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The Use of Conditioned Air

For Maintaining Quality Of Stored Sorghum Grain

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Summary

Research was conducted to determine the feasibility of maintaining the quality of stored grain by controlling the temperature and relative humidity of the interstice air.

An important consideration in storing sorghum grain is the loss in quality resulting from improper storage conditions. Requirements for quality preservation, based on mold development, insect activity and germination are presented in this report.

Thermodynamic considerations as they relate to the design of controlled storage environments for bulk grain are discussed. The initial vapor pressure of the moisture in the grain and the partial pressure of the vapor in the conditioned air circulating through the grain mass were found to be very important thermodynamically in the design of controlled environment storage systems.

In this research, grain quality was maintained when air conditions surrounding the grain were maintained at a desirable temperature and relative humidity level. For example, no loss in quality resulted when grain having an initial moisture content of 18.19 percent (w.b.) was conditioned with air having a dry-bulb temperature of 45° F. and a relative humidity of approximately 70 percent for 194 days. Low-moisture grain with moisture contents as high as 14 percent was stored without quality deterioration by using conditioned air having a dry-bulb temperature of 55° F. and a relative humidity in equilibrium with 14-percent-moisture grain.

Factors governing the economical design of conditioned-air storage systems were studied. Results show that the time required to cool bulk-stored grain can be described by two time periods. The first period takes into account the lag time required for the leading edge of the cooling zone to move out of the grain mass. The other period is determined by the depth of the zone and the rate at which it moves. Both zone movements are described mathematically in this report.

Several methods were developed to determine the refrigeration capacity needed to maintain the quality of stored grain. These methods are described with a discussion of the effects of each on the equipment capacity and load requirements.

In all tests conducted to date, there was always a decrease in grain moisture content at the beginning of the cooling period regardless of the entering air relative humidity. This was due to the cold entering air being at a lower dew-point temperature, or vapor pressure, than the grain at its initial temperature. It was also shown that this moisture loss can be re-established and maintained at an initial level by controlling the conditions of the interstice air. Several methods of maintaining predetermined moisture levels are reported.

A solution to a typical design problem for a conditioned-air storage system, based on the findings of this research, is presented and discussed.

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The Use of Conditioned Air

For Maintaining Quality Of Stored Sorghum Grain

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SIGNIFICANT PROGRESS HAS BEEN MADE in recent years in the use of aeration equipment and techniques for cooling grain in storage. These practices have been effective for maintaining quality of low-moisture grain when accompanied by a program of inspection and adequate fumigation for insect control. However, problems have developed in aerated storages that are of great concern to elevator operators.

When natural air is used for aeration, the moisture content of the stored grain is often reduced below the desired level during the storage period. This is due to the drying which occurs when grain is cooled with natural air (1). This drying results in substantial loss in weight and in turn sizeable monetary loss.

For example, based on the amount of grain stored in Texas at the present time, a 1 percent reduction in moisture below the original storage level represents a potential annual loss of more than $2.7 million for sorghum grain alone. This points to a need for methods and equipment to reduce or, if possible, eliminate this loss. In tests conducted in Texas (2), a large part of the weight loss occurring over a 9-10-month storage period was due to a reduction in moisture during aeration.

A major hazard in storing grain is possible insect damage. Insects destroy an estimated 2 percent of the nation's stored grain annually (3). Not only do insects consume grain, but losses are also caused by heating, spoiling and reduction in grade caused by infestation. With the passage of the Pure Food, Drug, and Cosmetic Act in 1938, the Food and Drug Administration (FDA) recognized that the problem of internal or hidden infestation must be reduced or eliminated at the point of contamination if the consumer was to be protected from unsanitary foodstuffs. Flour mills, bakeries and other food industries are able to clean and remove foreign matter from grain but are only partially successful in removing internal grain infestation.

It is, of course, possible to control insects in stored grain by frequent inspections and fumigations. However, since insect infestation is a cycling problem, repeated use of chemical fumigants has caused additional problems which in some cases are more dangerous than contaminated grain. All fumigants are quickly lethal, or acutely toxic, to man at concentrations effective against insects, and elaborate safety precautions must be observed at all times if they are used (4). Also, the use of bromide and cyanide fumigants has created residue hazards in grain which is held in storage for long periods (4).

Preliminary tests have shown that the use of conditioned air may be an effective and economical

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method of controlling the quality of stored grain by eliminating insects, controlling moisture contents and maintaining the germination. Even though conditioned-air storage is nothing more than a method by which the temperature and moisture content of stored grain can be controlled in order to maintain quality, it accomplishes the same purposes as natural aeration systems without the same degree of risk.

REQUIREMENTS FOR GRAIN QUALITY PRESERVATION

An important consideration in storing sorghum grain is the loss in quality which may result from mold and insect activity if the grain is not stored in the proper condition. There is a definite relationship between grain temperature and moisture content and the time which grain can be held in storage before mold development and insect activity become a problem.

Mold Development

Christensen (5) found that moisture content, temperature and time are all intimately related to the growth of molds in stored grains. Thus, the higher the moisture and temperature, within the limits of growth of the fungi involved, the shorter the permissible storage time. Christensen found that wheat at a moisture content of 14.5-15.0 percent can be stored safely at 68°-77° F. for a few months, but not for a year, while at the same moisture content and at a temperature of 50°-59° F. it presumably could be stored for a year without serious damage from molds.

The lower limit of temperature for the growth of most storage molds is about 40° F., and the optimum temperature for growth of most of them is 80°-90° F. (6). Semeniuk (7) found that a minimum relative humidity of 80 percent in bulk bins is required for continued growth of molds. A relative humidity of 80 percent corresponds to an equilibrium moisture content of about 15 percent, wet basis, for sorghum grain at 70° F. (8).

Insect Activity

Insect activity in stored grain can also be correlated with time, temperature and moisture content. Sorghum grain having a moisture content of 9-10 percent normally will not support insect activity due to the low relative humidity of the interstice air. On the other hand, humidities in high-moisture grain still cannot support insect activity if the temperature is not in the optimum range. Pedersen (9) reports that stored-grain insects are capable of functioning only over certain ranges of temperature. Because of their inability to maintain a constant body temperature, they have to rely on the temperature of their environment. Grain temperature is probably the most significant factor affecting the distribution of stored-grain insects, since research has shown that these insects become inactive and eventually die of starvation at temperatures of 50°-60° F. (10).

Germination

Under official grain grading standards, germination tests are not required to establish grades for sorghum grain unless stored for planting purposes. However, germination tests are good indicators as to the condition of the stored grain. Grain which is low in germination is usually subjected to lower grades than high quality seed because of an increase in the damaged and cracked kernels. Germination tests are also an indicator as to the storage conditions, and there is evidence that grain with low germinating properties is also low in quality.

Recommendations

In order to maintain the quality of stored sorghum grain, the temperature and moisture content of the grain must be controlled. The maximum time to obtain a controlled grain condition before loss in quality occurs cannot be stated precisely. However, the time periods shown in Table 1 have proven satisfactory when grain is stored under natural aeration methods at various moisture contents and an initial temperature of 85°-95° F.

EQUIPMENT AND PROCEDURES

The overall objectives of this research follow:

1. To determine the design factors, equipment requirements and operating procedures for maintaining quality of both low- and high-moisture grain stored under controlled air conditions. Factors used as a measure of quality were mold damage, insect damage, U.S. grade and germination.

2. To develop methods, procedures and equipment for maintaining a selected or predetermined level of moisture in stored grain.

To accomplish the objectives of this research, three test bins were used, Figure 1. Each bin was 6 feet in diameter. They were designated Bin 1, Bin 2 and Bin 3. Test Bin 1 consisted of a 4-foot diameter bin centered inside the 6-foot bin. This provided a 12-inch layer of grain which acted as an insulator and reduced the heat flow into the test section. The walls of the outer bin were insulated with 2 inches of insulation having a thermal conductivity of 0.25 Btu per (hour) (square foot) (°F. per inch). Both bins were installed on a perforated floor

TABLE 1. MAXIMUM ALLOWABLE DAYS FOR COOLING SORGHUM GRAIN STORED AT INITIAL TEMPERATURES OF 85°-95° F. BEFORE LOSS IN QUALITY RESULTS FROM INSECT AND/OR MOLD ACTIVITY

Grain moisture content—percent (W.B.)	Maximum allowable cooling time—days
12 to 14	30
14 to 16	20
16 to 18	7

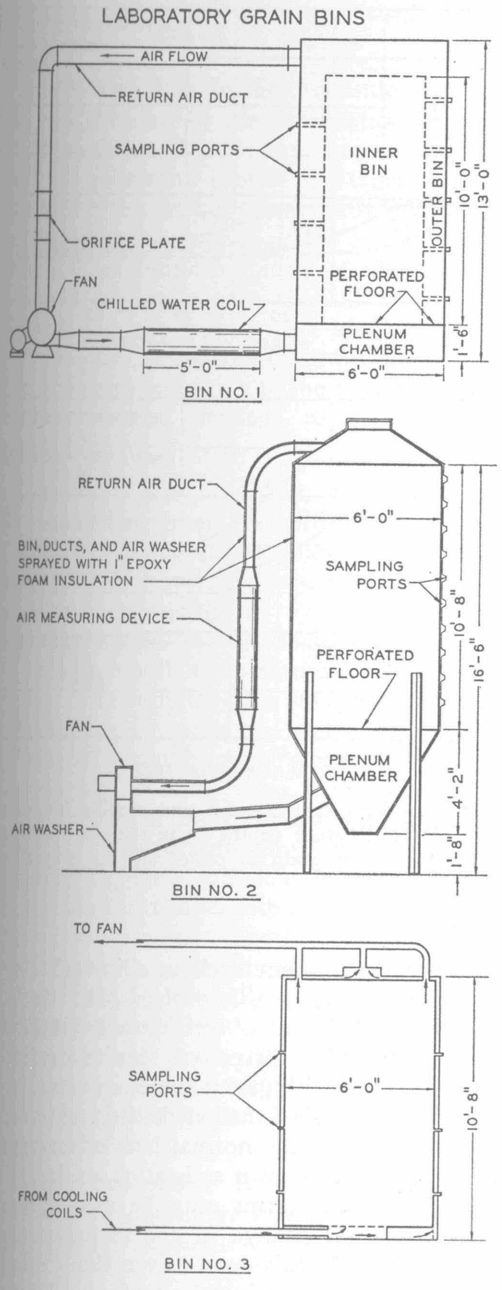

Figure 1. Cross-sectional view of experimental conditioned-air storage bins.

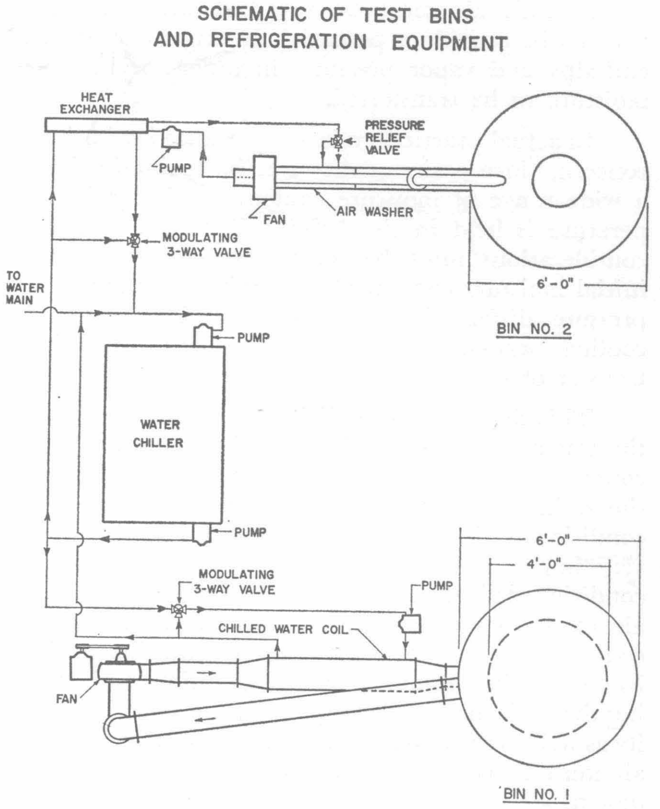

Figure 2. Schematic of test bins and refrigeration equipment.

attached to a plenum chamber 6 feet in diameter and 18 inches high.

Test Bin 2 was constructed from a standard 290-bushel capacity hopper-bottom bin sprayed with 1 inch of epoxy foam insulation. A perforated floor was installed creating an air plenum in the hopper-shaped portion of the bin. Bin 3 was constructed with a double wall so that a 2-inch space was provided for air circulation in order to minimize heat gain. Four inches of insulation having the same thermal conductivity as Bin 1 were installed on this bin. Each test bin was sealed at the top, and air ducts were installed to provide a closed system for recirculating conditioned air. The direction of air flow was from the bottom to the top of the grain mass in each bin.

Two conditioning units were constructed in order to provide conditioned air to the test bins. Bin 1 and Bin 3 were interchanged so that the same conditioning unit could be used for both bins. This unit consisted of a chilled water coil with the necessary controls. The bins were arranged so that when Bin 3 was connected to the conditioning system, Bin 1 could be operated as a conventional aeration system using atmospheric air. A chilled-water-spray air washer was used to condition air for Bin 2. Each of the conditioning units was piped to a 2-ton water chiller, Figure 2.

In each bin, sampling ports were installed at each foot level to allow grain samples to be taken for test purposes. Thermocouples were also located at each foot along the center axis of the bins and at various other points in the bin and circulating systems for temperature measurements.

THERMODYNAMIC CONSIDERATIONS

Grain temperature and moisture content can be established in bulk stored grain by controlling the conditions of the air surrounding the individual kernels. To maintain these air conditions for a period long enough to establish a predetermined grain condition requires that air be circulated through the grain mass. This air must be supplied with properties which will provide a potential driving force, such as enthalpy and vapor pressure, in order for heat and moisture to be transferred.

In actual practice, grain temperature is of primary concern, since grain quality can be maintained over a wide range of moisture contents if the grain temperature is held in the 50°-60° F. range. Moisture considerations must be secondary regardless of the initial moisture content of the grain, because a vapor pressure differential will usually exist during the cooling period. This differential will result in a transfer of moisture from the grain to the air.

This does not mean that the partial pressure of the water vapor in the air is not important. On the contrary, the initial vapor pressure of the moisture in the grain and the partial pressure of the vapor in the conditioned air circulating through the grain mass is very important thermodynamically in the design of conditioned-air-storage systems. In fact, within certain air dry-bulb temperature limits and grain moisture contents, grain temperatures can be controlled entirely by the specific humidity of the conditioned air entering the grain mass. Consequently, the specific humidity as well as the dry-bulb temperature of the circulated air must be considered in any environmental storage design.

Consider the effects of an improper design humidity on the grain temperature in an installation designed to cool the grain in small increments of temperature. The results obtained are shown in Figure 3. Grain was placed in storage at a temperature of approximately 90° F., and a design dry-bulb temperature of the air entering the grain was selected at 85° F. The specific humidity of the air entering the grain was at a level to maintain the entering air relative humidity at 80 percent, while the equilibrium relative humidity of the grain was 60 percent. This caused moisture to be transferred from the air to the grain at a rate which allowed a net exchange of heat to the grain, thereby increasing its temperature.

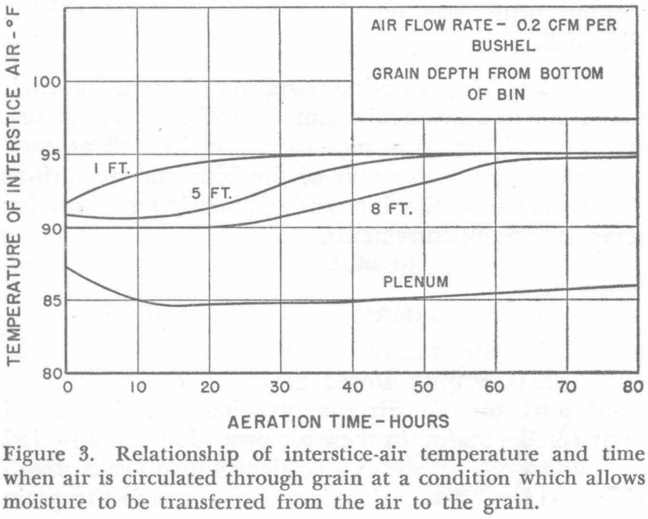

Figure 3. Relationship of interstice-air temperature and time when air is circulated through grain at a condition which allows moisture to be transferred from the air to the grain.

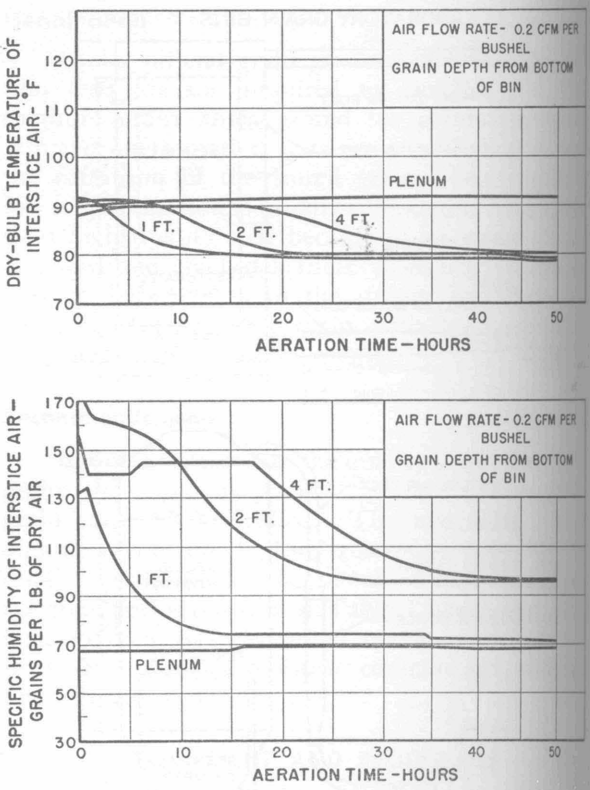

Figure 4. Dry-bulb temperatures and specific humidities of interstice air at different depths along the center axis of the mass of grain when grain is cooled by the evaporation of moisture.

When an adsorbent such as silica gel is wetted by a film of water, heat is evolved (11). Such heat liberation is called heat of wetting and is believed to be due to attractive forces. A similar condition is expected to occur with grain; therefore, the total heat adsorbed by the grain is called the heat of adsorption which is the sum of the normal heat of condensation plus some quantity known as heat of wetting. These two individual heat terms must be used in order to calculate the total change in the enthalpy of grain during a process of this type. Even though the temperature of the air entering the grain was lower than the temperature of the grain, a sufficient amount of moisture was transferred to the grain for the heat of adsorption to more than offset the decrease in grain enthalpy due to the grain-air temperature differential. This resulted in a net increase in the enthalpy of the grain.

The thermodynamic effects resulting from the proper design specific humidity in controlled environments required to cool grain can be beneficial in bulk stored grain. If grain moisture content can be reduced during the storage period without adversely affecting the economy of the storage operation, then the time required for cooling or the refrigeration load can be reduced. These reductions result from the evaporative cooling effect obtained when moisture is transferred to the air from the grain. The actual reduction in the cooling time and load cannot be predicted at present.

To illustrate this further, consider the results when grain is cooled by the evaporation of moisture alone. The relationships of temperature and humidity to time of cooling are shown in Figure 4. In this storage facility the dry-bulb temperature of the conditioned air was held constant at the initial grain temperature. The specific humidity of the entering air was selected at a value that allowed moisture to be transferred from the grain to the air. Results show that as moisture is evaporated from the grain, the air and grain are cooled due to the sensible and latent heat of vaporization exchange, as is the case in an evaporative cooling process.

In the adiabatic saturation process, Figure 5, partly saturated air enters the saturator at state 1. If the chamber contains an adequate supply of water, then the air will leave in a saturated condition. If water is at the entering air wet-bulb temperature, then the leaving air dry-bulb temperature will equal the entering wet-bulb temperature. However, if the water cannot be supplied at a sufficient rate to saturate the leaving air, as is the case with grain, then the leaving air dry-bulb temperature would not equal the entering air wet-bulb. Also, if the water temperature was initially at the dry-bulb temperature (T_{d1}) then the water temperature could only approach the temperature T_{w1} until sufficient moisture was evaporated to attain this temperature. If the dry matter in the grain is at the same temperature as the water during the process, then the internal grain temperature would approach the wet-bulb temperature during the period in which moisture was removed. The temperature at which the grain approaches the wet-bulb temperature of the surrounding air depends upon the quantity of moisture evaporated and the latent heat of vaporization.

The diagram shown in Figure 6 represents a heat balance for cooling grain in storage. The total heat entering and leaving the system must be equal, or

$$H_2 = H_1 + H_W + H_K + Q \qquad (1)$$

where

H_1 = enthalpy of entering air mixture, Btu
H_2 = enthalpy of leaving air mixture, Btu
H_W = enthalpy of rejected moisture, Btu
H_K = enthalpy of rejected heat, Btu
Q = heat gain from outside, Btu

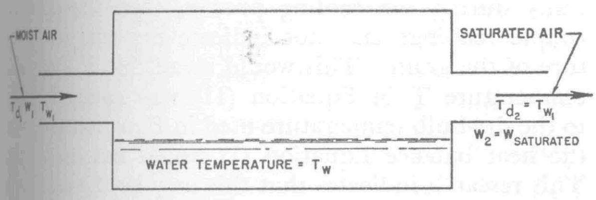

Figure 5. Schematic of an adiabatic saturation device.

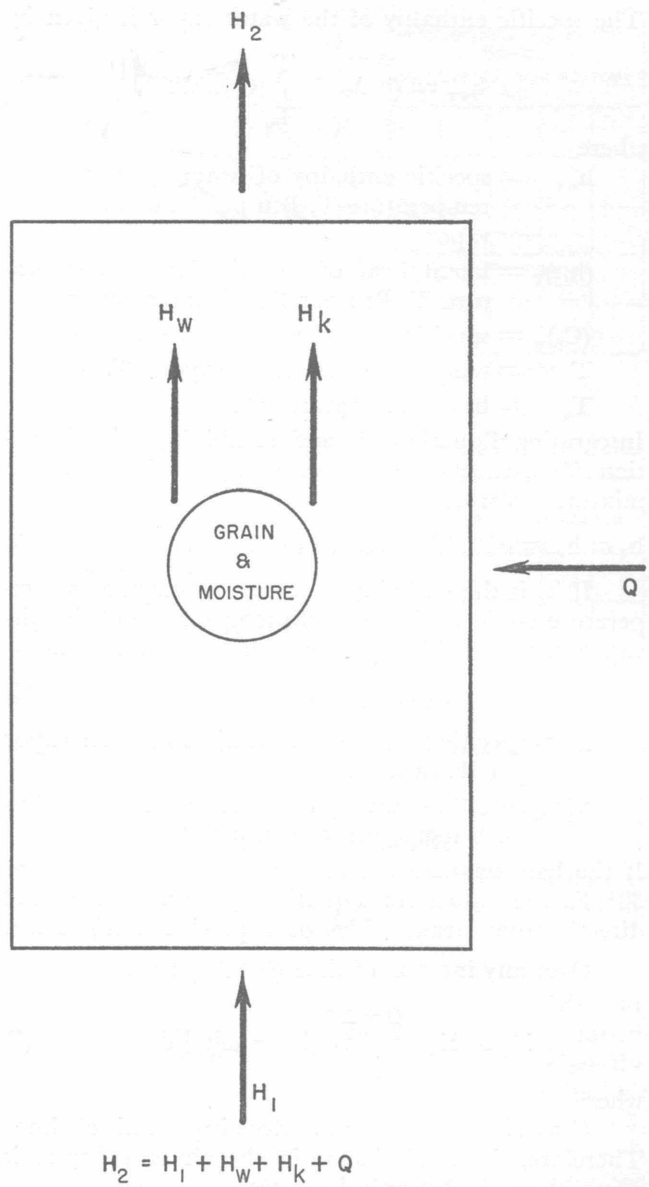

Figure 6. Heat balance diagram for cooling grain in storage.

The enthalpy of the moist air entering (H_1) and leaving (H_2) can be calculated as the sum of the individual specific enthalpies of the dry air and water vapor components of the mixture. The specific enthalpy of the dry air is given as

$$h_a = \int_{T_o}^{T} (C_p)_a \, dT \qquad (2)$$

where

h_a = specific enthalpy of dry air, Btu per lb.
T = temperature of air, °F.
T_o = base temperature, °F.
$(C_p)_a$ = specific heat of dry air, Btu per lb.-°F.

Assuming a base temperature (T_o) of 0° F. then the solution to Equation (2) becomes

$$h_a = (C_p)_a T \qquad (3)$$

The specific enthalpy of the water vapor is given by

$$h_{wv} = (h_{fg})_T + \int_{T_o}^{T} (C_p)_w \, dT \tag{4}$$

where

h_{wv} = specific enthalpy of water vapor at temperature T, Btu per lb. of water vapor

$(h_{fg})_T$ = latent heat of vaporization at temperature T, Btu per lb. of water vapor

$(C_p)_w$ = specific heat of water, Btu per lb.-°F.

T = temperature of water vapor, °F.

T_o = base temperature, °F.

Integrating Equation (4) and combining with Equation (3) then the specific enthalpy of the saturated mixture becomes

$$h_1 \text{ or } h_2 = (C_p)_a T + [(C_p)_w (T - T_o) + (h_{fg})_T] \, \omega \tag{5}$$

If h_g is the enthalpy of saturated steam at a temperature equal to the dry-bulb temperature of the air-vapor mixture, this equation can be expressed as

$$h_1 \text{ or } h_2 = h_a + \omega h_g \tag{6}$$

ω = specific humidity of the air, lbs. water vapor per lb. of dry air

h_g = specific enthalpy of water vapor at the dry-bulb temperature, Btu per lb.

If the base temperature of water vapor is selected as 32° F., the values for Equation (6) can be obtained directly from steam tables or a psychrometric chart.

Over any interval of time $(\Theta + \Delta\Theta)$ then

$$H_1 \text{ or } H_2 = \int_{\Theta}^{\Theta+\Delta\Theta} G \, [h_a + \omega h_g] \, d\Theta \tag{7}$$

where

G = air flow rate, lbs. of dry air per unit of time

Therefore, the total change in the air enthalpy as it passes through the grain becomes

$$H_2 - H_1 = \int_{\Theta}^{\Theta+\Delta\Theta} G \, [h_{a2} + \omega_2 h_{g2}] \, d\Theta - \int_{\Theta}^{\Theta+\Delta\Theta} G \, [h_{a1} + \omega_1 h_{g1}] \, d\Theta \tag{8}$$

At the present time, it is assumed that the terms H_W and H_K in Equation (1) represent the total enthalpy change of the grain. There is some question as to the validity of this assumption, and this will be discussed later. With this assumption, the specific enthalpy of the grain can be calculated by

$$h_G = h_{dg} + \omega_{wg} h_f \tag{9}$$

where

h_G = specific enthalpy of moist grain, Btu per lb. dry grain

h_{dg} = specific enthalpy of dry grain, Btu per lb.

ω_{wg} = specific humidity of grain, lbs. water per lb. dry grain

h_f = enthalpy of water in grain, Btu per lb.

Making the proper substitutions in Equation (9) the total numerical change in the grain enthalpy for a total depth L during any time period Θ to $\Theta + \Delta\Theta$ can be determined by

$$\Delta H_G = \left[\int_0^L [W_{dg} (C_p)_{dg} T] \, dL + \int_0^L [W_{wg} (C_p)_w T] \, dL \right]_{\Theta+\Delta\Theta} - \left[\int_0^L [W_{dg} (C_p)_{dg} T] \, dL + \int_0^L [W_{wg} (C_p)_w T] \, dL \right]_{\Theta} \tag{10}$$

where

ΔH_G = enthalpy change of grain, Btu

W_{dg} = weight of dry grain, lbs. per unit depth

W_{wg} = weight of water in grain, lbs. per unit depth

$(C_p)_{dg}$ = specific heat of dry grain, Btu per lb.-°F.

$(C_p)_w$ = specific heat of water, Btu per lb.-°F.

T = grain temperature as a function of depth, °F.

Due to the difficulty of expressing T and W_{wg} in terms of L in Equation (10) an accurate numerical approximation of ΔH_G can be obtained by the following equation for (n) small increments of L, where $n = \frac{L}{\Delta L}$

$$\Delta H_G = \sum_{a=1}^{n} (W_{dg})_a (C_p)_{dg} (T_f)_a + \sum_{a=1}^{n} (W_{wgf})_a (C_p)_w (T_f)_a - \sum_{a=1}^{n} (W_{dg})_a (C_p)_{dg} (T_i)_a - \sum_{a=1}^{n} (W_{wgi})_a (C_p)_w (T_i)_a \tag{11}$$

Subscripts f and i denote final and initial conditions, respectively.

The normal procedures for determining the internal conditions of a grain mass in a controlled environment storage are to observe the temperatures with thermocouples located within the mass and to determine the moisture content of the grain periodically. These temperatures are usually assumed to be both air and grain temperatures. If moisture is being transferred from the grain to the air adiabatically during the cooling process, then the thermocouple readings may not indicate the true temperature of the grain. This would mean that if the grain temperature T in Equation (11) was assumed equal to the dry-bulb temperature used in Equation (8), then the heat balance Equation (1) would not hold true. This research indicates that this may be the case even though no work has been reported which proves that the grain temperature is lower than the indicated air temperature while moisture is being transferred.

If the grain temperature is assumed to be at the observed air temperature, it then appears that Equation (*1*) must have an additional term on the right side of the equation for equality. Written in terms of specific enthalpy then, the following may apply whenever moisture has been transferred from the grain to the air.

$$h_G = h_{dg} + \omega_{wg} h_f - \Delta H \qquad (12)$$

where

h_G = final specific enthalpy of the moist grain, Btu per lb. dry grain

h_{dg} = final specific enthalpy of the dry grain, Btu per lb.

ω_{wg} = final specific humidity of grain, lbs. water per lb. of dry grain

h_f = enthalpy of water in grain, Btu per lb.

ΔH = heat due to moisture transfer, Btu per lb. dry grain

Discounting the ΔH term in Equation (12) could result in an under design of the system cooling capacity. This is, of course, still under the assumption that the final air and grain temperatures are equal. In this case, the ΔH term must be reduced to some mathematical expression for design purposes.

The simplest procedure for estimating the required cooling capacity in a system design would be to solve for the sensible grain load in the following equation:

$$Q_G = W\, C_p\, \Delta T \qquad (13)$$

where

Q_G = grain load, Btu

W = initial weight of grain, lbs.

C_p = specific heat of grain at initial moisture content, Btu per lb.-°F.

ΔT = difference in initial and final grain temperatures, °F.

However, tests have shown that the calculated load in this equation is not sufficient to cool the grain to the

TABLE 2. SPECIFIC HEAT OF GRAIN—BTU PER POUND —DEGREES FAHRENHEIT

Investigator	Approximate grain moisture content, percent	Wheat	Barley	Oats	Rice	Sorghum grain
Pfalzner (12)	13.0	0.396				
Babbitt (13)	13.0	0.370				
Disney (14)	13.0	0.405	0.387			
Haswell (15)	13.0			0.419	0.402	
Kelley (16)	13.0	0.390				
Miller (17)	12.0					0.447
Miller (17)	18.0					0.490
Miller (17)	24.0					0.533

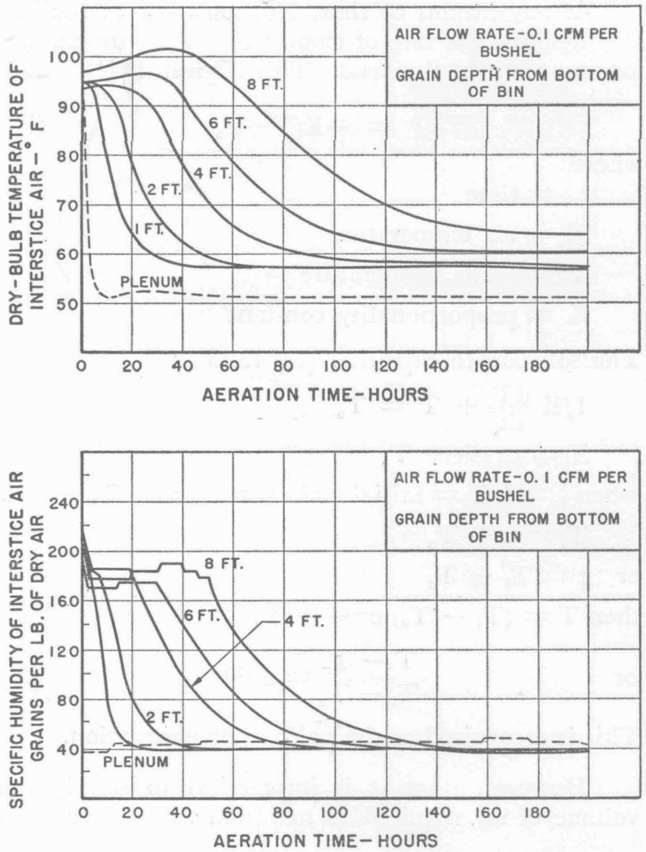

Figure 7. Dry-bulb temperatures and specific humidities of interstice air at different depths along the center axis of the mass of grain during the cooling period.

final design temperature unless the entering air temperature is lower than the final design temperature of the grain. If Equation (13) is used to calculate the load required to cool the grain to the entering air temperature under the conditions shown in Figure 7, then the final temperature of the grain would be that shown after 200 hours of operation if it is assumed that a thermocouple indicates the true grain temperature. This load compares closely to the change in the air load during the cooling period when the specific heat of grain at different moisture contents is selected from Table 2. It should be noted that the grain did not reach the design temperature. The term ΔH could be responsible for this difference under the equal grain-air temperature assumption.

DESIGN FACTORS

Cooling Zone Movement

When cool air is forced through a mass of grain, a cooling zone will develop and progress through the grain in the direction of air flow. This zone movement is illustrated by the temperature patterns in Figure 7. The thickness of the zone and the speed at which it can progress through the grain mass depends upon some function which describes the cooling rate of grain in relation to air velocity.

At any instant of time, Newton's law of cooling may describe the rate of change of grain temperature for any point in the mass. This is given by

$$\frac{dT}{dt} = -K(T-T_a) \qquad (14)$$

where

t = time

T_a = air temperature

T = grain temperature

K = proportionality constant

The solution to Equation (14) follows:

$$1/K\frac{dT}{dt} + T = T_a$$

$$T = ce^{-Kt} + T_a$$

when $t = 0$, T = initial grain temperature, T_i, so that

$$T_i = c + T_a$$

or $c = T_i - T_a$

then $T = (T_i - T_a)\, e^{-Kt} + T_a$

or

$$\frac{T - T_a}{T_i - T_a} = e^{-Kt} \qquad (15)$$

This is recognized as the half-response equation.

However, since it is impractical to supply the volume of air required to maintain the air temperature surrounding each kernel of grain at a constant level, Equation (15) cannot be applied directly but must be related to airflow. For design purposes, the effect of air flow rate on cooling time must be described by two different time periods. The first period would account for the lag time required for the leading edge of the zone to move out of the grain mass. The other period would be determined by the depth of the zone and the rate at which it moves.

The time for the leading edge of the zone to move through the grain (T_L) has been described by Miller (18) as the time at which the air leaving the grain first starts to decrease in temperature. This time can be expressed as

$$T_L = 3.9\ Q^{-0.94} \qquad (16)$$

where

T_L = time for leading edge to move through grain, hours

Q = flow rate of air entering the grain, cfm per bushel

Equation (16) is valid over all ranges of air and grain conditions.

The conditions of the air and grain will influence the movement of the trailing edge of the zone due to such factors as evaporative cooling and heat of wetting.

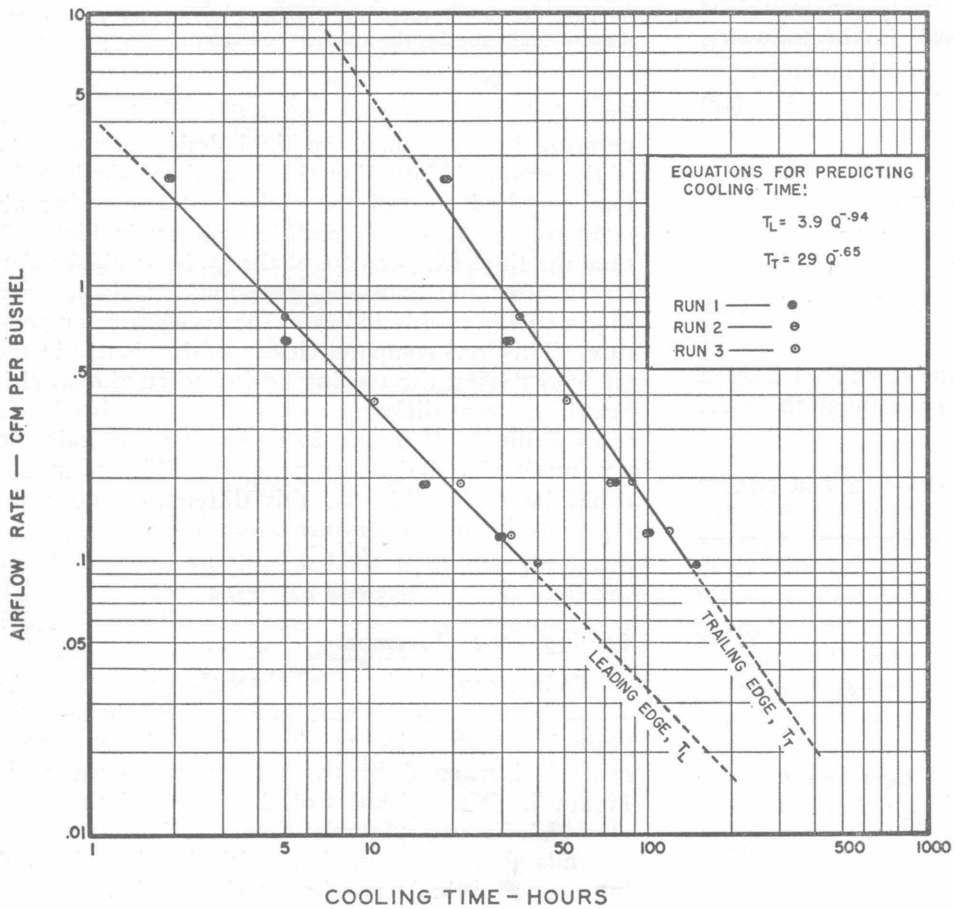

Figure 8. Time required for cooling zone to move through sorghum grain aerated with air in equilibrium with the initial grain moisture content and the desired final temperature.

Whenever the air conditions entering the grain are controlled at a constant dry-bulb temperature and a relative humidity which would be in equilibrium with the cooled grain at the initial moisture content, then the movement can be closely approximated by

$$T_T = 29\ Q^{-0.65} \tag{17}$$

where

T_T = time for trailing edge to move through grain, hours

Equations (16) and (17) have been plotted in Figure 8 for convenience.

The time for the trailing edge of the zone to move through a grain mass, Figure 8, would represent the total cooling time for the mass whenever the entering air dry-bulb temperature has been selected at a value equal to the final desired grain temperature.

Grain Temperature

The final temperature which the grain can attain in conditioned-air storages may not be the same as the dry-bulb temperature of the entering air. In most cases, it will only approach the entering air temperature. This is probably due to one of the following factors: (1) method used to determine the actual grain temperature; (2) amount of moisture transferred from the air to the grain; and (3) quantity of heat moving from the outside into the grain mass.

If the interstice-air temperatures shown in Figure 9 are considered to be the true grain temperature, then the difference between grain and entering-air temperatures would be due to heat gain and moisture transfer. The major portion of this difference, in this case, probably resulted from heat gain due to the low airflow rate used and the small distance from the wall to the point of measurement. The temperatures in Figure 10 as compared to those in Figure 9 indicate that the increased airflow rate decreases the difference between the interstice air and entering-air temperatures. All the difference in temperature, however, is

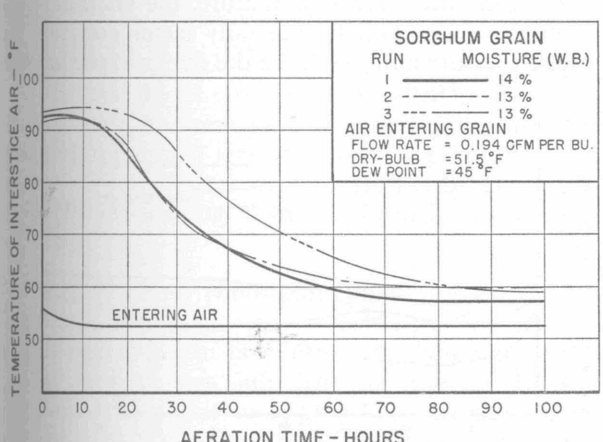

Figure 9. Grain cooling pattern showing the relationship of interstice-air and entering-air temperatures.

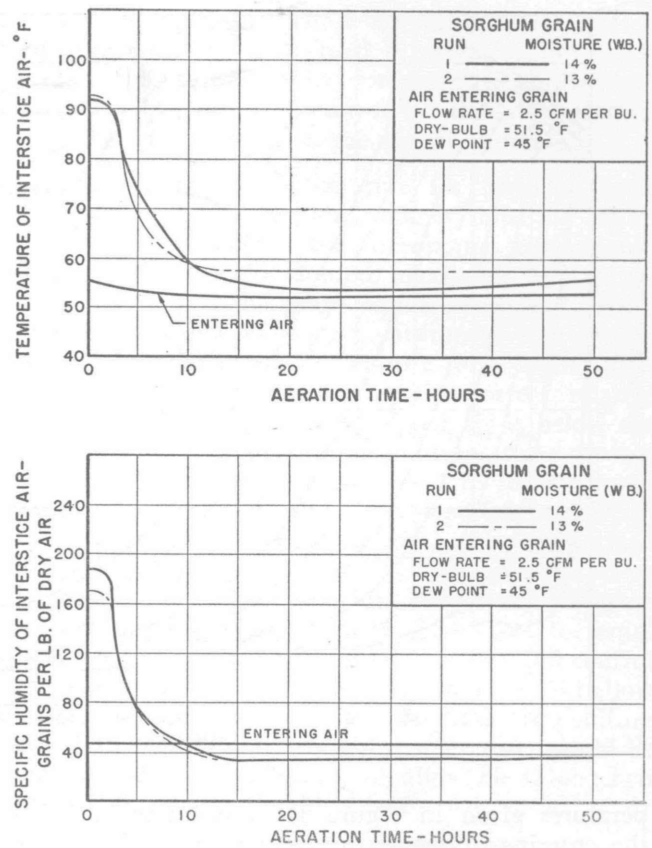

Figure 10. Temperature and specific humidity of interstice air in relation to aeration time when cooling a mass of grain in storage.

not due to heat gain. Since the interstice-air temperature of 14 percent grain approaches closer to the entering-air temperature, then it must be concluded that the rate of moisture transfer has some influence. The interstice-air specific humidity leaving both the 13 and 14 percent moisture grain was lower than the specific humidity of the entering air after approximately 10 hours for an airflow rate of 2.5 cfm per bushel. This indicates that moisture was being transferred to the grain from the air after the initial 10-hour period. Even though it appears from Figure 10 that an equal quantity of moisture was transferred, it must be assumed that the 13 percent grain would adsorb more moisture than the 14 percent moisture grain when the specific humidities of the interstice air are the same. This being the case, the energy required to condense the water vapor onto the grain surface would cause the greater difference in temperatures observed for 13 percent moisture grain than the 14 percent grain.

If energy is released from the grain in order to evaporate moisture, the temperature of the interstice air could be lower than the dry-bulb temperature of the entering air. This would be true if sufficient moisture could be transferred during the entire cooling period, as shown in Figure 11. Interstice-air tem-

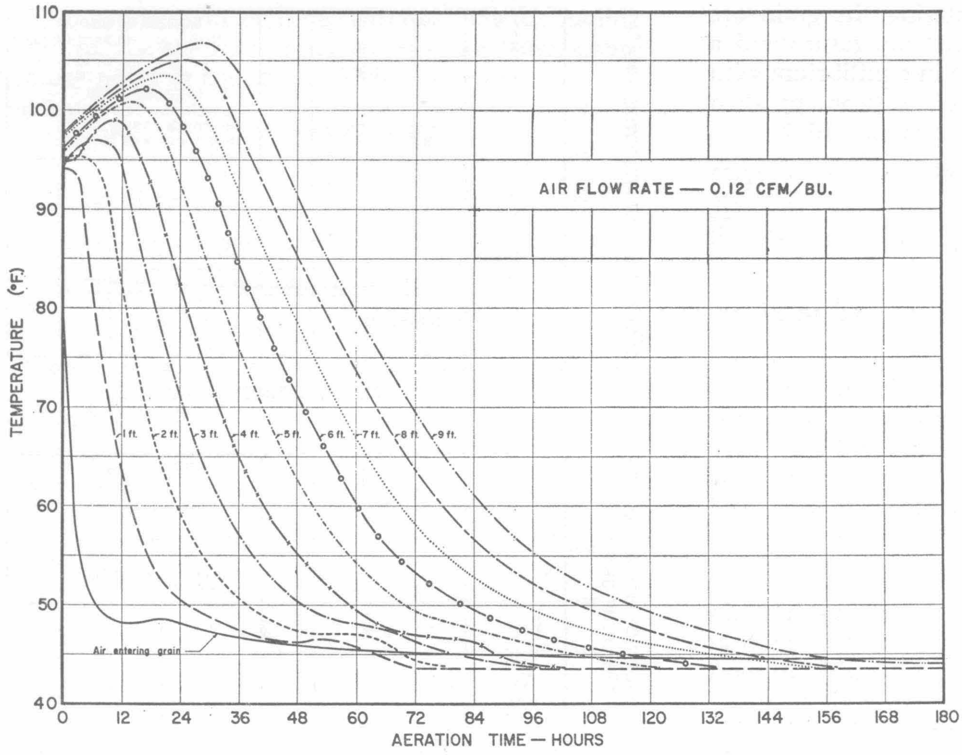

Figure 11. Relationship of grain depth and aeration time to interstice-air temperature.

peratures given in Figure 11 would remain below the entering-air temperture as long as moisture was being evaporated from the grain. Once the moisture transfer is reversed, then the interstice-air temperatures would probably rise above the entering-air temperature.

Grain Moisture Content

The effect of any storage method or procedure on the final grain moisture content is related to the difference in the vapor pressures of the moisture in the grain and the air surrounding the grain. The flow of moisture between air and grain is always from points of high to low vapor pressure. In order to prevent any transfer of moisture during the storage period, the difference in the vapor pressures between the grain and air must be zero.

In storage facilities using atmospheric air to cool grain, the air is allowed to circulate through the grain any time the dry-bulb temperature is at a desirable level without any regard to the vapor pressure. In most areas of Texas when this procedure is used, there is usually a differential in the grain and air vapor pressures which results in a loss in grain moisture content. A comparison of the moisture content of grain aerated with atmospheric air and grain aerated with conditioned-air is given in Figure 12. After a storage period of 188 days, the average moisture content of grain aerated with atmospheric-air was reduced from 12.80 to 12.35 percent (wet basis). This was a moisture loss of 0.45 percentage point. Two bins of grain aerated with conditioned-air gained 0.53 and 0.26 percentage points in moisture over the same storage period. The amount of moisture loss depends on the entering air conditions and rate of airflow.

In order to prevent grain moisture transfer during the storage period, it must be realized that grain at any constant level of temperature and moisture content can have one and only one partial pressure associated with its internal moisture. Consequently, there can be only one pressure due to the water vapor in the air which would prevent any moisture transfer between the air and grain. Also the temperature of the grain must be reduced to some maximum level for quality preservation.

The dry-bulb temperature of the air entering the grain would eventually be fixed at some level since the circulating air would be the heat transfer medium. For a constant grain temperature, the grain moisture content would then be the only factor governing its vapor pressure and would be the design vapor pressure

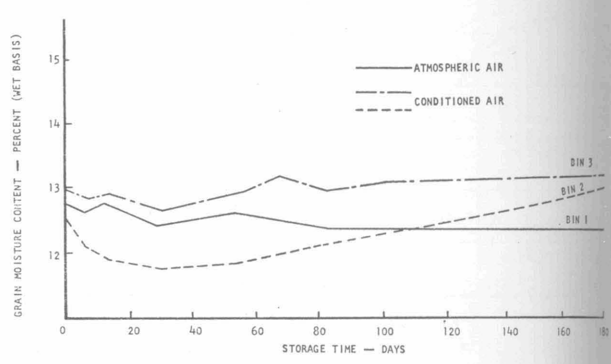

Figure 12. Comparative moisture contents of grain aerated with natural air and conditioned air.

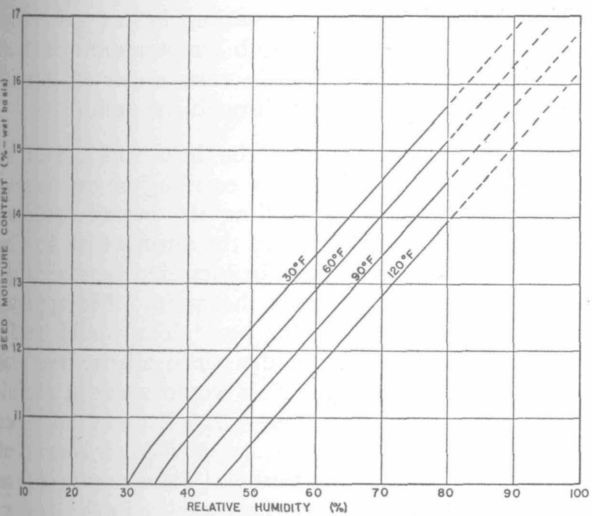

Figure 13. Sorghum grain equilibrium relative humidity curves.

of the air required to maintain the desired grain moisture.

In order to maintain an equilibrium condition while circulating air through a grain mass, it is necessary to control the specific humidity of the entering air since there is only one specific humidity at which the vapor pressures are equal. To maintain moisture equilibrium, the specific humidity must be considered as one property of the air condition which is required. The other property is the dry-bulb temperature of the air, as determined by the necessary final grain temperature to prevent loss of quality. These two properties will then define a state at which only one relative humidity can exist. Therefore, the necessary air conditions for moisture equilibrium can then be defined in terms of the relative humidity at a predetermined dry-bulb temperature and can be obtained from graphs similar to the one in Figure 13.

Once the equilibrium relative humidity has been determined, some economical method must be used to regulate the humidity of the entering air at this value. Three methods were investigated in this research. These methods involved two possible principles: reheat and return-air bypass.

Electrical duct heaters were installed in two bins, each having a different type of conditioning unit for obtaining the proper air conditions for grain moisture control. These heaters were controlled by Dunmore-type humidity sensing elements through suitable on-off relays.

One of the air conditioning units was a chilled water coil, while the other system used was an air-washer type unit. In both cases, the flow rate of the cooling medium was sufficient for the air leaving the conditioning unit to approach a saturated state. As long as this saturated condition was maintained, the dew-point temperature of the air could be controlled accurately by controlling the temperature of the cooling medium. The temperature of the cooling medium could then be used to establish the vapor pressure of the air needed for moisture equilibrium since the dew-point temperature has the same relationship to vapor pressure as the specific humidity.

The dew-point temperature for moisture equilibrium in conditioned-air storages must be maintained at or below the dew-point temperature determined from the desired maximum final grain temperature and the grain equilibrium relative humidity. Any dew-point temperature above this value would cause the grain temperature to be higher than the maximum temperature at the correct relative humidity. If the dew point of the air is below the desired value, the equilibrium grain moisture content would increase since the air is still in the saturated state. However, if sensible heat is added to this air, the dew point and vapor pressure can be maintained at the same level while the dry-bulb temperature is increased. This increase in dry bulb would decrease the relative humidity to the desired value for equilibrium. It is then necessary only to sense and control the relative humidity at some value which would allow no transfer of moisture. The dry-bulb temperature of the air, and hence the grain temperature, would be maintained below the maximum allowable value when the proper relative humidity is reached.

The grain temperature can be maintained at the maximum value only if the dew-point temperature of the air is equal to the dew-point temperature corresponding to the grain moisture vapor pressure. The needed relative humidity would be reached at the exact maximum grain temperature whenever the air was heated sensibly. The dry-bulb temperature as well as the relative humidity could then be used for control purposes.

Figure 14 illustrates the control of grain moisture content by using a dew-point temperature below the

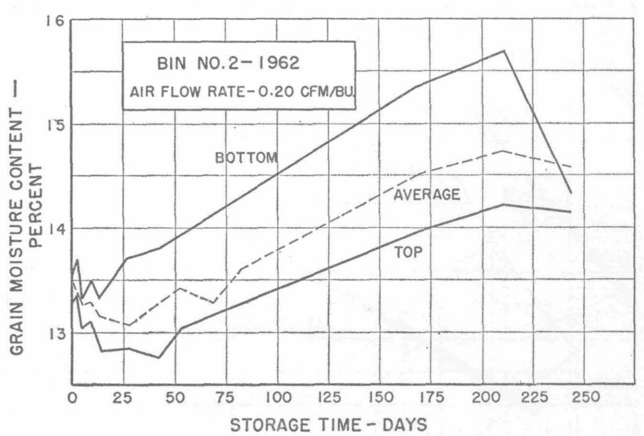

Figure 14. Illustration showing the relationship of grain moisture content to storage time when the dew-point temperature was maintained below the required level for moisture equilibrium and the relative humidity controlled to provide equilibrium at the original grain moisture content.

needed level and controlling the relative humidity. The relative humidity of the circulated air was maintained at approximately 80 percent for the first 210 days of storage after which time the air was controlled at the proper condition for moisture equilibrium at the original grain moisture content. The curves in Figure 14 show that the moisture content increased, after the initial loss, up to the time the control was started. After this period the moisture began to decrease and would have returned to the initial level if the proper air conditions had been maintained for sufficient time.

Another procedure which can be used in conjunction with reheat is to limit the quantity of heat supplied to the air without any type of control. The amount of heat necessary for maintaining the correct air conditions for a zero vapor pressure differential can be accurately calculated if the air conditions entering the grain are not influenced by outside conditions. This is not the case in practical installations because of heat gain into the air-supply duct. The amount of heat moving into the duct varies with the outside air temperature but can be determined as some mean value. If this value is below the quantity of heat needed to establish the relative humidity, then additional heat must be supplied by the heater.

This procedure was used in Bin 2 during one of the tests. The available results from this test indicate that grain moisture equilibrium can be maintained within certain limits. The relative humidity of the air which was supplied to the grain was selected at a level to maintain the moisture content at approximately 13 percent instead of the initial moisture content of approximately 12.5 percent. The heater voltage was then set to maintain a mean relative humidity for moisture equilibrium at this level. The curve for Bin 2 in Figure 12 shows the results. It is anticipated that the curve would level out at approximately 13.25 percent moisture content after sufficient time but would oscillate about this moisture level due to the hourly fluctuations in heat gain. It should also be pointed out that these fluctuations in heat gain cause an oscillation in the dry-bulb temperature of the air entering the grain. This in turn causes an infinite number of zones to move through the grain.

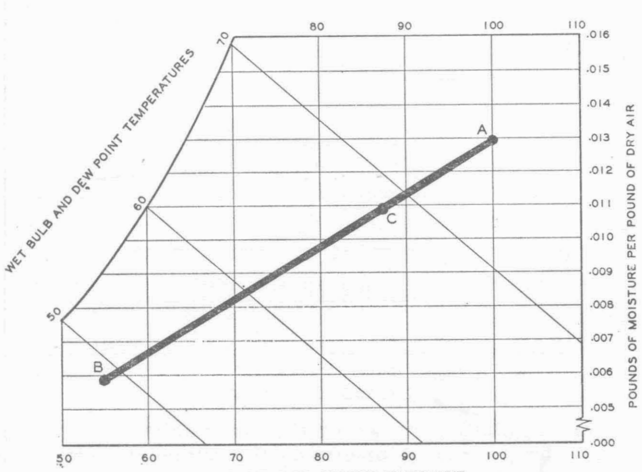

Figure 15. Illustration of properties of air resulting from the mixture of two air streams.

In the reheat system, the heat which must be added to the air for moisture equilibrium can become an excessive factor controlling the overall economy. The dry-bulb temperature of the saturated air leaving the conditioning unit must be increased approximately 10°-15° F. before it enters the grain. The approximate load in this case for heat alone would be 3.5 to 4.0 Btu per pound of dry air if all the heat was supplied by the heater. This would result in a minimum reheat load of about 0.825 Btu per hour per bushel for conditioned-air storages when air is supplied at a rate of 0.05 cfm per bushel. The quantity of heat entering the supply duct, however, could be used to reduce this load on the heater.

The return-air-bypass method has the advantage in most cases of reducing the power requirements to the system as compared to the reheat type system. This method depends upon the psychrometric principle of air mixtures. For example, consider the mixing of two air streams having properties at points A and B in Figure 15. The resultant properties of the mixture must lie on the line AB and may be located at point C. The location of point C depends upon the percentage of air supplied from each source. Point C would therefore lie half-way between points A and B if each air stream supplied 50 percent of the total mix. If only one-third of the total air was supplied from the stream at point B, then point C would lie one-third the distance from point A. The mixture point will always lie closest to the point representing the air that forms the largest percentage of the mixture.

After the grain has been cooled in conditioned-air storages, the air leaving the grain and hence the air entering the conditioning unit will approach the condition of the air entering the grain. The specific humidity of the air leaving the grain after the cooling period would be higher than the entering specific humidity if the entering air relative humidity was lower than the equilibrium relative humidity of the grain. If the relative humidity of the entering air was in equilibrium or higher than the grain equilibrium relative humidity, the specific humidity of the air leaving the grain would then be equal to or lower than the entering humidity. In any case, the properties of the entering and leaving air establish two states so that a straight line drawn between these states will pass through a region which can be used for maintaining the original grain moisture content with the bypass method.

During one test, the relative humidity of the air entering Bin 2 after a storage period of 28 days was 85 percent. The corresponding average grain temperature and moisture content was 63.5° F. and 11.77 percent, respectively. The equilibrium relative hu-

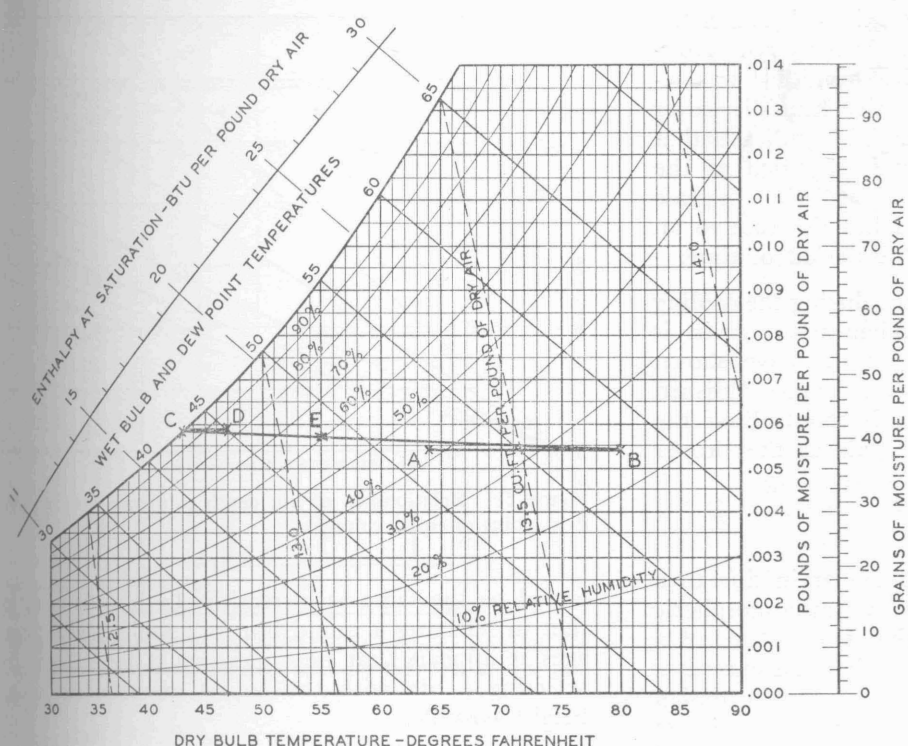

Figure 16. Psychrometric process of circulated air after 28 days of conditioned-air storage.

midity for grain under these conditions is 50.5 percent; therefore, the air was losing moisture to the grain. Under these conditions, the specific humidity would be increasing across the conditioning unit.

The complete process after 28 days of storage is shown in Figure 16. Point A represents the air conditions leaving the grain. This air was sensibly heated to point B as a result of the heat gain due to the fan and outside heat moving into the ducts. At point B the air entered the conditioning unit where the dry-bulb temperature was lowered, and the dew-point temperature was increased, point C. Point D represents additional heat gain between the conditioning unit and the grain.

Under these conditions, the relative humidity of the air entering the grain must be lowered in order to establish a relative humidity to maintain grain moisture equilibrium at approximately 13.1 percent. This relative humidity would be approximately 61 percent for a maximum grain temperature of 55° F., point E. Since point E lies on the air mixing line BC, the condition may be obtained through the return-air-bypass method by controlling the quantity of air passing through the conditioning unit or bypass duct. Point E may be selected at a lower dry-bulb temperature to take care of any heat gain in the supply ducts without affecting the relative humidity to any great extent. If point E does not fall in line BC, adjusting the solution circulating through the conditioning unit would cause point C to move along the saturation line to any desired point. This would allow point E to be on line BC, and proper grain moisture could still be controlled by the return-air-bypass method.

In all tests conducted, there was always a decrease in grain moisture content at the beginning of the cooling period regardless of the entering relative humidity. This was due to the cold entering air being at a lower dew-point temperature, or vapor pressure, than the grain at its initial temperature.

To illustrate, consider the effects on the grain moisture content when saturated air was supplied to Bin 1 for 104 days, Figure 17. A decrease in grain moisture content occurred during the cooling period even though the entering air was saturated.

Figure 17 also points out the wide variation in moisture contents during the periods of moisture

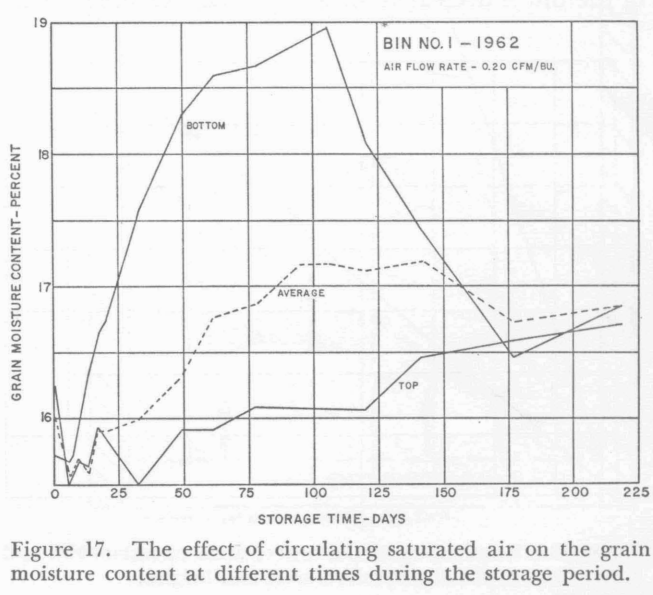

Figure 17. The effect of circulating saturated air on the grain moisture content at different times during the storage period.

transfer. An increasing gradient existed between the top and bottom up to approximately 104 days. However, when the supply-air relative humidity was reduced below the equilibrium relative humidity after 104 days, the gradients decreased. This indicates that when the entering air humidity is supplied at the equilibrium humidity the gradient will increase during the initial cooling period but decrease to an insignificant value in a reasonable period of time.

The moisture loss which occurs during the cooling period is not considered a problem when grain is stored for a reasonable time. Results have shown that the average moisture content can be re-established after the grain has been cooled by controlling the relative humidity of the entering air at the proper level.

Heat Gain

The heat which moves into the grain due to a temperature differential between warm atmospheric air and cool grain may cause the temperature of a layer of grain next to the wall to become excessive. Any increase in grain temperature above the design temperature may result in serious damage from insect infestation. To maintain temperatures at the desired level in the outer layers of grain, some provision must be made to reduce the amount of heat transferred through the walls of a structure.

One way to accomplish this is to reduce the exposed surface. This can be done by the proper selection of the storage structure. For instance, consider the surface area of a rectangular bin (length twice the width) compared to a cylindrical bin of equal volume, Figure 18. For the same volume and height, less surface area is exposed in a round bin than a rectangular bin with the given configuration. Surface area can further be minimized by reducing the height of a round bin. Figure 19 shows that the exposed surface area can be materially decreased as the height of the bin is decreased for any constant volume. This

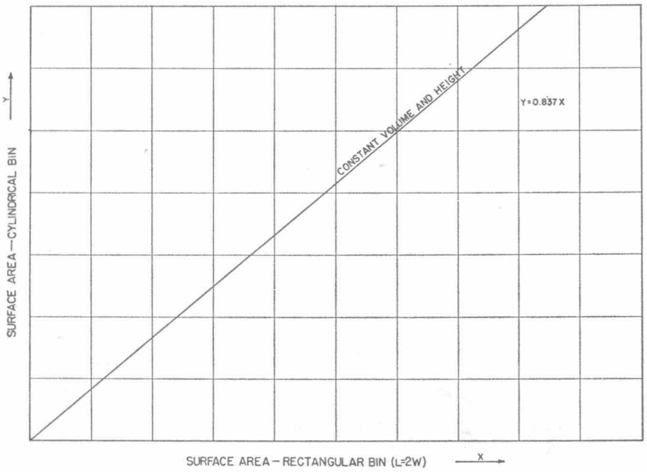

Figure 18. Relationship of the surface areas of rectangular and cylindrical bins having equal volumes and heights.

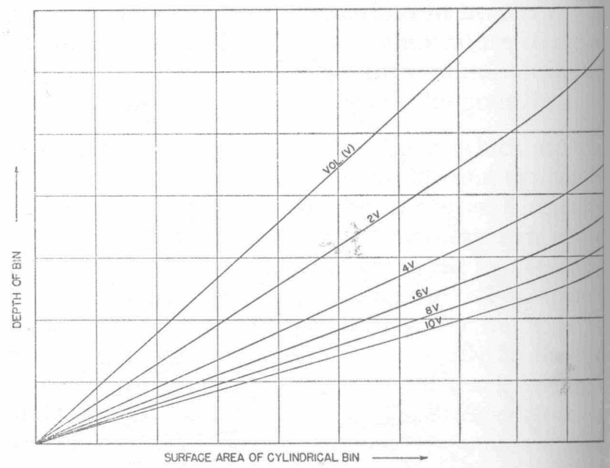

Figure 19. Relationship of bin depth to surface area of cylindrical bins of equal volumes.

would be the same as increasing the bin diameter so that in selecting a round bin for conditioned-air storages it appears that heat gain could be reduced by selecting a bin with the largest diameter or lowest height possible.

Another way of reducing the heat transfer into the bin is to improve the insulating quality of the walls with building insulation. The thickness of insulation required will depend on the type of insulating material used, the bin construction and the temperature difference between the outside and inside of the bin wall.

Regardless of the thickness of insulation, the circulating air inside the grain bin must still be used to remove the heat which moves in. Increasing the thickness will, however, reduce the rate at which this heat can be transferred to the inside. Since the air moving through the grain must be used to move the heat out, some increase in grain temperature will always result. This increase in grain temperature will exist as a temperature gradient within the grain mass in both the horizontal and vertical planes similar to those shown in Figure 20. The temperature pattern in this installation resulted from the heat gain across one inch of epoxy foam insulation. The airflow rate was 0.2 cfm per bushel.

Until further tests can be conducted on the influence of temperatures on stored-grain insects, 60° F. should be considered the maximum temperature for effective control of insects. By limiting this temperature to 60° F., the heat which the air could remove can be calculated from

$$q_a = W_a C_{p(a)} (T_f - T_i) \qquad (18)$$

where

q_a = heat air can remove which was transferred into bin from outside, Btu per hr.

W_a = airflow rate available for removing heat transferred from outside, lbs. dry air per hr.

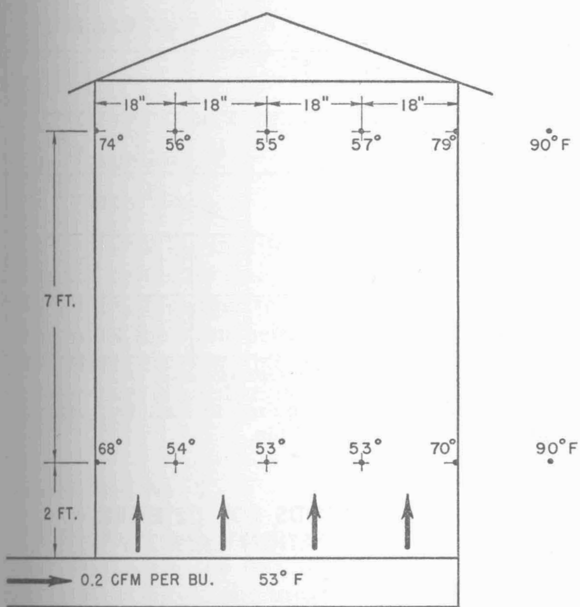

Figure 20. Typical horizontal and vertical temperature gradients within a grain mass resulting from heat moving from outside into storage bin.

$C_{p(a)}$ = specific heat of air, Btu per lb. dry air-°F.

T_f = final average air temperature, °F.

T_i = initial air temperature, °F.

Several assumptions must be made before Equation (18) can be used directly. It must be assumed that the total heat which moves from outside the bin into the grain can be accurately calculated from a mean wall temperature. Also, it must be assumed that this mean wall temperature is a true average between the wall temperatures at the bottom and top of the bin wall and that the temperature lag within the grain mass would be such that the average outside temperature over a 24-hour period would be appropriate for design purposes. Another assumption is that the final dry-bulb temperature of the air leaving the grain mass can be assumed to equal the average temperature over a distance Y, which is the horizontal distance at the top of the grain mass over which the heat will be allowed to penetrate.

The total heat which the conditioned air could remove can then be estimated from Equation (18) by selecting a maximum wall temperature and a distance Y over which the temperature gradient would be allowed to exist. This Y distance would actually establish the quantity of air (W_a) which would be available to remove the heat.

Consider the following example which is illustrated in Figure 21:

Bin size – 18 ft. diameter, 20 ft. high

Bin capacity – 4000 bu.

Airflow rate – 0.1 cfm per bu.

Specific volume of entering air – 13.0 cu. ft. per lb. dry air

Maximum wall temperature = 60° F.
$T_a = 80°$ F.
$T_i = 55°$ F.
$Y = 9$ ft.
Surface area (A) = 1130 sq. ft.
$T_f = 57.5°$ F.
$C_{p(a)} = 0.24$ Btu per lb. of dry air °F.

$$W_a = \frac{\frac{\pi D^2}{4} - \frac{\pi(D-2Y)^2}{4}}{\frac{\pi D^2}{4}} \times \frac{(Q)(C)(60)}{\text{Sp. Vol.}} \qquad (19)$$

where

D = bin diameter, ft.
Y = distance which heat is allowed to move into grain mass, ft.
Q = design airflow rate, cfm per bu.
C = bin capacity, bu.
Sp. Vol. = specific volume of entering air, cu. ft. per lb. dry air

$$W_a = \frac{\frac{\pi(18)^2}{4} - \frac{\pi[18-2(9)]^2}{4}}{\frac{\pi(18)^2}{4}} \times \frac{(0.1)(4000)(60)}{13.0}$$

$= 1846$ lbs. dry air per hr.

$q_a = (1846)(0.24)(57.5-55)$
$= 1108$ Btu per hr.

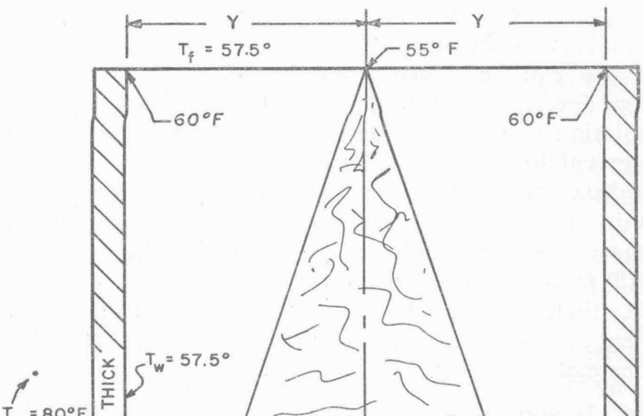

Figure 21. Cross-sectional view of storage bin used to illustrate method to calculate heat gain and insulation thickness.

Under normal atmospheric conditions during the summer months, W_a would not be large enough to prevent excessive grain temperatures without the addition of insulation. With an average outside temperature of T_a and an inside mean wall temperature of T_m, the necessary thickness of insulation would be as follows, if the resistance of the bin wall and the inside thermal conductance is assumed to be negligible.

$$X = K \left[\frac{A\,(T_a - T_m)}{q_a} - \frac{1}{f_o} \right] \qquad (20)$$

where

- X = insulation thickness, in.
- K = insulation thermal conductivity, Btu-in. per hr.-sq. ft.-°F.
- A = surface area, sq. ft.
- T_a = average outside air temperature, °F.
- T_m = mean wall temperature, °F.
- q_a = heat transferred into bin, Btu per hr.
- f_o = outside thermal conductance, Btu per hr.-sq. ft.-°F.

The thickness of epoxy foam insulation in the above illustration would then be:

$$X = 0.17 \left[\frac{1130(80-57.5)}{1108} - \frac{1}{4} \right]$$

X = 4 inches

It can be shown from Equations (18) and (20) that for equal volumes of grain the thickness of insulation to minimize heat gain can be reduced as the bin height decreases whenever Y is equal to the bin radius. If Y is less than the bin radius, this relationship does not hold true. As long as Y is a maximum, the expression which contains D and Y in Equation (19) is unity. However, once this expression becomes less than unity the quantity of air available to remove heat decreases faster than the exposed surface area with decreased bin heights.

It appears from test data that some method in addition to insulation may have to be used to limit grain temperatures to a safe level in small bins. One proposed method would be to use a double-wall storage structure and circulate cold air between the walls. The outside wall could then have a practical thickness of insulation installed on it since the volume of air necessary to remove the heat transferred from the outside would not be limited by high fan power requirements as it would be inside the grain bin. Air could then be entered into the double wall at approximately 55° F. and exhausted at 60° F. in order to maintain the inside wall temperatures at or below 60° F. The pounds of dry air circulating in the double wall per hour per square foot to remove this heat gain without excessive wall temperatures can be determined from Figure 22.

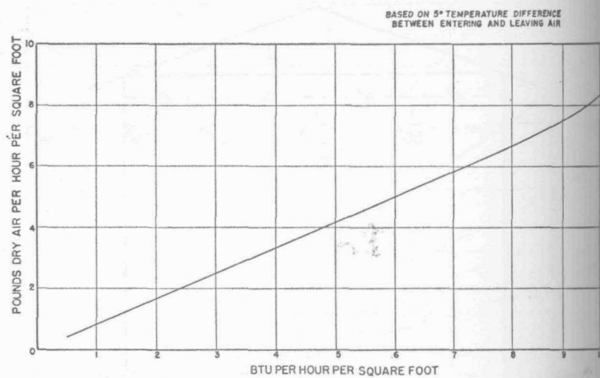

Figure 22. Airflow rate required to limit heat transfer into grain stored in a double-wall bin.

DESIGN METHODS FOR DETERMINING REFRIGERATION CAPACITY

Design Method No. 1

The design method most familiar to engineers is the use of Equation (13), which stated:

$$Q_G = W\,C_p\,\Delta T$$

This equation considers sensible heat exchange only and assumes that no latent and biological heat exchange takes place during the cooling process. Therefore, when this equation is used, it must be assumed that there is no exchange in moisture from the grain to the air and no significant load due to heat of respiration during the cooling process.

The following equation takes into account the moisture transferred from the grain to the air and the heat of respiration and was found to be reliable for calculating refrigeration requirements for cooling grain:

$$Q_G = [W_{dg}\,C_{p_{(dg)}}\,(t_i - t_f)] + [h_{f_{(i)}}\,W_{w_{(i)}} - h_{f_{(f)}}\,Ww_{(f)}] + H_R \qquad (21)$$

where:

- Q_G = grain load, Btu.
- W_{dg} = initial weight of dry grain, lbs.
- $C_{p_{(dg)}}$ = specific heat of dry grain, Btu per (lb.) (°F.).
- t_i = initial temperature of grain, °F.
- t_f = final temperature of grain, °F.
- $h_{f_{(i)}}$ = enthalpy of water at initial grain temperature, Btu per lb.
- $h_{f_{(f)}}$ = enthalpy of water at final grain temperature.
- $W_{w_{(i)}}$ = initial weight of water in grain, lbs.
- $W_{w_{(f)}}$ = final weight of water in grain, lbs.
- H_R = heat of respiration, Btu.

The refrigeration capacity required to cool grain based on the above equation may be expressed as follows:

$$Q_R = \frac{Q_G}{12000\ T_c} \qquad (22)$$

where

Q_R = refrigeration capacity, tons

Q_G = grain load, Btu

T_c = time, hours

The value of Q_R in Equation (22) represents the average load on the conditioning unit. Therefore, in actual practice, the time required to cool the grain will not equal the value selected for T_c but will require a longer period of time. At present, there is no accurate method of predicting cooling time when this method is used.

Design Method No. 2

A design method which simplifies load calculations is based upon the fact that the air exhausting from the grain during the initial stages of cooling is in approximate equilibrium with the grain at its initial temperature and moisture content when the cooling air is supplied at rates normally used for aeration. Knowing the initial condition of the grain and the heat evolved due to respiration, the conditions of the exhaust air can be calculated. Research has shown that respiration heat values for sorghum grain at moisture contents of 14.6 percent or lower and temperatures as high as 100° F. are small compared to the total heat load and therefore can be neglected (19). However, for higher moisture contents, the respiration heat was found to be very significant due to the tremendous increase in respiration and should be considered in the cooling load requirements. In a recirculating system, the exhaust air would represent the conditions of the air entering the conditioning unit assuming no heat gain into the air duct. The required condition of the air leaving the conditioning unit can be determined by knowing the necessary air conditions entering the grain. Thus, the enthalpy difference of the air entering and leaving the conditioning unit can easily be determined. When the rate at which air is supplied through the grain is established, the capacity of the refrigeration equipment can be determined.

The diagram in Figure 23 illustrates the relationship of the enthalpy of the air entering and leaving the conditioning unit to cooling time when air is recirculated through the grain. The enthalpy difference, (h_2-h_1), represents the load on the conditioning unit in Btu per pound of dry air and includes the reheat load required to reduce the relative humidity of the air leaving the conditioner to the design level.

It should be pointed out that the refrigeration capacity obtained by this method is based on the maximum load which is the initial difference in exhaust and entering-air enthalpies. After the leading edge of the cooling zone passes through the grain, the enthalpy of air exhausted from the grain starts to decrease and continues to decrease until it reaches a minimum value at the end of the cooling period.

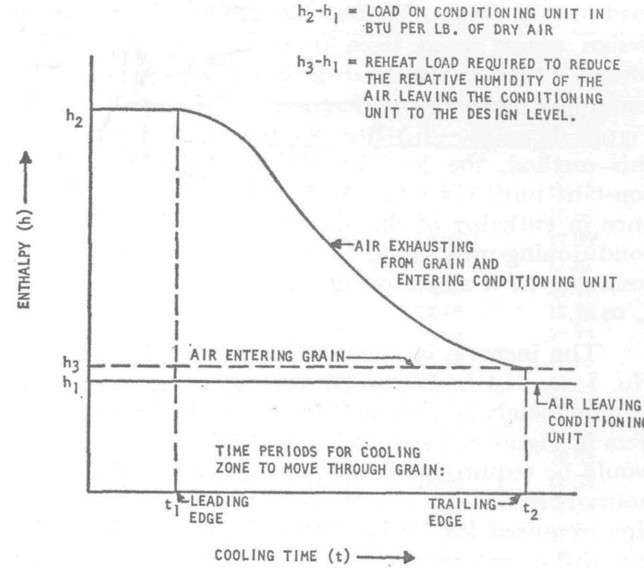

Figure 23. Diagram illustrating the relationship of the enthalpy of air entering and leaving the conditioning unit to the cooling time when air is recirculated through the grain.

Design Method No. 3

Theoretical consideration will be given to a proposed method of design that may minimize the overdesign problem encountered in Design Method No. 2. In this method, air is supplied at twice the rate used in Design Method No. 2, and the design load is based on one-half the initial difference in enthalpy of the exhaust air and the entering air. The total load on the conditioning unit is the same for both methods. However, as shown in Figure 8, the cooling time is not exactly inversely proportional to airflow rate, and, for this reason, the cooling time for Design Method No. 3 is slightly longer than the time required for Design Method No. 2.

A comparison of these two design methods is illustrated in Figure 24. The enthalpy difference, (h_2-h_1), represents the load in Btu per pound of dry air when the design is based on the initial maximum

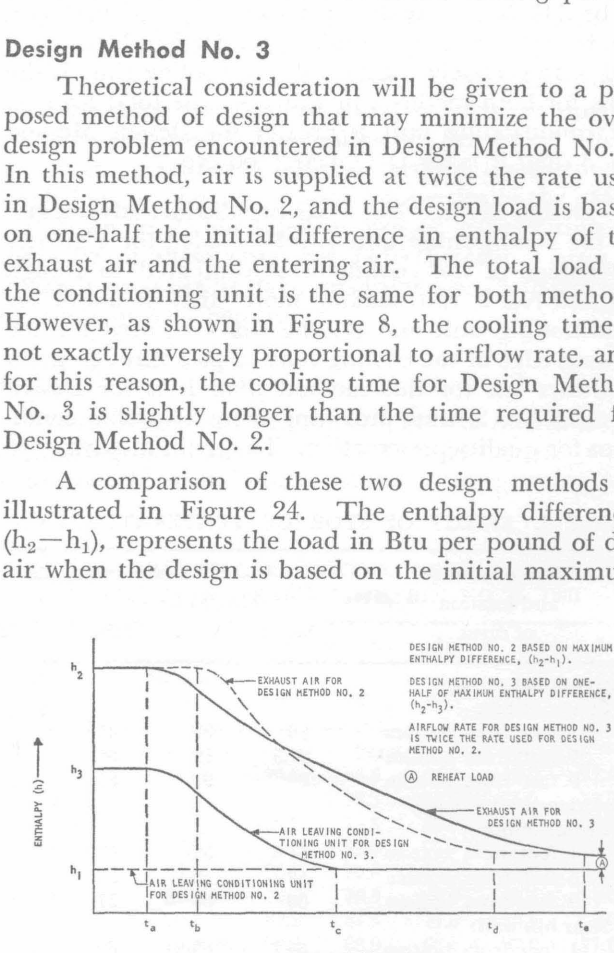

Figure 24. Enthalpy of recirculated air versus cooling time for design methods 2 and 3.

load used in Design Method No. 2. In this case, overdesign would occur from time t_b to t_d. In Design Method No. 3, the load is based on one-half the maximum enthalpy difference. This is shown in Figure 24 as $(h_2 - h_3)$ Btu per pound of dry air. In this method, the load on the conditioning unit is constant until time t_c. After this period, the difference in enthalpy of the air entering and leaving the conditioning unit would be less than the design value, resulting in a slight overdesign situation from time t_c to t_e.

The increase in cooling time for Design Method No. 3, $(t_e - t_d)$ in Figure 24, represents an increase of approximately 20 percent. For example, based on the data in Figure 8, an airflow rate of 0.10 cfm per bushel would be required if it is desired to cool grain in 145 hours based on Design Method No. 2. The cooling time required for Design Method No. 3, using twice the airflow rate, or 0.20 cfm per bushel, would then be 174 hours.

If it is desired to cool the grain in the same time period required for Design Method No. 2, an airflow rate could be selected from Figure 8 to correspond to a cooling time equal to one-half the desired total cooling time. If the cooling time is again assumed to be 145 hours, airflow rates of 0.10 and 0.25 cfm per bushel would be required for Design Methods 2 and 3, respectively. Although the cooling time is the same for both methods in this case, the total load on the conditioning unit is greater for Design Method No. 3 than it is for Design Method No. 2.

Design Method No. 3 has two distinct advantages: (1) the conditioning unit is fully loaded for a greater percentage of the cooling time compared to Design Method No. 2, thus making more efficient use of the conditioning unit and (2) the time required for the leading edge of the cooling zone to pass through grain is always less for this method than it is for Design Method No. 2, thus providing more favorable conditions for quality preservation. The main disadvantage to Design Method No. 3 is that higher airflow rates are required which would, in turn, increase the static pressure requirements for the system and necessitate an increase in the size of the fan and motor.

EFFECTS OF CONDITIONED-AIR STORAGE ON GRAIN QUALITY

Insect Control

Adult rice weevils in brass, cylindrical cages were placed at various locations in Bins 1 and 2 to determine the effects of conditioned-air storage facilities and natural aeration systems on insect control. Each cage was 1 inch in diameter and 5¼ inches long. Fifteen pairs of adult rice weevils and 37 grams of 14.5 percent-moisture-sorghum grain were placed in each cage. The depth of grain in each bin was 10 feet. Cages were placed in the center of Bin 1 at the following locations: 1, 5, 9-foot levels and at the center near the grain surface. Two cages, one near the wall and one at the center, were placed in Bin 2 at the following depths: 1, 5 and 9 feet.

All insect cages remained in the test bins under storage temperatures given in Table 3 for 35 days. Then the cages were removed, and the insects counted. The weevils were removed, and the test cages replaced at the same locations in the bins to determine the reproductive ability of these insects under each storage condition. Cages were removed after 25 days and the progeny counted.

Results of the storage conditions on insect activity are given in Table 3. Rice weevil infestation does not appear to be a storage problem in conditioned-air storage structures under the temperatures of these tests. The percent mortality of insects exposed to the high wall temperatures was 86.3 percent after the first 35 days. The insects which were located in the cooler center region of the bin had a mortality of 98.8 percent. The insects which were exposed to the center temperatures in the natural aeration bin had a mortality of 61.8 percent during the same 35-day period.

TABLE 3. EFFECT OF STORAGE TEMPERATURES ON RICE WEEVIL INFESTATION IN SORGHUM GRAIN

Type of storage and location of cages	Temperature, degrees Fahrenheit						Number of insects at end of storage			
	First 35 days			Additional 25 days			First 35 days		Additional 25 days	
	Mean	Max.	Min.	Mean	Max.	Min.	Live	Dead	Live	Dead
Conventional storage										
Center of bin:										
1 foot from bottom	86	89	83	78	88	66	11	19	29	4
5 feet from bottom	87.5	89	86	80	89	71	15	15	69	6
9 feet from bottom	89	91	87	82	91	93	8	21	33	0
Conditioned air storage										
Center of bin:										
1 foot from bottom	51.5	54	49	51	53	49	0	29	0	0
5 feet from bottom	57	59	55	56	58	54	0	28	0	0
9 feet from bottom	60	63	57	61	60	59	1	27	0	0
Near bin wall:										
1 foot from bottom	65.7	69.5	62	65	70	60	0	29	0	0
5 feet from bottom	71.5	75	68	69	73	65	3	17	0	0
9 feet from bottom	77.2	81.5	73	72	75	69	8	23	0	0

TABLE 4. RELATIONSHIP OF AERATION TIME TO MOISTURE CONTENT OF GRAIN[1]

Aeration time days	Moisture content — Percent									
	Grain depth — Feet									
	1	2	3	4	5	6	7	8	9	Average
Beginning of test	17.45	17.65	17.75	18.50	18.30	18.15	18.30	18.70	18.90	18.19
1	17.35	17.70	17.90	19.10	18.80	18.45	19.10	19.50	19.85	18.64
2	17.23	17.18	17.25	18.20	18.05	17.80	18.40	18.75	19.05	17.99
3	16.92	17.14	17.29	18.00	17.75	17.45	17.75	17.85	18.30	17.61
5	17.20	17.16	17.42	18.05	17.63	17.24	17.14	17.75	17.75	17.48
6	16.98	16.99	17.42	17.70	17.52	17.16	17.24	17.60	17.48	17.34
7	17.30	17.08	17.22	17.80	17.78	17.27	17.47	17.55	17.62	17.45
8	17.07	17.18	17.37	17.90	17.51	17.07	17.22	17.30	17.56	17.35
12	17.10	17.15	17.32	17.65	17.51	17.16	17.40	17.35	17.42	17.34
19	16.96	17.14	17.37	17.60	17.41	17.10	17.20	17.19	17.33	17.26
33	16.55	16.99	17.13	17.61	17.10	17.09	17.08	16.95	17.03	17.06
47	16.00	16.51	16.90	17.19	16.78	16.65	16.75	16.64	16.75	16.68
69	15.71	16.05	16.60	17.01	16.62	16.91	16.84	16.45	16.60	16.53
85	15.62	15.75	16.42	16.70	16.82	16.82	16.54	16.54	16.15	16.36
114	15.22	15.45	16.15	16.51	16.00	16.74	16.51	16.51	15.82	16.10
148	15.55	15.09	15.32	15.65	15.88	16.52	16.11	16.26	15.32	15.74
194	15.64	15.45	15.99	15.66	15.78	15.79	15.73	16.17	15.50	15.75

[1]Airflow rate: 0.12 cfm per bushel.

At the end of the second storage period of 25 days, no emergence occurred in the conditioned-air storage bin. However, in the bin aerated with natural air, an emergence of 158.4 percent was observed based on the original number of insects which were first placed in the cages. Of this number, only 7.1 percent were found dead after the additional 25 days.

These results do not necessarily compare with published data concerning the temperature limitations of rice weevils. The mean wall temperatures in the conditioned-air bin were as high as 77.2° F. with a maximum of 81.5° F. These temperatures would normally support insect activity in natural aeration systems. The high mortality and zero percent emergence in the conditioned-air bin may have resulted from the procedures used. It was not possible to place the insects in the test bins at the start of the cooling period. Consequently, the insects were subjected to a rapid temperature change when they were placed in the bin. The same temperature conditions existed from the beginning of the test as well as during the test period. Additional tests need to be conducted to determine if this rapid change in temperature influenced the test results.

Germination

In order to determine the effects of conditioned-air storage procedures on the germinating qualities of grain, specific tests were conducted on high-moisture grain. If it can be shown that no significant loss in germination occurs in high-moisture grain, it can be assumed that low-moisture grain can be stored under similar conditions without any germ damage. Grain having an initial moisture content of 18.19 percent was conditioned with air having a dry-bulb temperature of 45° F. for 194 days. The relative humidity of the entering air was approximately 73 percent for about 50 days after which time it gradually decreased to approximately 65 percent at the end of 194 days. The relative humidity was allowed to decrease so that the grain moisture content could be reduced during the storage period. The results of this drying effect are given in Table 4. The effects of these storage conditions on the germination properties of the grain are shown in Table 5. The average germination at the start of these tests was 77.95 percent. After a storage period of 194 days, this value was 74.32 percent or a loss of 3.63 percent.

Mold

Since molds will develop at a greater rate in high-moisture grain than low-moisture, mold analyses were made on the same stored grain as the germination

TABLE 5. PERCENT GERMINATION OF STORED GRAIN AS RELATED TO STORAGE TIME, BIN NO. 1, 1961

Depth of grain, feet	Storage time — days					
	Start of test	8	47	85	114	194
	Percent					
1	78.0	75.0	68.0	64.5	73.0	77.0
2	77.0	79.7	76.5	75.0	75.0	72.0
3	77.7	75.0	73.5	69.0	73.0	72.0
4	76.0	76.0	59.0	75.0	76.0	76.0
5	74.5	76.7	70.0	73.0	74.0	70.0
6	84.2	79.0	76.5	69.0	83.5	74.0
7	78.7	82.2	84.5	77.0	83.5	79.0
8	76.2	78.2	68.0	66.5	82.0	77.0
9	79.3	79.0	73.0	70.5	74.0	72.0
Average	77.95	77.84	72.11	71.05	77.11	74.32

TABLE 6. MOLD COUNTS FROM SORGHUM SEED STORED IN TEST BIN, 1961-62

Depth of grain, feet	Aeration time, days	Kernels from which molds were isolated, percent	Percent kernels infested with: Field molds	Storage molds
1	Beginning	100	100	0
2	of test	100	99	2
3		99	99	0
4		99	99	1
5		100	100	1
6		100	100	1
7		100	100	1
8		100	100	0
9		97	97	1
1	85	100	100	0
2	85	100	100	0
3	85	100	100	0
4	85	100	99	1
5	85	100	100	0
6	85	100	100	0
7	85	100	100	0
8	85	100	100	2
9	85	100	100	4
1	194	100	100	1
2	194	100	100	1
3	194	100	100	0
4	194	100	100	1
5	194	100	100	2
6	194	100	100	0
7	194	100	100	4
8	194	100	100	0
9	194	100	99	4

tests. Results of these tests are shown in Table 6. Qualitatively, the mold flora were the same after 194 days of storage as at the start, which is an indication that the storage conditions maintained were unfavorable for storage mold growth. No adverse effects are anticipated from the high infestation of field fungi since field fungi do not seem to be associated with deterioration of sorghum seed during storage (2).

Grain Moisture Content

Results of all tests to date show that the grain moisture content can be controlled with conditioned-air storage procedures. Even though test results prove that a reduction in grain moisture is inherent during the cooling period in this type of storage, moisture can be transferred back to the grain by controlling the relative humidity of the interstice air. Although no difficulty was encountered by adding moisture to stored grain in these tests, research has indicated that there may be some limitations in this practice due to excessive bin wall pressures resulting from large increases in grain moisture contest (20).

Since the quantity of water removed from the grain depends upon the relative humidity of the entering air, it should be pointed out that this type of storage can be used to reduce the moisture content of stored grain as well as maintain the original moisture level. If it is desirable only to maintain the original moisture content, then the grain should remain in storage for a period of time sufficient to return the moisture back to the grain. It is not possible to predict this time because of the variables involved. The airflow rate, grain moisture content, relative humidity of the entering air are some of these variables. Normally the quantity of water which must be returned to the grain after the cooling period in order to re-establish the original moisture content would be approximately the quantity required to increase the moisture by one-half of one percent on a wet basis.

DESIGN PROCEDURE

To illustrate the procedure to be used in the design of a conditioned-air storage system, consider the following problem and its solution.

Problem: To design a controlled environment system at College Station, Texas, for storing 288,000 bushels of sorghum grain. The storage facility consists of eight steel bins having a capacity of 36,000 bushels each. These bins are 32 feet in diameter and 56 feet high. The grain will be received with an initial temperature and moisture content of 95° F. and 15 percent, respectively. It is required that the grain mass be cooled to 55° F. with a final grain moisture content of 14 percent. It is desired that the grain be cooled to the final temperature in a maximum of 25 days after filling. The estimated filling rate is one bin per day. Each bin has an aeration system capable of supplying air at a rate of 0.1 cfm per bushel at present.

Cooling Procedure

When more than one storage bin is involved in a system design, a cooling procedure must be developed to select the most economical number and size of conditioning units. This procedure must take into account the number of bins, filling rate of each bin and the maximum time which grain can be stored in these bins before it should be cooled.

In the example problem, the number of days required to fill the bins is 8. This would mean that if one conditioning unit was used the cooling time would have to be 4 days for each bin in order to cool the last bin filled in 25 days. The airflow rate for a 4-day cooling period would be approximately 0.17 cfm per bushel, Figure 8. Since this rate is in excess of

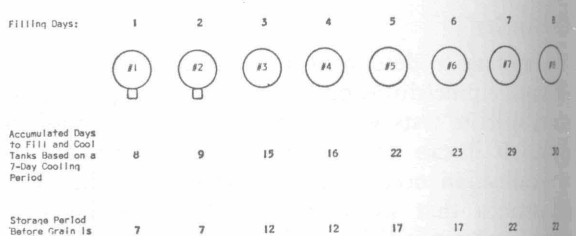

Figure 25. Filling and cooling procedure for grain stored in eight bins and aerated with two conditioning units.

the design rate of 0.1 cfm per bushel, it would be more economical to use two units. In this case the cooling procedure would be as shown in Figure 25.

Design Conditions

The conditions on which the design must be based follow:

GRAIN:

1. Initial temperature = 95° F.
2. Initial moisture content = 15%

OUTSIDE AIR: (based on design maximums for July at College Station, Texas)

1. Dry bulb = 95° F.
2. Wet bulb = 80° F.
3. Dew point = 75° F.
4. Relative humidity = 52%
5. Enthalpy = 43.6 Btu per lb. dry air
6. Specific volume = 14.4 cu. ft. per lb. dry air

CONDITIONED AIR:

Leaving grain

1. Dry bulb = 95° F.
2. Wet bulb = 91° F.
3. Dew point = 90° F.
4. Relative humidity = 85%
5. Enthalpy = 57.1 Btu per lb. dry air
6. Specific volume = 14.7 cu. ft. per lb. dry air

The above conditions leaving the grain are based on the fact that the air conditions leaving the mass will be in equilibrium with the grain.

Entering grain

1. Dry bulb = 55° F.
2. Wet bulb = 50° F.
3. Dew point = 45° F.
4. Relative humidity = 70%
5. Enthalpy = 20.2 Btu per lb. dry air
6. Specific volume = 13.1 cu. ft. per lb. dry air

The dry-bulb temperature of the air entering the grain was selected at the same level as the desired final grain temperature. The relative humidity of this air was selected from Figure 13 so that it would be in equilibrium with 14 percent moisture grain at 55° F.

Leaving conditioner

1. Dry bulb = 45° F.
2. Wet bulb = 45° F.
3. Dew point = 45° F.
4. Relative humidity = 100%
5. Enthalpy = 17.62 Btu per lb. dry air
6. Specific volume = 12.84 cu. ft. per lb. dry air

These conditions assume that the air leaving the conditioner will be saturated. A dew-point temperature of 45° F. was selected so that this air can be heated sensibly to 55° F. and have a relative humidity of 70 percent as required by the entering air conditions to the grain. Research has indicated that no supplemental heat is necessary since normal heat gain will be approximately 5°-10° F., the amount needed to maintain the entering air conditions to the grain. In case some supplemental heat is needed in a particular design, heaters may be installed or some provision may be made to use the heat exhausted from the condenser coils of the conditioning unit.

Airflow Requirements

Airflow rate = 0.1 cfm/bu.

This value was selected from the available airflow in the existing aeration system and from Figure 8.

Total flow = (0.1) (36000) = 3600 cfm

$$\text{Mass flow rate} = \frac{(3600)\,(60)}{13.1} = 16{,}488 \text{ lbs. dry air per hr.}$$

Refrigeration Capacity

If the design was based on a re-circulating air system, the change in the enthalpy of the air, Δh, as it passes through the grain mass would be equal to $57.10 - 17.62 = 39.48$ Btu per pound dry air. The resulting refrigeration capacity per unit would then be

$$\frac{(16{,}488)\,(39.48)}{12{,}000} = 54.2 \text{ tons}$$

If the design was based on circulating outside air through the conditioning unit then

$$\Delta h = 43.60 - 17.62 = 25.98 \text{ Btu per lb. dry air}$$

or

$$\frac{(16{,}488)\,(25.98)}{12{,}000} = 35.7 \text{ tons}$$

It will then be more economical to condition outside air until the enthalpy of the air leaving the grain mass is less than that of the outside air. This period can be determined by the time required for the leading edge of the zone to move through the mass, Figure 8. After approximately 40 hours, the system can be operated as a re-circulating system.

It should be noted that in this design method, no consideration need be given to the heat gain due to outside air conditions since the grain is at the same dry-bulb temperature as the outside air. By basing the design load on either the outside air conditions or the air leaving the grain mass, the refrigeration capacity would be large enough to take care of heat gain as the grain cools.

The refrigeration capacity for the design problem was calculated to be 35.7 tons. However, two 35-ton units could probably be selected because some evaporative cooling will occur.

Insulation Requirements

If the maximum wall temperature is 60° F., the average outside air temperature is 85° F. and the temperature gradient in the horizontal plane is allowed to extend over the full radius of the bin, the thickness of insulation could be calculated from Equations 18, 19 and 20. Using Equation 19, the quantity of air available to remove the heat transmitted from outside the bin would be as follows:

$$W_a = \frac{\frac{\pi D^2}{4} - \frac{\pi(D-2Y)^2}{4}}{\frac{\pi D^2}{4}} \times \frac{(Q)\,(C)\,(60)}{\text{Sp. Vol.}}$$

$$= \frac{\frac{\pi(32)^2}{4} - \frac{\pi(32-32)^2}{4}}{\frac{\pi(32)^2}{4}} \times \frac{(0.1)\,(36000)\,(60)}{13.1}$$

$$= 16{,}488 \text{ lbs. dry air per hr.}$$

If the final average air temperature exhausting from the grain is 57.5° F., then from equation 18, the heat which can be removed by the air is:

$$q_a = W_a\, C_{p(a)}\, (T_f - T_i)$$
$$= (16488)\,(.24)\,(57.5-55)$$
$$= 9893 \text{ Btu per hr.}$$

Select an insulation such as epoxy or polyurethane spray-foam insulation. If the mean wall temperature is 57.5° F., then by equation 20, the thickness of this type of insulation would be:

$$X = K \left[\frac{A(T_a - T_m)}{q_a} - \frac{1}{f_o} \right]$$

$$X = .17 \left[\frac{5628\,(85-57.5)}{9893} - .17 \right]$$

$$= 2.63 \text{ or } 3''$$

The solution to the example problem would be to install two 35-ton units on the first bins loaded. With an airflow rate of 0.1 cfm per bushel, these units could be moved to the other bins every 7 days until all bins were cooled. Outside air should be conditioned for approximately 40 hours, after which the system should be connected so that the air leaving the grain mass can be re-circulated through the conditioning unit. The units will have to be operated periodically on each bin after all of the grain is cooled in order to prevent excessive increase in grain temperatures due to heat gain and to reduce the grain moisture content to 14 percent.

This solution appears to be the most practical at present. This would, however, depend on the cooling procedure. If only one bin is used as a conditioned-air storage system, the refrigeration capacity selected from this design method would be based on maximum conditions. This would mean that as soon as the leading edge of the zone has moved out of the grain mass, the refrigeration equipment would not be fully

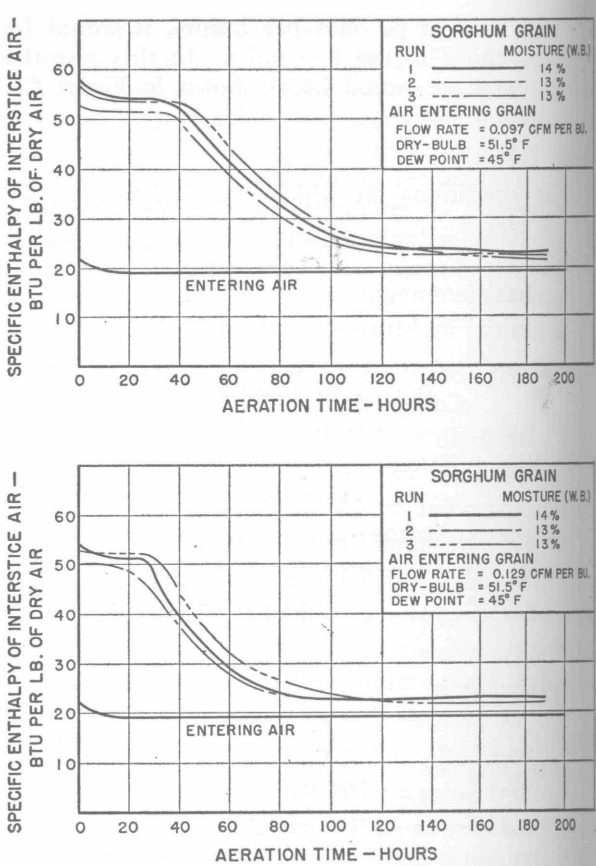

Figure 26. Specific enthalpy of interstice air in relation to aeration time when cooling a mass of grain in storage.

loaded. The load would gradually decrease with time and reach a minimum at the end of the cooling period. This relationship is shown in Figure 26 by the difference in the leaving and entering-air enthalpies.

The solution of Equation (13) would yield a close approximation of the total load on the conditioning unit. The normal procedure would be to divide this total load by the cooling time in order to obtain an average load. However, the system would be overloaded during the initial cooling period and over designed by the same amount at the end of the cooling period. Preliminary tests indicate that this method may be practical for reducing the initial refrigeration unit size. Additional tests need to be conducted to determine the effect of this design method on the actual cooling time.

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