00

An analysis of variance of the temperatures given in Table I shows that the differences in mean temperatures are very highly significant and that paint no. 3, the black asphalt paint, is significantly better than the other three paints tested. Therefore the black asphalt paint was selected for use on the absorber plate of the collector.

Collector Design

The purpose of building a collector and testing it was to determine the operating characteristics as affected by the design, incoming solar energy, and rate of air flow through the collector. Therefore it was felt that the unit should be built so that it could be rotated and also tilted to keep it at a certain orientation with respect to the sun. Thereby more constant operating conditions for test purposes could be achieved. To make the collector rotatable, it was constructed on a two-wheeled trailer. The tilting was obtained by mounting the collector on a frame so it could be tilted about a pipe running along the underside of the collector. A collector unit that would fit on the trailer and also be large enough to simulate a farm size collector was a unit three feet wide and five feet long. At each end of the collector, air chambers were constructed to give a uniform distribution of air velocity across the entire width of the absorber plate. The collector unit is 9 1/2 inches deep overall. The bottom is covered with 1/4-inch plywood. In order to keep the heat loss through the bottom of the collector at a minimum, three inches of redwood fiber insulation were placed over the bottom surface of the collector and covered with another sheet of 1/4-inch plywood. The bottom portion of the air chambers was not insulated.

In order to keep pressure drop through the collector low, the air space between the top of the insulated surface and the absorber plate, and between the absorber plate and the glass covering were made 9/16-inch deep. The spacing was obtained by using three rows of spacers, one along each side and one down the center. Thus the air ducts through the collector were each about 1.5 feet wide and 9/16-inch deep. The drawing of the unit is shown in Figure 6. It is possible to change the depth of the air spaces by using different thicknesses of spacer blocks. The effective area of the absorbing plate for the unit as constructed was 13.9 square feet. The length of the air passage was five feet. The air movement through the collector was brought about by connecting the intake opening of a fan (ILG No. 182-007 DA Style F2 with a 1/3 HP motor) to the outlet end of the collector with a flexible three-inch hose. The inlet for the collector was a three-inch aluminum pipe five feet long. The air flow rate was controlled with an adjustable flap on the outlet opening of the fan. The entire apparatus is shown in Figure 7.

The absorber plate was a 3-ft. by 5-ft. sheet of 22-gage sheet steel, painted on the top side with three coats of black asphalt paint.

Instrumentation

The rate of air movement through the collector was determined by using a three-inch vane anemometer (Keuffel and Esser, calibrated between 0 and 1200 ft. per min.) placed over the end of the inlet pipe, and a stop watch. This method of measuring air flow was chosen rather than an orifice or a pitot tube because of the wide range of air

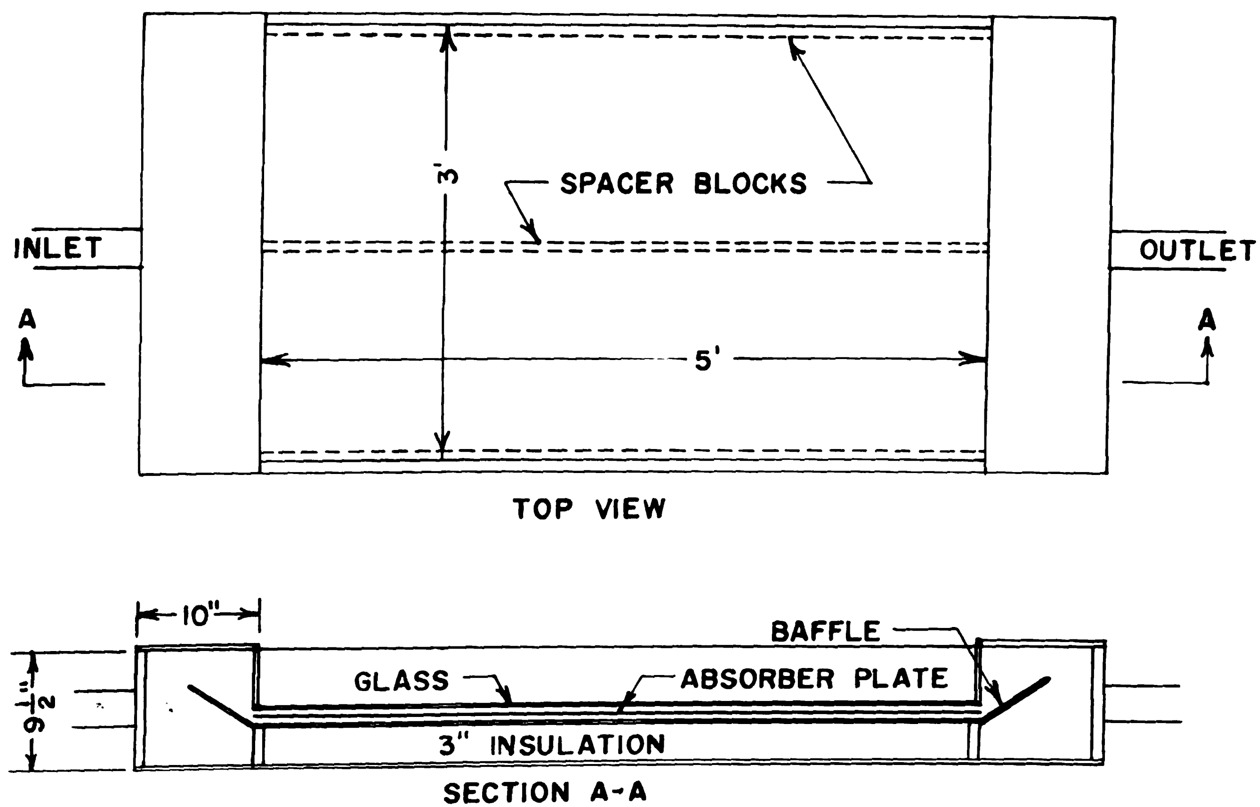

Fig. 6. Experimental solar energy air heater.

Fig. 7. Experimental solar energy air heater mounted on trailer.

flow rates which had to be measured. Both the orifice and the pitot tube measure fluid flow by indicating a differential pressure. The differential pressure varies as the square of the flow rate and therefore a manometer with a range great enough for the high flow rates could not be read accurately at the low flow rates. Also, an orifice would require more pressure drop at the higher flow rates than the fan could supply.

For energy calculations the mass flow rate of the air is required. Since the anemometer measures air velocity which can be converted to volume flow rate, it was necessary to know the barometric pressure, the temperature of the air at the anemometer, and the amount of moisture in the air. A mercury-in-glass barometer was used for pressure, a mercury-in-glass thermometer placed inside the inlet pipe was used for measurement of incoming air temperature, and a Friez Model HA-2 aspiration psychrometer was used to obtain the wet and dry bulb temperatures of the air.

The incoming solar energy was measured with an Eppley pyrheliometer (no. 1551) connected to an Eppley potentiometer (no. 328). The pyrheliometer contained a ten-junction thermopile. The surface of the exposed thermocouples was coated with lampblack. In the range at which the potentiometer was operating it could be read to the nearest 50 millivolts. With the possibility of another 50 millivolts adjustment error, the readings were accurate within 100 millivolts. The calibration of the instrument was 1 BTU/(sq. ft. hr.) = 8.52 millivolts. Therefore the 100 millivolts is equal to 11.7 BTU/(sq. ft. hr.). If the incoming energy were 300 BTU/(sq. ft. hr.), then the accuracy of the instrument readings would be within about four percent.

Figure 8 shows the vane anemometer, the stop watch, the aspiration psychrometer, the pyrheliometer, and the potentiometer for the pyrheliometer.

The temperature measurement was made with copper-constantan thermocouples connected to a Leeds and Northrup Portable Precision potentiometer (serial no. 300083) through a rotary switch and an ice bath.

Five thermocouples spaced 15 inches apart were soldered to the under side of the absorber plate near the center spacer block to obtain readings of the temperatures at these points on the absorber plate. Three thermocouples, one in the inlet chamber and two in the outlet chamber, measured the air temperatures at those points so that the temperature rise of the air could be calculated. All thermocouples used were made of 38-calibration copper and constantan wires. The millivolt readings were converted to temperatures by charts published by Leeds and Northrup Company (19). Before installation each of the thermocouples was submerged in a water bath of known temperature and the indicated temperature

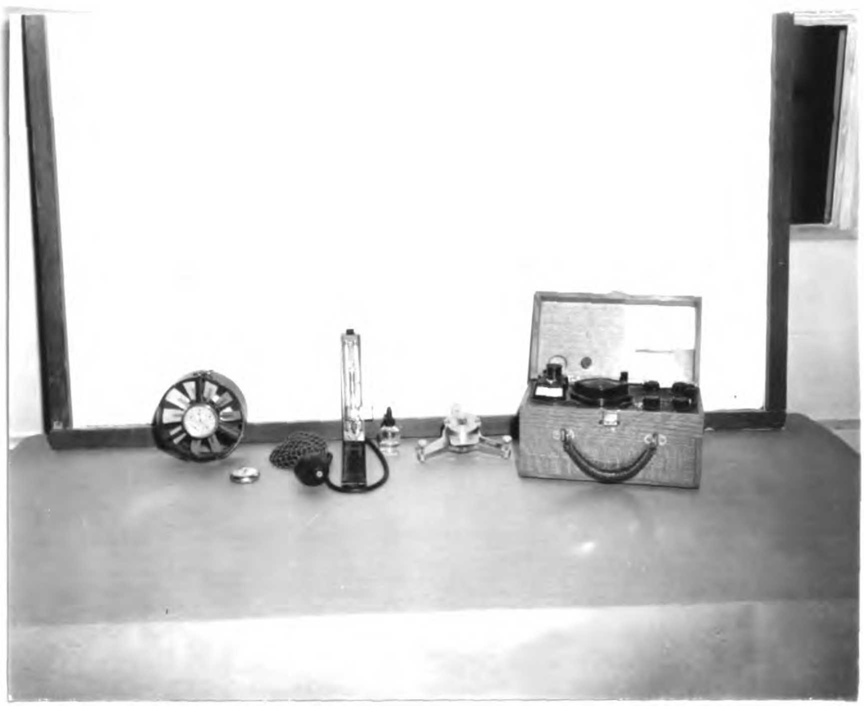

Fig. 8. Vane anemometer, stop watch, aspiration psychrometer, pyrheliometer, and potentiometer for the pyrheliometer used in tests.

checked. All thermocouples installed gave readings within one degree F. of the true temperature.

Experimental Procedure

The test runs with the solar energy air heater were made on days when less than about two percent of the sky was clouded, because it was necessary to have steady incoming radiation to obtain valid test data. The solar energy falling on the collector was also kept constant during the day by rotating and tilting the collector about every 15 minutes to keep it normal to the sun. Thus on a clear day the incoming energy was constant from about 10 a.m. to 3 p.m.

Before any test data was taken the collector was allowed to warm up for at least half an hour. No data was taken until all temperatures remained constant for at least ten minutes. While the temperatures were approaching constant values, the wet and dry bulb temperatures, the pyrheliometer reading, and the air flow data were recorded. The barometric pressure was read once each testing day at noon.

After all temperatures had remained constant for ten minutes, the millivolt readings were recorded. Upon completion of the test, the air flow rate through the unit was changed and the procedure repeated.

Analysis of Data

Pyrheliometer Readings

All pyrheliometer readings were in millivolts. The conversion into solar energy intensity was by the calibrated relationship, 8.52 mv. = 1 BTU/(sq. ft. hr.). The values which result are the rate at which solar energy falls on one square foot of surface normal to the direct radiation.

Air Flow Data

The calibration of the vane anemometer was 0.0436 x anemometer reading in ft. per min. = air flow rate in cfm. The volume flow rate was converted into mass flow rate by the use of a nomograph which gave the specific volume and enthalpy of air at temperatures between 50 and 250 degrees F. and between 0 and 0.2 lb. water vapor per lb. dry air. The humidity of the air was determined from a psychrometric chart, using the wet and dry bulb data.

The nomograph for specific volume and enthalpy of air is shown in Appendix II. It was constructed from data and charts given in Zimmerman and Lavine (32). The nomograph is for air at a barometric pressure of 29.92 in. Hg. When the pressure was different, the specific volumes were corrected by the perfect gas law, $p_1v_1 = p_2v_2$. No pressure corrections were made for enthalpy because the errors introduced in the enthalpy values by a pressure different from standard nearly cancel when change in enthalpy is determined.

Collector Efficiency

The efficiency of the collector is defined as 100 times the fraction of the solar energy impinging on the collector that is carried off by the air passing through the collector. The energy gain of the air is the mass flow rate times the change in enthalpy. If the incoming solar energy rate is R and the area of the collector is A, then the collector efficiency E is

$$E = \frac{w \Delta h}{AR}$$

Presentation of Data

The results of tests of the collector using 9/16-inch deep air passages are given in Table II. A single sheet of glass was used for a cover plate. The glass was Pennvernon double strength window glass, grade B. The complete test data is found in Appendix I.

The two series of tests were with identical collector arrangement but were run on different days.

The absorber plate temperatures are also given in Appendix I. The curves of typical temperature distributions on the absorber plate are shown in Figure 9. It is noted

TABLE II

RESULTS OF TESTS OF SOLAR ENERGY AIR HEATER WITH SINGLE COVER GLASS AND 9/16" AIR PASSAGE DEPTH

Test No.	Air Flow Rate cfm.	Air Flow Rate lb/min.	Temp. Rise, F.	Energy Gain BTU/hr.	Efficiency %
A-1	23.1	1.59	75	1774	41.8
A-2	23.1	1.58	80	1896	43.9
A-3	48.8	3.33	55	2737	61.0
A-4	73.0	4.99	38	2844	65.8
A-5	86.5	5.89	36	3216	74.4
A-6	94.6	6.70	34	3296	70.8
A-7	120.9	8.55	28	3540	76.0
B-1	22.3	1.67	88	2184	44.9
B-2	39.0	2.90	61	2627	54.0
B-3	53.3	3.96	48	2851	58.6
B-4	85.8	6.41	41	3923	80.6
B-5	108.8	8.14	34	4151	87.8
B-6	112.7	8.42	30	3789	80.2
B-7	130.6	9.75	27	3861	87.6

15 sq. ft

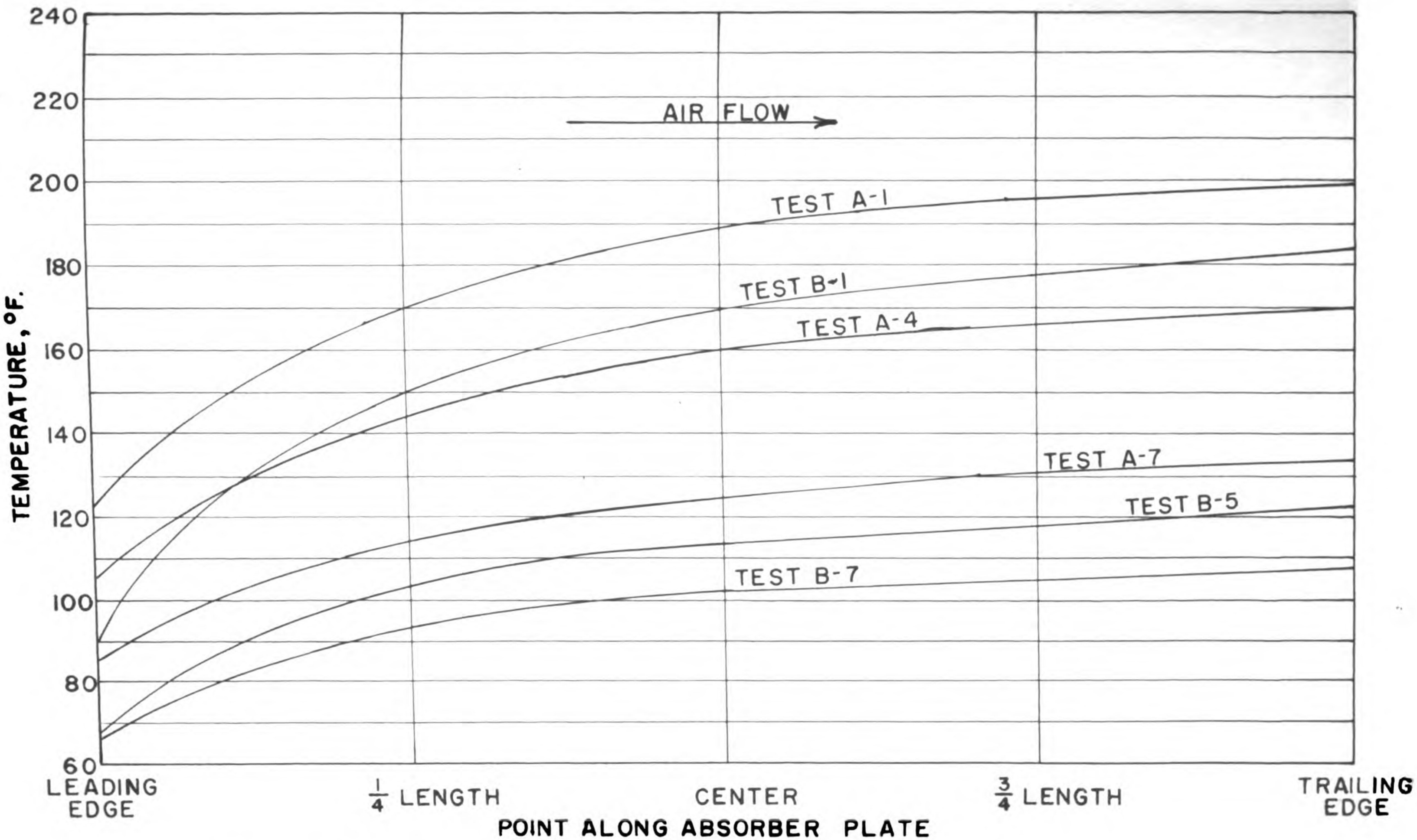

Fig. 9. Typical temperature distribution curves for absorber plate with air passing over surfaces.

that the gradients of the curves are greater near the leading edge, indicating that the convection coefficient there is higher than at other points on the absorber plate.

The complete analysis of the data is given in the following sections of this thesis.

MATHEMATICAL ANALYSIS OF COLLECTOR OPERATION

Derivation of Equations for Collector Operation

The amount of testing required to define completely the operating characteristics of solar energy air heaters would be very extensive. Therefore an attempt has been made to analyze the operating characteristics mathematically. Several assumptions were necessary to make a general analysis possible. Since all the factors of a given collector design cannot be defined exactly, the approximate results given by the analysis would be valuable for determining the tendencies of the operating characteristics.

Factors Affecting Operation

The factors which determine the efficiency with which the air passing through the collector is heated and the temperature rises which result are all associated with the heat losses of the collector. The first energy which is lost is that caused by reflection of the incoming radiation by the glass and the absorber plate. The absorber plate reflection will not be stopped by the cover glass because the wavelength of the radiation is not changed by reflection. For this reason it is necessary to have an absorber plate that has a high absorptivity of the solar radiation spectrum.

The incoming radiation is therefore reduced by a fraction equal to the combined reflectivity of the glass and the absorber plate.

Another energy loss is that lost by radiation from the absorber plate to the cover glass and from the cover glass to the sky. An estimate of the net radiation passing from the absorber plate to the glass may be obtained by using values obtained in tests with the collector. The data in Appendix I show that the absorber plate temperature is at a temperature of about 150 degrees F. when the air is passing through the collector at a moderate velocity. The minimum temperature of the glass is equal to the outside air temperature. On the day on which the tests were performed the air temperature was about 95 degrees F. Therefore an absorber plate temperature of 150 degrees F. and a glass temperature of 120 degrees F. may be used to give an approximate value for transfer of heat by radiation from the absorber plate to the glass. The equation for heat exchange by radiation between large parallel planes of different emissivities (17) may be used.

$$\frac{q}{A} = \frac{\sigma}{\frac{1}{e_1} + \frac{1}{e_2} - 1} (T_1^4 - T_2^4)$$

Substituting 0.174×10^{-8} BTU per (hr. ft.^{2} °F.^{4}) for σ, 0.92 for e_1 (12), 0.92 for e_2, 610 for T_1, and 580 for T_2 gives a net radiation of 37.3 BTU per (ft.^{2} hr.). Since the temperatures of the absorber plate and the glass are always in the same range on the absolute temperature scale, it is possible to assume a linear relationship between the temperature difference and the net radiation without a large error, as shown in Table III. The temperatures given in the table are typical of those encountered in the collector.

The loss of heat through the insulation on the bottom of the collector is very small as compared with the other heat losses and therefore no serious error will result if it is neglected. If less insulation were used, it would be an important factor.

The energy loss by convection occurs on the outside surface of the glass. The heat lost at the glass surface includes both convection and radiation losses, and as shown above, the radiation loss may be considered linear with temperature, just as the convection loss, for approximate calculations.

Derivation of Expressions

In order to derive mathematical expressions for the collector characteristics, the following notation is used:

- A is the area of the collector in sq. ft.
- R is the rate at which the solar energy falls on the surface of the collector in BTU per (ft.^{2} hr.).
- U is the overall coefficient of heat transfer between the air in the collector and the outside air in BTU per (ft.^{2} hr. °F.).

TABLE III

COMPARISON OF RADIANT HEAT TRANSFER VALUES CALCULATED FROM THE LINEAR EQUATION WITH CORRECT VALUES

$$q_1/A = 1.19 \Delta t + 8.7$$

t_1, °F.	t_2, °F	Δt, °F.	q/A	q_1/A	% Error
180	100	80	103	103.9	0.87
180	110	70	92	92.0	0
180	120	60	81	80.1	0.12
180	130	50	69	68.2	1.16
180	140	40	57	56.3	1.23
180	150	30	43	44.4	3.26

t_1 is absorber plate temperature, °F.

t_2 is glass temperature, °F.

Δt is temperature difference, °F.

q/A is correct value of radiant heat transfer, BTU/(sq. ft. hr.).

q_1/A is value of radiant heat transfer calculated from linear relationship, BTU/(sq. ft. hr.).

m is the mass flow rate of air through the collector in lb. per hr.

E is the fraction of incoming radiation absorbed by the absorber plate.

C is the specific heat at constant pressure of the air passing through the collector in BTU per (lb. °F.).

t is the mixing cup temperature of the air at any point in the collector a distance A from the leading edge, in °F.

t_o is the outside air temperature in °F.

t_e is the temperature of the air entering the collector in °F.

An energy balance may be written for the collector shown in Figure 10 by assuming that steady state conditions exist, that the radiant energy falling on the surface of the collector is constant with time and at all points, and that there is negligible heat conduction in the absorber plate. The area A is equal to the collector length when the width is unity. Therefore the length may be expressed as A.

The energy absorbed by the absorber plate for a collector of length or area dA is

$$\text{Energy in} = E\,R\,dA.$$

Assuming that the energy lost at any point in the collector is proportional to the temperature difference between the air inside and the air outside the collector, one can express the energy lost as

$$\text{Energy lost} = U\,(t - t_o)\,dA.$$

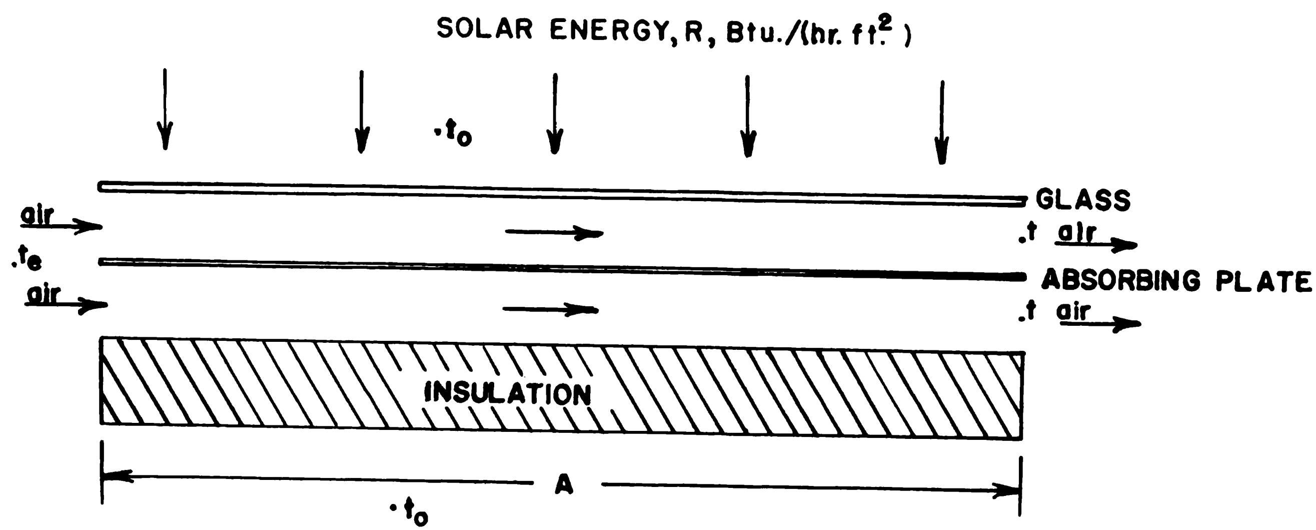

Fig. 10. Collector model used for mathematical analysis.

The energy carried out of area dA by the air passing through is

$$\text{Energy gain by air} = m\ C\ dt.$$

The energy balance, in which the energy absorbed is equated to the energy lost and that picked up by the air is

$$E\ R\ dA = U\ (t - t_o)\ dA + m\ C\ dt.$$

Separating the variables gives

$$dA = \frac{m\ C\ dt}{E\ R - U\ (t - t_o)}\ .$$

Integrating gives

$$A = \frac{m\ C}{-U}\ \ln\ \left[E\ R - U\ (t - t_o)\right] + c\ ,$$

where c is the constant of integration.

Let $t = t_e$ when $A = 0$; then

$$c = \frac{m\ C}{U}\ \ln\ \left[E\ R - U\ (t_e - t_o)\right]$$

and

$$A = \frac{m\ C}{U}\ \ln\ \frac{E\ R - U\ (t_e - t_o)}{E\ R - U\ (t - t_o)}$$

Letting $N = \frac{U\ A}{m\ C}$ and solving for temperature rise $(t - t_o)$ gives

$$t - t_o = \frac{E\ R}{U}\ (1 - e^{-N}) + (t_e - t_o)\ e^{-N}\ .$$

If the entering temperature is the same as the outside air temperature, the equation simplifies to the form

$$t - t_o = \frac{E\ R}{U}\ (1 - e^{-N})\ .$$

The equation for the maximum temperature rise is obtained when the outgoing air temperature is equal to the entering temperature and is

$$t - t_o = \frac{E\,R}{U} .$$

The efficiency of the collector is equal to the energy gain of the air divided by the incoming solar energy, or

$$\text{Efficiency} = \frac{m\,C\,(t - t_o)}{A\,R} .$$

When the entering air temperature is equal to the outside air temperature, the efficiency may be expressed by the equation

$$\text{Efficiency} = \frac{E}{N}\,(1 - e^{-N}) .$$

The above equations were derived for a collector with unit width. If the collector is of a different width, the area and the mass flow rate will be changed by the same amount. Since one is divided by the other in the equations, no other values in the equations will be changed.

Comparison of Equations and Test Results

Application of Equations to Collector Tested

The values for temperature rise, incoming energy, mass flow rate, area, and specific heat of the air may be obtained directly from the test data for use in the equations. The fraction of energy absorbed by the absorber plate is approximately equal to the transmissivity of the glass used times the absorptivity of the absorber plate. The value given for the transmissivity of the glass used is about 0.91 (26), and the absorptivity of the asphalt paint is about 0.93 (12, 13). Therefore the value of E for the equation is about 0.85.

The only unknown in the equation is the overall coefficient of heat transfer U. The value of U is determined by the coefficients of convection on the inside and the outside of the glass, and by the emissivities of the glass and the absorber plate. Their relative importance for the determination of the U value is not known. Therefore the logical procedure is to determine the U value for each of the tests and from them determine an average U value. Then, by using this average U value, the experimental results may be compared with the results predicted by the equation. The equation is of a form that requires the U value be determined by trial and error methods or by a graphical method. A rough estimate may be obtained from Figure 11 in which curves of constant U values are plotted on a graph with coordinates $(t - t_o)/ER$ and A/mC. Table IV shows the values of each of the parameters in the equation for each of the tests and the resulting U values. The mean U value from the tests is 2.35, if Tests B-5 and B-7 are omitted. These tests were not considered valid because the experimental efficiency of the unit appeared to be over 100 percent, and therefore the U value according to the equation was zero or less, which is not possible.

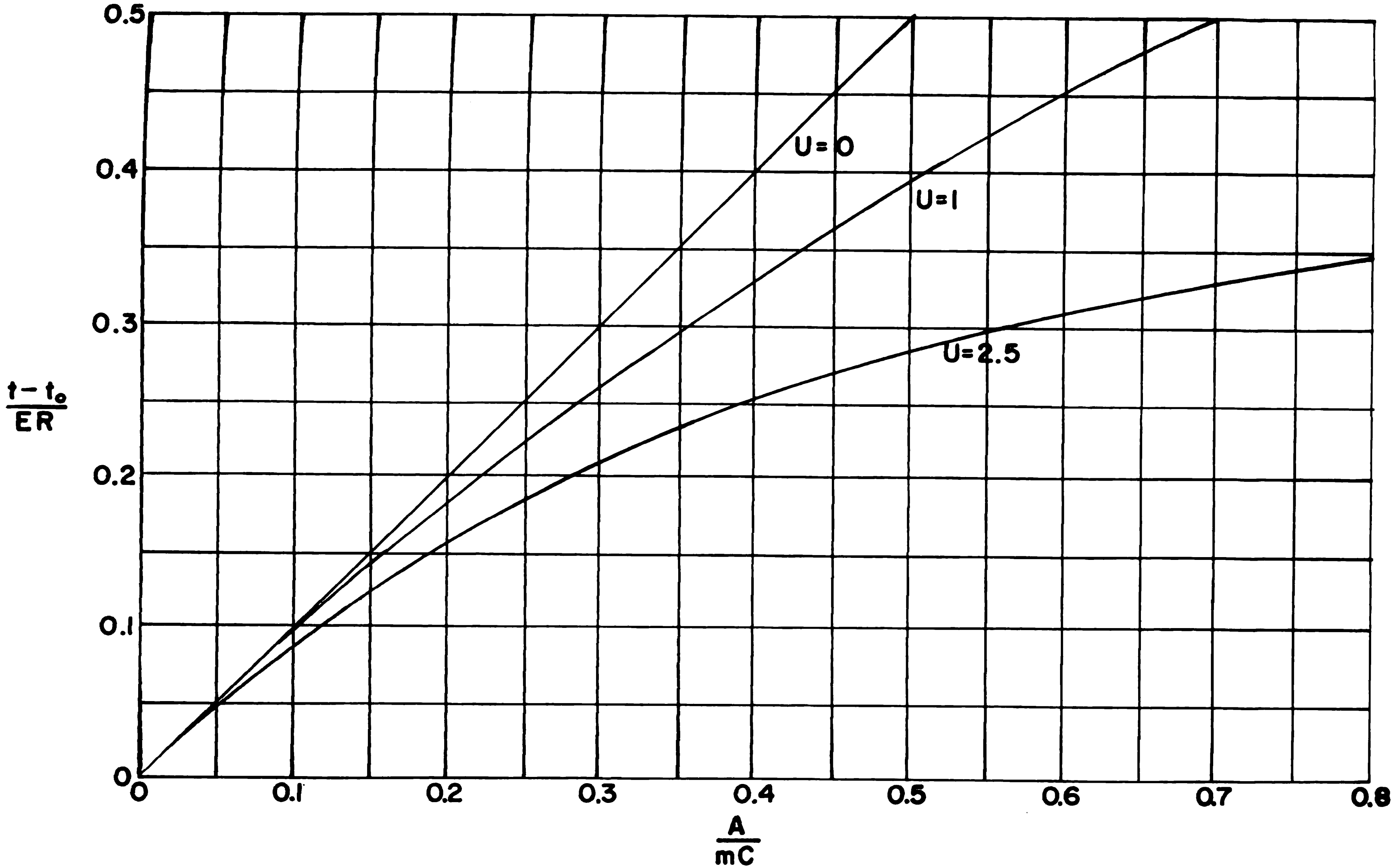

Fig. 11. Graphical representation of the equation $(t - t_o)/ER = (1 - e^{-N})/U$ where $N = UA/mC$.

TABLE IV

PARAMETERS FOR EQUATION

$$t - t_o = \frac{E\,R}{U}(1 - e^{-N}) \text{ where } N = \frac{UA}{mC}$$

$E = 0.85$, $A = 13.9$ ft.^{2}

Test No.	$t-t_o$	R	m	C	U
A-1	75	305	95.4	0.248	2.78
A-2	80	311	94.8	0.250	2.58
A-3	55	323	199.8	0.249	2.53
A-4	38	311	299.4	0.250	2.88
A-5	36	311	353.4	0.253	1.74
A-6	34	335	402.0	0.241	2.66
A-7	28	335	513.0	0.246	2.18
B-1	88	350	100.2	0.248	2.58
B-2	61	350	174.0	0.248	3.06
B-3	48	350	237.6	0.250	3.40
B-4	41	350	384.6	0.249	0.80
B-5	34	340	486.4	0.250	0.00
B-6	30	340	505.2	0.250	1.0
B-7	27	317	585.0	0.244	0.00

When the U value of 2.35 is substituted into the equation, the temperature rises may be calculated and compared with the experimental results as shown in Table V. As shown by the table, the greatest difference between calculated and measured values was 7.5 degrees F. Percentagewise, the greatest difference was 15 percent.

Validity of the Mathematical Analysis

The comparison of experimental and test results shows that the equations derived for predicting collector operation correspond to test results fairly closely for the range of operating conditions tested. The differences between the two results are due to several things. One has been discussed previously, namely that the error in measurement with the pyrheliometer could easily be four percent, which would change the calculated temperature rise by the same percentage. Other possible sources of error are in the measurement of air flow rate with the vane anemometer and in the determination of moisture content of the air with the aspiration psychrometer, both of which influence the calculated mass flow rate of the air. The fraction of incoming radiation absorbed by the absorber plate was determined from information published by other researchers and may not be exact for the collector tested. The assumption that the net rate of radiant energy flow from the absorber plate to the glass and from the glass to the sky is linear with temperature difference is not

TABLE V

COMPARISON OF CALCULATED AND MEASURED VALUES OF TEMPERATURE RISE USING A U VALUE OF 2.35

Test No.	$t-t_o$ Calculated	$t-t_o$ Measured	Difference °F.	% Difference
A-1	82.5	75	7.5	10.0
A-2	84.2	80	4.2	5.3
A-3	56.3	55	1.3	2.4
A-4	39.7	38	1.7	4.5
A-5	34.6	36	-1.4	3.9
A-6	34.6	34	0.6	1.8
A-7	27.6	28	-0.4	1.4
B-1	92.6	88	4.6	5.2
B-2	67.2	61	6.2	10.2
B-3	53.6	48	5.6	11.7
B-4	36.6	41	-4.4	10.7
B-5	28.9	34	-5.1	15.0
B-6	28.0	30	-2.0	6.7
B-7	23.4	27	-3.6	13.3
			Total	102.1
			Average	7.3%

exact and therefore another source of error. As shown in the previous section, the coefficient of convection varies with air flow rate and therefore the U value is not exactly constant for a given collector.

Another possible source of error is introduced by changes in air movement across the outside surface of the glass which changes the convection coefficient of the glass and as a result changes the U value in the equation.

When the equation is used for collector design, a factor of safety in the required temperature rise will assure an adequate design. A factor of safety of at least 1.1 is recommended on the basis of the test results given previously. This factor of safety does not account for any design assumptions made and may have to be increased if all of the parameters are not known exactly. A larger factor of safety may also be necessary if the collector design is different from the one tested.

Comparison with Equations Proposed by Other Researchers

Two equations have been proposed by other researchers that can be compared with the equations proposed in this thesis. Both the equations proposed by Hottel and Whillier (15) as given on page 12 , and the one by Masson (23) as given on page 13 , agree with the equations proposed in that the temperature change or energy gain of the fluid passing through the collector is proportional to a factor of the form

$(1 - e^{-F})$ where F is a function of the flow rate, specific heat and an overall coefficient of heat transfer. All of the equations also agree that the energy collected and temperature rise of the fluid increase linearly with rate of incoming energy. Neither of the equations referenced were derived for the specific type of collector used in the tests described in this thesis.

Equation Characteristics and Relationships of Factors

Since the equations developed predict the operation of the collector closely and are similar to the equations developed by other researchers, they may be used to determine the effects of the parameters on the operating characteristics of a collector. It is assumed that the entering air temperature is the same as the outside air temperature in order to simplify the discussion.

Figure 12 shows the relationship of the rate of air flow through the collector and the temperature rise in the air when the U value of the collector is 2.5, 1.0 or zero. For this plot it was assumed that the rate of incoming radiation was 350 BTU/(hr. ft.^{2}), the absorptivity was 0.9, the specific heat of the air was 0.245 BTU/(lb. °F.), and the area of the collector was one square foot. It may be noted that if only a small temperature rise is required, the U value of the design is not a large factor in determining

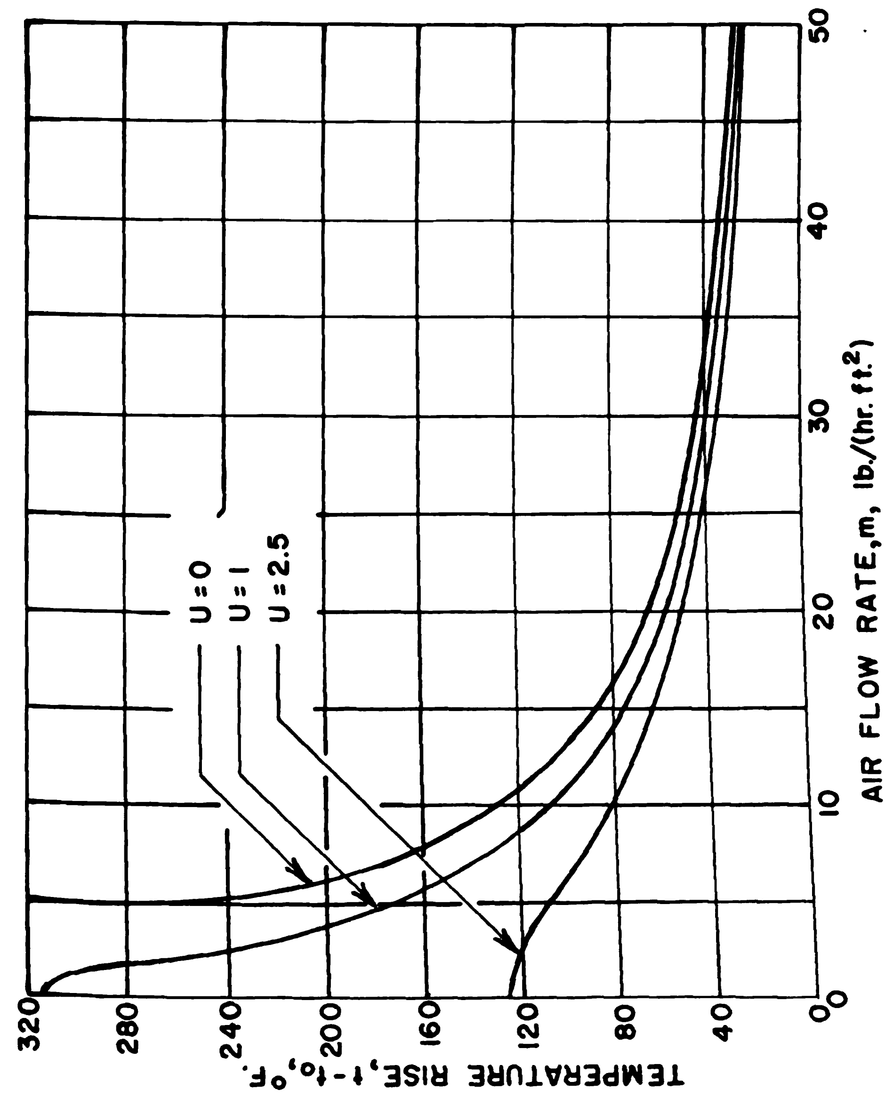

Fig. 12. Relationship of temperature rise and air flow rate with three different overall coefficients of heat transfer.

the temperature rise of the air because the curves converge at the high air flow rates. Therefore in many designs it may be more economical to use one sheet of glass rather than two. The temperature may be increased by decreasing the air flow rate per unit area, which would mean a slight increase in area to maintain a given total air flow rate.

The efficiencies and air heat gains for the three U values plotted against the air flow rate are shown in Figure 13. The same constant parameters were used for these curves as for those in Figure 12. The curves show a rapid increase in efficiency and energy gain with increasing air flow rate at the low air flow rates and less rapid increases at the high rates. The maximum efficiency that can be obtained is the absorptivity of the collector, which was assumed to be 90 percent for these curves.

The relationship between the temperature rise and the heat gain and efficiency is shown by Figure 14. This set of curves also illustrates that the efficiency drops rapidly with an increase of temperature rise when temperature rise is great and that the U value of the collector is most important with large temperature rises. Therefore with large temperature increases the large loss in efficiency may be avoided by adding more insulation through the use of more layers of glass.

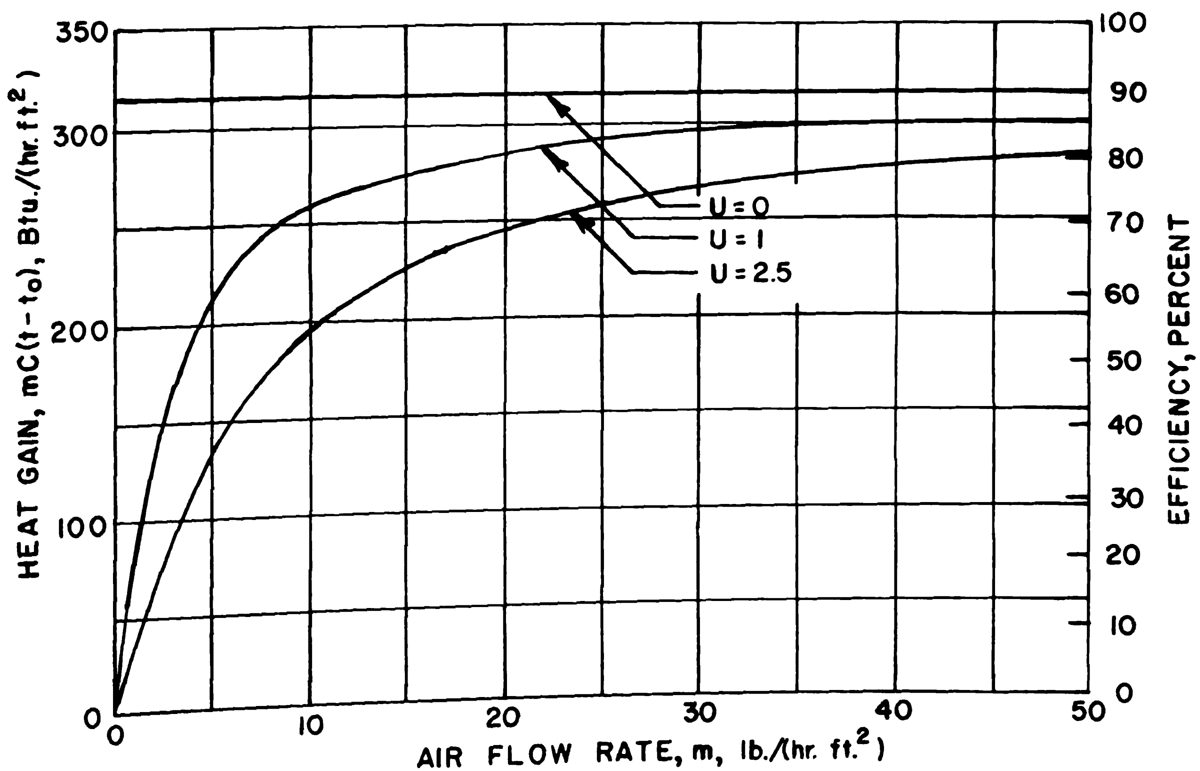

Fig. 13. Relationship of efficiency and heat gain with air flow rate for three different overall coefficients of heat transfer.

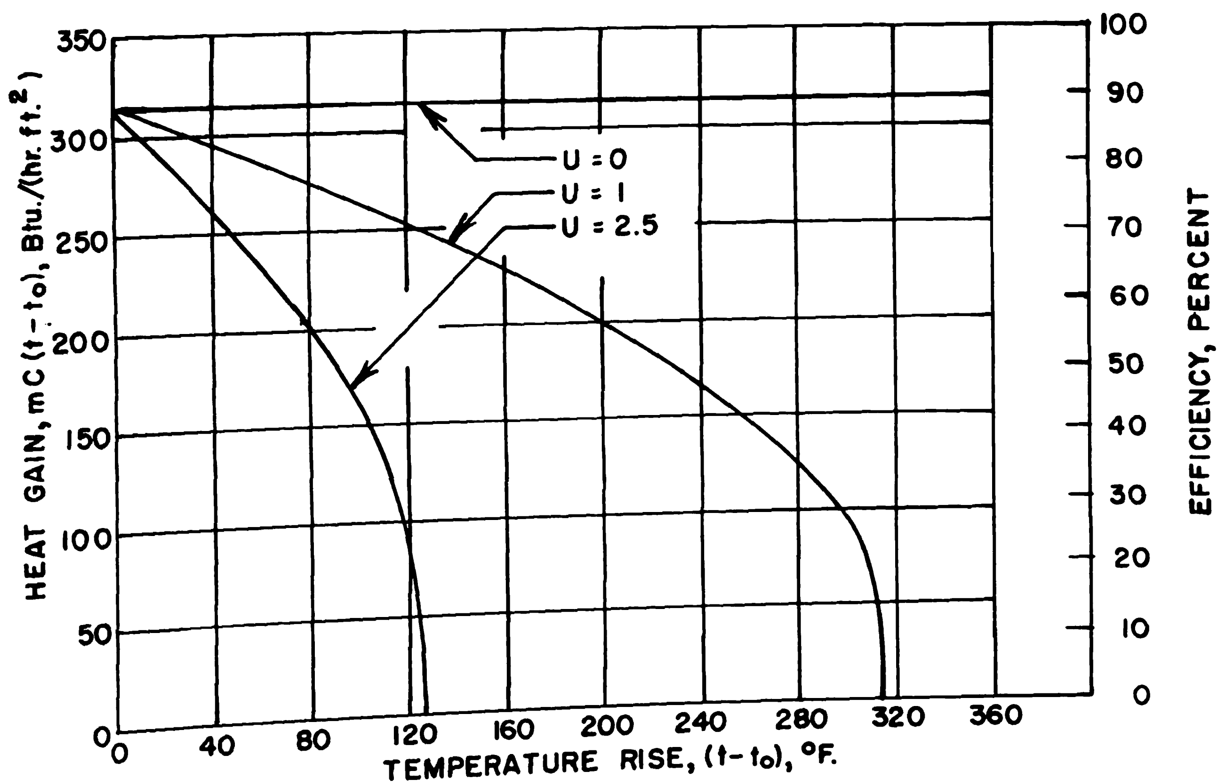

Fig. 14. Relationship of efficiency and heat gain with temperature rise for three different overall coefficients of heat transfer.

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The equations proposed seem to indicate that the temperature rise of air passing through the collector is a function of area and so does not depend on the length and width of the air passages. However it has been noted that the convection coefficients do depend on the length of the air passages and therefore a different U value will result if the length of the ducts are changed. The U value will be less for a shorter duct. The design requirements of a collector will dictate the practical minimum length for a duct.

The portion of incoming solar energy absorbed by the absorber plate does not remain constant as more layers of glass are added, but decreases with each layer. Therefore the U value of the collector will not only be decreased by the addition of more glass, but the absorptivity will also be decreased somewhat and will partially counteract the advantages of a lower U value. The number of glass cover plates will have to be determined for each design on the basis of both the additional energy gained and the cost of the additional glass as compared to a larger collector with less glass.

The general equation developed shows that temperature rise of the air passing through the collector is decreased when the entering air temperature increases and the outside air temperature remains the same. Therefore the efficiency is highest when the entering air temperature is the same or lower than the outside air temperature.

SUMMARY

The purpose of the investigation was to find a design for a solar energy air heater that can be used for drying hay and grain, and for heating buildings on the farm; and to determine the operating characteristics of the unit.

A review of the literature showed that solar heaters had been designed for heating water and air for household uses. The air heaters were designed to give a temperature rise of at least 110 degrees F., which is higher than is required for drying or heating farm buildings. On the basis of investigations performed by other researchers, a basic design was prepared. The collector consisted of an absorber plate with an air space and insulation below and an air space and one sheet of glass above the plate. The air to be heated was drawn through the two air spaces and thus passed over both sides of the absorber plate.

An application of the formulas for convection coefficient and pressure drop to the collector design showed that a shorter air passage is preferable to a longer one, and that the spacing between the absorber plate and the glass or insulation should be reduced until the pressure drop is the maximum that can be tolerated. The air flow rate considered optimum was one giving a Reynolds Number near the critical value.

The type of absorber plate judged to be best consisted of sheet metal coated with a mixture of lampblack and asphalt paint on the side exposed to solar radiation. The tests were performed using only asphalt paint.

Ordinary window glass was chosen for the covering because of its transmissivity and resistance to weather. It was found that more than one sheet of glass over the absorber plate would usually not increase the heat gain and temperature rise enough to justify the additional cost. It was also found that the air temperature rise could be increased by reducing the air flow rate per unit area of collector and increasing the total area of the collector to maintain the same total air flow rate.

Equations were developed to describe the operating characteristics of the collector. The validity of the equations was shown by comparing the air temperature rises predicted by the equations with those obtained with a collector which had been constructed and instrumented for testing. The maximum difference between the experimental and calculated values was 15 percent and the average was 7.3 percent. The range of temperature rises used was from 27 to 88 degrees F.

The equations indicate that the efficiency drops rapidly when air temperature rises approach their maximum values and air flow rates become low. At the smaller temperature rises and higher flow rates, the efficiency change is not so pronounced with air flow rate and air temperature rise change. The equations also indicate that the efficiency of the collector and the air temperature rise drop as the entering air temperature increases and the outside air temperature remains the same.

CONCLUSIONS

Collector Design

Air Passages

The shape of the passages in the heating section of the collector are important for efficient collector design. A study of formulas in literature giving convective heat transfer characteristics and pressure drops indicates that the distance the air moves between the plates should be as short as practical design will permit because the average coefficient of convection is higher for a short section with a shallow air space than for a longer one with a wider air space and the same pressure drop. The study also indicates that the depth of the air space should be as narrow as the maximum allowable pressure drop will permit because the narrower spacing has a higher convection coefficient for the same air flow rate. The rates of change of convection coefficient and of pressure drop are approximately equal when air flow rate changes. An air flow rate giving a Reynolds Number near the critical value between laminar and turbulent flow appears to give the best operating conditions.

Absorber Plate

The material which seems best suited for use as an absorber plate is sheet metal coated on the exposed side with a mixture of lampblack and asphalt paint. In order to keep the temperature of the absorber plate low and minimize radiant heat transfer to the glass surface, the air should pass on both sides of the plate. A design of this type has essentially twice the convective heat transfer potential as does one in which the air passes only along one side.

Glass Covering

The covering material which most nearly meets the requirements for a collector is glass. Since there are only slight differences in the transmissivities of glasses, the lowest priced, namely ordinary window glass, is recommended for use on solar energy air heaters. Several sheets of glass with air spaces between each will have a higher efficiency than one sheet, but at high air flow rates or small temperature rises the small decrease in heat loss by the several layers of glass may be more than offset by the additional cost of the glass. Fewer layers of glass may be used in some cases by decreasing the air flow rate per unit area and increasing the area. Thereby the temperature rise of the air and the total air flow rate will not be changed.

Operating Characteristics

Equations

The equations proposed to describe the operating characteristics of the collector combine the temperature rise of the air, the air flow rate, the incoming radiation, the absorptivity of the collector, and the entering air temperature. One equation describes the general relationship, another the efficiency, and two others are simplified forms of the general equation for certain operating conditions. It was found that the temperature rises of air predicted by the formulas differed from the temperature rises measured a maximum of 15 percent. The average difference was 7.3 percent.

Relationships of Parameters

The equations developed and the results of tests which verify the equations show that if only a small temperature rise is required, the overall coefficient of heat transfer is not a large factor in determining the temperature rise of the air. The opposite is true for large temperature rises. For that reason one sheet of glass is probably more economical than two for solar air heaters for farm use. At low air flow rates the efficiency increases rapidly with increase in air flow rate and less rapidly at the high rates. The maximum efficiency expressed as a decimal is equal to the absorptivity of the collector. It was also noted that the efficiency of the collector drops rapidly with an increase of air temperature when air temperature rise is great.

The general equation shows that the temperature rise of the air going through the collector is less if the entering air temperature is higher than outside air temperature.

Therefore it may be concluded that the efficiency increases with a decrease in entering air temperature.

RECOMMENDATIONS FOR FUTURE RESEARCH

Investigations of Paints and Coatings

The information that has been published concerning the short wave absorption and long wave emissivities of paints and other coatings and materials of construction is incomplete and often does not identify the paint, coating, or material exactly. Therefore it is recommended that an investigation be made of paints and coatings to determine the absorptivity in the solar radiation spectrum and emissivity at low temperatures. It may be possible to determine the effect of certain ingredients on the absorptivity and emissivity of the surfaces.

Tests with a Large Solar Energy Air Heater

The work which has been done was concerned with a small collector that was movable. Tests on a large collector would further verify the conclusions reached and also determine the operating characteristics of a stationary collector. A storage unit in conjunction with the collector would give information on the use of a complete air heating installation. Further tests should be performed by using the heated air to keep a building warm over winter and to dry hay and grain. Thus the problems that may be encountered in actual farm operations could be studied more closely.

Economy Study or Cost Analysis

Before the practicality of using a solar energy air heater on the farm can be determined, a cost analysis of such a system is necessary.

Energy Storage Units

Facilities for storing captured solar energy have been proposed but none have been designed specifically for farm use. It may be possible to arrive at a more practical and economical design that would fit the requirements for a farm heating system.

Farm Heating System Design

The application of the solar heating system to a farm requires studies to determine the best place for the collector, ducts, storage unit, and fans. The study would also make possible recommendations for orientation of buildings to make the best use of the heated air. The studies should also include an analysis of the warm air requirements to determine the size and slope of the collector.

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APPENDIX I

EXPERIMENTAL RESULTS

TABLE VI

TESTS WITH SINGLE COVER GLASS

AIR PASSAGES 9/16" DEEP

Test Dates: A-1 to A-5, September 19, 1955
A-6 and A-7, September 20, 1955

Barometric Pressures: A-1 to A-5, 29.19 in. Hg.
A-6 and A-7, 29.09 in. Hg.

Test No.	A-1	A-2	A-3	A-4	A-5	A-6	A-7
Dry bulb temp., °F.	91	93	93	93	94	81	82
Wet bulb temp., °F.	74	77	77	77	78	66	67
Lb. H_2O/lb. dry air	.0142	.0163	.0163	.0163	.0171	.0102	.0108
Anemometer, ft./min.	530	530	1120	1675	1985	2170	2773
Flow rate, cfm.	23.1	23.1	48.8	73.0	86.5	94.6	120.9
Incoming air, mv.	2.19	2.25	2.24	2.21	2.32	1.81	1.82
Incoming air, °F.	100	103	102	101	106	84	84
Ft^3/lb. at 29.92" Hg.	14.21	14.26	14.29	14.27	14.32	13.73	13.75
Ft^3/lb. at B. P. read.	14.57	14.62	14.64	14.63	14.68	14.12	14.14
Flow rate, lb/min.	1.59	1.58	3.33	4.99	5.89	6.70	8.55
Outgoing air, mv.	3.95	4.14	3.52	3.08	3.15	2.55	2.45
Outgoing air, mv.	4.07	4.26	3.58	3.15	3.24	2.66	2.49
Outgoing air, °F.	175	183	157	139	142	118	112
Outgoing air h, BTU/lb.	50.7	55.0	48.6	44.1	45.8	32.1	31.2
Entering air h, BTU/lb.	32.1	35.0	34.9	34.6	36.7	23.9	24.3
Enthalpy change, BTU/lb.	18.6	20.0	13.7	9.5	9.1	8.2	6.9
BTU/hr. gain by air	1774	1896	2737	2844	3216	3296	3540
Pyrheliometer, mv.	2600	2650	2750	2650	2650	2850	2850
Incoming BTU/(ft.^{2}hr.)	305	311	323	311	311	335	335
Collector efficiency, %	41.8	43.9	61.0	65.8	74.4	70.8	76.0
Absorber plate temp.							
Leading edge, mv.	2.69	2.67	2.42	2.33	2.34	1.89	1.86
Leading edge, °F.	122	120	110	106	106	87	86
1/4 length, mv.	3.87	3.98	3.50	3.23	3.21	2.65	2.52
1/4 length, °F.	170	174	155	144	143	119	114
Center, mv.	4.37	4.54	4.00	3.65	3.60	3.01	2.81
Center, °F.	190	196	175	161	159	135	126
3/4 length, mv.	*	*	*	*	*	*	*
3/4 length, °F	*	*	*	*	*	*	*
Trailing edge, mv.	4.61	4.93	4.38	3.88	3.82	3.18	3.00
Trailing edge, °F.	199	211	190	170	168	142	134

*Thermocouple leads were broken

TABLE VII

TESTS WITH SINGLE COVER GLASS

AIR PASSAGES 9/16" DEEP

Test Date: October 1, 1955

Barometric Pressure: 29.54 in. Hg.

Test No.	B-1	B-2	B-3	B-4	B-5	B-6	B-7
Dry bulb temp., °F.	69	65	66	65	64	64	65
Wet bulb temp., °F	56	54	55	55	54	54	55
Lb. H_2O/lb. dry air	.0065	.0064	.0070	.0069	.0066	.0066	.0069
Anemometer, ft./min.	511	895	1223	1967	2496	2586	2995
Flow rate, cfm.	22.3	39.0	53.3	85.8	108.8	112.7	130.6
Incoming air, mv.	1.34	1.42	1.45	1.38	1.30	1.34	1.30
Incoming air, °F.	63	66	68	64	61	63	61
Ft^3/lb. at 29.92" Hg.	13.21	13.26	13.28	13.22	13.19	13.22	13.23
Ft^3/lb. at B.P. read.	13.38	13.43	13.45	13.39	13.36	13.39	13.40
Flow rate, lb./min.	1.67	2.90	3.96	6.41	8.14	8.42	9.75
Outgoing air, mv.	3.32	2.78	2.50	2.18	2.01	1.96	1.86
Outgoing air, mv.	3.47	2.87	2.62	2.41	2.14	2.08	1.95
Outgoing air, °F.	151	127	116	105	95	93	88
Outgoing air h, BTU/lb.	36.3	30.2	28.2	25.3	22.5	22.0	21.1
Entering air h, BTU/lb.	14.5	15.1	16.2	15.1	14.0	14.5	14.5
Enthalpy change, BTU/lb.	21.8	15.1	12.0	10.2	8.5	7.5	6.6
BTU/hr. gain by air	2184	2627	2851	3923	4151	3789	3861
Pyrheliometer, mv.	2980	2980	2980	2980	2900	2900	2700
Incoming BTU/(ft.^{2}hr.)	350	350	350	350	340	340	317
Collector efficiency, %	44.9	54.0	58.6	80.6	87.8	80.2	87.6
Absorber plate temp.							
Leading edge, mv.	1.96	1.72	1.61	1.50	1.45	1.47	1.44
Leading edge, °F.	90	80	77	70	68	68	67
1/4 length, mv.	3.38	2.96	2.73	2.41	2.28	2.25	2.06
1/4 length, °F.	150	133	123	110	104	103	94
Center, mv.	3.89	3.36	3.11	2.72	2.52	2.47	2.23
Center, °F.	170	149	139	123	114	112	102
3/4 length, mv.	*	*	*	*	*	*	*
3/4 length, °F.	*	*	*	*	*	*	*
Trailing edge, mv.	4.25	3.55	3.38	2.94	2.73	2.65	2.38
Trailing edge, °F.	185	157	150	132	123	120	108

*Thermocouple leads were broken

APPENDIX II

NOMOGRAPH OF TEMPERATURE, SPECIFIC VOLUME ENTHALPY AND HUMIDITY OF AIR

SPECIFIC VOLUME AND ENTHAPY
OF AIR AT 29.92 IN. HG.