# NOCTURNAL COOLING OF A SOLAR COLLECTOR-STORAGE UNIT

by

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TABLE OF CONTENTS

LIST OF TABLES

LIST OF TABLES. continued

## LIST OF SYMBOLS

$A$, $A_s$: surface area for convective heat transfer, $ft^2$
$b$: distance separating parallel plates
$C$: flow nozzle coefficient of discharge
$c_p$: specific heat, Btu/lb-F
$D$: duct diameter
$d$: flow nozzle throat diameter
$d_e$: hydraulic diameter
$F$: flow nozzle velocity approach factor
$F_a$: flow nozzle thermal expansion factor
$G$: mass velocity, pounds per square foot per hour
$g$: gravitational constant
$h$: convective heat transfer coefficient
$h_e$: enthalpy
$h_w$: pressure drop across flow nozzle
$k$: thermal conductivity
$L$: passage length
$L_e$: development length for flow in duct
mv: millivolt output of thermocouples
Nu: Nusselt number
Pe: Peclet number
Pr: Prandtl number
$p_v$: velocity pressure
$Q$: heat transferred to the air from the storage
Re: Reynolds number
$T$: storage temperature
$T_o$: initial temperature of storage
$T_\infty$: average air temperature in system
TA: ambient temperature
TC: temperature of air entering system
TF: temperature front surface of storage
TIN: air temperature at bottom of space between inside cover and storage wall
TT: air temperature entering space between inside cover and storage wall
T1: air temperature entering passages through storage wall
$\Delta t$: temperature difference between storage and ventilating air
$V$: volume of storage, cubic feet
$V_o$: air velocity
$v$: specific volume, cubic feet/pound
$W_h$: weight rate of flow, pounds/hour
$x$: distance from duct entrance to point of interest
$Y_a$: expansion factor for flow nozzle
$\beta$: ratio of throat diameter to duct diameter for flow nozzles
$\gamma$: specific weight at inlet side of flow nozzle, $lb/ft^3$
$\mu$: dynamic viscosity, lb/ft-hr
$\theta$: time
$\rho$: air density, $lb/ft^3$

# INTRODUCTION

As agriculture develops into an increasingly complex industry, new problems are encountered, perhaps created by solutions to other problems. One such problem is maintaining a comfortable environment inside swine confinement shelters in a time of increasing fuel and energy costs. Heat stress has long been recognized in swine as a cause for poor performance, even deaths, of animals during warm summer months. Traditionally, swine in outdoor areas have been provided with wallows to relieve this heat stress.

Confinement buildings present an added problem in that most are unadaptable to wallows due to slatted floors or other considerations. Also, ventilating rates are too high for economical use of traditional air cooling devices. Therefore, new innovations are being used to try to maintain comfort in swine buildings during warm periods.

One such innovation is the use of night air to remove excess heat from the thermal storage in solar collector systems. By cooling the storage during the relative cool nights of summer with air and drawing the daytime ventilating air through the storage, some cooling of a swine building may be accomplished. When considering the magnitude of cooling load and ventilating rates, a lowering of inside temperatures by even 10 F by using the cooled collector storage could result in dramatic energy savings. For example, in a building 80 x 30 x 8 feet inside dimensions, if the ventilating rate amounted to even one air change per hour, a reduction of 10 F in the inlet air would reduce the cooling load by over 3000 Btu per hour.

Little has been done to investigate these possibilities. Therefore, the purpose of the research reported here is to examine the use of this idea, develop some sort of procedure for utilizing this method, and to obtain useful information on the application of this method.

# REVIEW OF LITERATURE

## Swine Comfort and Performance

Studies have shown that swine perform better at certain temperatures and relative humidities. Heitman and Hughes (1949) found that hogs weighing 70 to 144 pounds converted feed to gain more efficiently at an ambient temperature of 75 F, while hogs weighing 166 to 260 pounds gained more rapidly at 60 F. Capstick and Wood (1922) suggested that the critical temperature of a 300 pound hog was 21 C (69.8 F). Kazarian, et al. (1958) found a 10% increase in growth rate and feed efficiency when comparing feeder pigs confined in ventilated and air-conditioned buildings to pigs in outdoor pens in the summertime.

Mangold, et al. (1967) concluded from their tests with finishing swine, that between 50 F and 75 F, self-fed pigs performed statistically the same. Above 75 F, feed intake and growth rates were lower than at 60 F, while efficiency was higher. Below 50 F, feed intake was higher, but growth rate and efficiency were lower, when compared to 60 F tests.

Little work has been done specifically on sow comfort levels or performance. Bond, et al. (1952) concluded, by observation and weight gain of lactating sows, that sows are more comfortable at 60 F and 70 F than at 80 F. Karhnak and Aldrich (1971) concluded that a range of 60 to 70 F should be the design temperature for sow comfort. Studies such as Heitman and Hughes (1949) also show that heavier feeder hogs are more comfortable at the lower temperatures, and some correlation may be assumed to sows of nearly the same weight.

Literature is available relating milk production by the sow to breed, but none was found relating to temperatures. Johnston (1958) did report that milk cows adapted to the temperature with a corresponding drop in milk production after an indeterminate lag time. It was theorized that the decrease in production was related to a decreased consumption of feed during times of heat stress.

Several studies have been made of the affect of heat stress on litter size. Roller, et al. (1967) tested a large number of gilts at three dry-bulb temperatures and four wet-bulb temperatures. At higher dry-bulb temperatures, higher wet-bulb temperatures, and combinations of the two, feed consumption and daily gain were lower than at lower temperatures. Evaluated at 25 days after breeding, the number of corpora lutea was significantly affected by increased dry-bulb temperatures, though the numbers of live embryo and percent embryo survival were not. They concluded that pre-breeding conditions were more influential than the same conditions imposed after breeding. It was suggested that an economic analysis of the benefits of environmental control for breeding stock should be based on a minimum of one month protection from heat stress per breeding.

Mahoney, et al. (1970) reported that the Oklahoma swine breeding project control sows averaged, over a three year period, 2.0 pigs farrowed and 1.2 pigs weaned per litter more for spring farrowing compared to fall farrowing. They stressed gilts in heat chambers and found a tendency towards reduced numbers of corpora lutea, fewer live embryos, lower embryo survival rates and smaller embryos when comparing the stressed gilts to gilts housed at a constant 74 F. They found that the most critical period for stress was 1 to 15 days post-breeding.

Tompkins, et al. (1967) reported a relationship between heat stress during the first five days after breeding and reduced litter size. Low level stress or late stress seemed to have little effect on the number of live embryos. They postulated that after 85 days of gestation, the sow would die from stress before the embryos would. Finally, Driggers, et al. (1976) reported no decrease in year round conception rate using boars continuously housed in a controlled environment and sows housed in a controlled environment either until breeding, or until 14 days after breeding.

## Ventilation

Swine buildings are generally ventilated such that excess moisture and odor can be removed while maintaining a tolerable air temperature inside the structure. Bond, et al. (1959) investigated heat loss by swine. There are four means by which heat is lost: radiation, convection, conduction and evaporation. The relative rates for each are given in Table 1 for various temperatures.

Table 1. Effect of Air Temperature on the Percentage of Total Heat Lost by Each of the Four Methods of Heat Transfer.*

| Air Temperature F | % of total heat loss | | | |
|---|---|---|---|---|
| | Radiation | Convection | Conduction | Evaporation |
| 40 | 34.9 | 37.8 | 12.8 | 14.5 |
| 50 | 33.0 | 38.7 | 12.8 | 15.5 |
| 60 | 32.9 | 38.7 | 11.8 | 16.6 |
| 70 | 27.0 | 34.3 | 10.7 | 28.4 |
| 80 | 23.0 | 32.0 | 7.7 | 37.3 |
| 90 | 17.2 | 20.7 | 7.4 | 54.7 |
| 100 | 2.6 | 5.0 | 2.8 | 89.6 |

*Table reproduced from Bond, et al. (1959).

As one can see, evaporation plays an increasingly important part in allowing the pig to maintain a more comfortable body temperature as the surrounding air temperature rises. Morrison, et al. (1967) reported that at close to optimal conditions, the source of the evaporative loss was from the lungs during respiration. As the dry-bulb temperature increased, however, with a constant relative humidity, the skin loss doubled while lung losses tripled due to a similar increase in the respiration rate. However, as the relative humidity increased from 30% to 90%, lung losses at first increased, then decreased rapidly until they accounted for only one third of the total moisture loss.

Should the pig be unable to lose enough heat to its surroundings, its body temperature will rise. Latent heat production will increase to help prevent heat stress in the animal. Mitchell and Kelley (1938) estimated that a lactating sow produces 2966 grams of water vapor a day under ordinary conditions and 11864 grams per day at and above 103 F, while producing 6881 calories per day of waste heat. Similarly, a ten pound pig would produce 800 calories along with 483 grams of moisture under ordinary conditions and 1379 grams at high temperatures. A 25 pound pig produces 1350 calories and 582 grams of water at ordinary conditions and 2328 grams of water at high temperatures. Bond, et al. (1952) estimated that a sow and litter would produce a combined sensible and latent heat loss of about 200 Btu per hour per hundredweight. Further, Bond, et al. (1959) reported that latent heat of approximately 1000 Btu per hour needs to be removed for each sow and litter in a confined area.

It has been recommended by the Midwest Plan Service (1975) that a ventilating rate of 210 cubic feet per minute per sow and litter be provided in the summer. Morrison, et al. (1966) reported that, at optimal temperatures, an increase in relative humidity from 45% to 95% produced no statistically significant difference in growth of pigs, while Morrison, et al. (1969) found that humidity has a significant effect on rate of gain but little effect on feed efficiency. With such studies in mind, ventilating rates are being recommended now, by some, of 100 cubic feet per minute per sow and litter. Also, Harman, et al. (1968) reported that slotted floored pens contain 42% of the water vapor of solid concrete floors. Many new farrowing buildings are of the slotted floor type, and would therfore, perhaps, not require the higher ventilating rates to remove moisture from the building.

## Cooling of Swine Environment

Even at high levels of ventilation, the lowest temperature in a building that a producer could hope to have would be close to the outside air temperature. Therefore, the question arises about how to lower the inside temperature so that the sows remain comfortable. Following the guideline of Bond, et al. (1959), trial calculations were made of the cooling load of a 30-sow farrowing house (see Appendix A). A cooling load of 249,013 Btu per hour was found to maintain 70 F and 50% relative humidity inside the farrowing house with outside design conditions of 95 F and 40% relative humidity. That figure represents the problems faced in trying to economically condition the air at high ventilating rates.

Several methods of cooling the inside environment exist. Kazarian, et al. (1958) found that the air was dusty enough that a normal window air-conditioner functioned erratically due to build up of foreign material on the controls and coils. Culver, et al. (1960) found that wallows helped fattening pigs maintain a lower respiration rate and lower body temperature. Although wallows are not common inside confinement buildings, Heitman and Hughes (1949) observed that heavy hogs were more comfortable when on a wet concrete floor than on a dry floor.

Sprinkling water on hogs has also been used to help them maintain comfortable conditions. Culver, et al. (1960) observed little change in respiration rate or body temperature of pigs sprinkled as the ambient temperature increased. Hale, et al. (1966) found that sprinkled pigs gain weight faster and ate more than pigs with shade only, although the feed conversion of the two groups was basically the same. Sprinkling may cause a problem by increasing the humidity inside a confinement building to uncomfortable levels. Also, small pigs in a farrowing house would be uncomfortable if wet, and it would be difficult to sprinkle the sows without also sprinkling the pigs.

Studies have also been done using cooled floors to help maintain pig comfort. Kelley, et al. (1964) found the relations shown in Table 2 between pig weight and floor temperature at outside ambient conditions of up to more than 100 F.

Table 2. Heat Transfer from Swine to Cooled Floor*

| Pig Weight Pounds | Floor Temperature F | Heat Transfer Coefficient Btu/hr-ft$^2$ |
|---|---|---|
| 70 | 65 | 49.3 |
| 125 | 65 | 38.9 |
| 150 | 70 | 61.2 |
| 150 | 80 | 61.0 |

*Table reproduced from Kelley, et al. (1964).

The higher heat transfer rates at higher floor temperatures were attributed to reduced blood flow to the skin when in contact with the colder floors. The pigs in this study were observed to be more comfortable, based on respiration rates and body temperatures, than similar pigs with outside wallows, but not as comfortable as if the air surrounding them had been cooled. The authors also reported that water condensed on the cooled floors, which resulted in an increased fly population. Spillman and Hinkle (1971) found the greatest heat transfer from the pig to the floor occurred with an air temperature of 92 F and a floor temperature of 75 F. Their analysis found that the heat transferred to the floor was affected by the air temperature and a combination of air temperature and floor temperature. Floor cooling could have the advantage of cooling the area where the sow would lay, but not the pig creep area in a farrowing house.

Other methods have been tried to cool sows without cooling the pigs in farrowing houses. Merkel and Hazan (1967) zone cooled sows by blowing air on the sow, rather than cooling the entire environment. They observed that sows zone cooled with 65 F air in a building with air temperature of 90 F showed less loss of weight, more feed intake, and better litter gain in the first two weeks after birth than sows which had inside air at 90 F blown over them.

Lipper and McGinty (1961) assumed that at high ambient temperatures swine depend heavily on evaporation of moisture from their lungs to maintain their body heat balance. Therefore, they provided cooled breathing air for sows. Based on increases in rectal temperatures and respiration rates of the sows, they found that heat stress could be reduced by using the cooled breathing air.

## Use of Solar Collector Storage

It becomes obvious that, for peak performance, sows should be protected from heat stress. As pointed out by Spillman (1977) this reduction in temperature could be provided, at least in part, by the use of nocturnal cooling of the storage in a solar heating system. It was reported that Manhattan, Kansas experiences an average daily temperature range of 24 F, with a greater fluctuation on days when the high temperature is above 95 F. By cooling the storage with night air, then drawing the ventilating air for a farrowing house through the storage, Spillman estimated that peak afternoon ventilating air temperatures could be lowered by 10 F in the building, providing a better environment for the sows.

The use of solar storage units for cooling depends on the type of system. Since the operation will be continuous, it should be such that maintainance and daily human input are minimal. Such a system also should use a minimum amount of electrical energy for its operation. Since the storage temperature would not have large changes, nor very high or low temperatures, a minimum of controls should be necessary. A passive, or nearly passive system therefore seems to be a possibility.

The exact meaning of "passive," as applied to solar energy uses, is a somewhat cloudy term. Yellott (1978) reported on three official definitions of the term "passive system."

First, a definition compiled by the National Bureau of Standards for the Department of Housing and Urban Development (HUD) reads: "A passive solar heating system is an assembly of collectors, thermal storage device(s) and transfer media which converts solar energy into thermal energy and in which no energy in addition to solar is used to accomplish the transfer of thermal energy. The prime element in a passive solar system is usually some form of thermal capacitance.

The second definition, also from HUD, is: "A 'passive' solar energy system is one which uses the building structure as a collector, storage and transfer mechanism with a minimum amount of mechanical equipment." This definition seems to ignore the use of thermosyphons as passive collectors.

The third definition was developed at the Conference on Passive Systems at Albuquerque, New Mexico in May, 1976: "Passive systems use the sun's radiation for heating and nocturnal processes for cooling; heat distribution is accomplished by convection, radiation and conduction. Non-renewable energy used for movement of insulation, diurnal transfer of water from one space to another, movement of dampers or valves, etc. must be so small in amount that the 'Coefficient of Performance' of the system, defined as the ratio of the useful heating or cooling accomplished by renewable energy sources or sinks to the non-renewable energy consumed, is greater than 50 to 1." In other words, when heating or cooling can be accomplished with negligible use of non-renewable energy, then the system may be called "passive."

Yellott further discussed three principal passive heating systems. Type 1 is "Sun + Space + Building Mass Storage." A very simple example of this type would be a window which lets solar energy into a building, which then heats up the contents of the space. This type, when permanently installed does not offer very much potential for summertime cooling. At best, it could be constructed to keep direct summer sunlight from entering the space.

A second type is "Sun + Natural Convection + Storage," typified by the thermosyphon. This system utilizes changes in density of transfer fluids, such as air or water, to cause the fluid to flow, thus transferring the thermal energy. This type would be better suited to cooling than type 1, but still has problems. The transfer fluid would have to be cooled by some means, either radiation to surroundings or the sky, or by some transfer means to cooler ambient air. However, the potential for cooling would be relatively small, since thermosyphon systems normally have a ten degree or smaller temperature change in the fluid, with correspondingly low flow rates. Also, if cooling was accomplished by heat transfer to ambient air, it stands to reason that the ambient air would be cooler than the transfer fluid, unless means were taken, by the use of outside energy, to increase the potential for heat transfer by compression of the transfer fluid or other means.

The third type of passive system is "Sun + Storage Mass + Space." This type includes the Indian pueblos of the Southwest and, more recently, "drumwalls" and buildings which use the so-called "Trombe wall." Developed by E.L. Morse at Salem, Massachusetts in the 1880's, Trombe and Michel have incorporated this concept in several houses around Odeillo, France. This wall consists of a south-facing masonry wall covered by a transparent cover or covers. Some means are available for natural convection to circulate air passed the front of the masonry into the space to be conditioned, but most of the heating is by radiation from the inside surface of the wall. The wall serves, therefore, both as collector and storage medium. Kohler (1978) recommended minimum storage for a south-facing passive masonry wall of 150 pounds per square foot of glazing, which corresponds to 30 Btu per square foot of glazing per degree

Fahrenheit temperature change. The "drumwall," developed by S. Baer of Conales, New Mexico, uses a similar concept, though water-filled drums are used as the storage medium.

Modification of this last type were chosen for use as farrowing house collector-storage units, as described by Spillman (1975), Murphy, et al. (1977), and Spillman (1977). The systems described consist of masonry walls through which the ventilating air for the buildings were drawn, heating the air as it passed through openings in the storage wall. Thus, non-renewable energy is used only to power fans to move the air. Also, the argument could be made that little or no additional energy is required for the system because the fans must be present to give necessary ventilation to the buildings anyway. The only additional energy may, therefore, be considered to be that energy needed to overcome the effects of the additional pressure drop of the system as the air follows a path through the covers and the wall. Though not analyzed, this energy requirement would seem to be small enough to satisfy the requirements of the third definition of passive systems given previously.

## Methods of Storage

Solar energy, as well as any other energy form, may be stored in several ways. These include thermal capacitance storage, using rock, water, or other appropriate medium, conversion of mechanical to potential energy, phase-change storage and electrical storage. Due to the nature of the desired application, electrical and potential energy storage are not acceptable methods here.

The use of thermal storage in water, rock or masonry, and the use of phase-change storage, such as eutectic salts, could be utilized for heating purposes. However, as pointed out by Duffie and Beckman (1974) most materials with suitable melting points have problems maintaining consistency over repeated cycling. At the present time, eutectic salts are also expensive.

Packed pebble beds, consisting of loosely packed rock of consistant size, are used for storing thermal energy. Godbey, et al. (1977) reported on the performance of packed gravel bed storage under a residential house. Though their cooling system had not been tested, they estimated that the storage could be flushed of its daytime heat buildup if the ambient temperature dropped below 70 F, and that just a 7 F temperature difference between ambient and storage temperatures would result in energy savings of 30.86 KWh of cooling per day. One could conclude that a swine building, with greater cooling and ventilating needs than a residence, could experience energy savings proportionately greater.

Forbes (1977) reported on a similar experiment for poultry houses using air charged packed gravel beds for storage. Using storage under the floor slab and in a confined enclosure in the conditioned space, he concluded that the storage needed to be as large as possible and that cooling could be provided with a buried bed whether cooled with night air or not.

Due to its relatively high percentage of void space, which helps reduce energy requirements to move air through it, packed beds do not conduct heat very well, depending on convection for heat transfer. Therefore, its use as a passive collector-storage unit for direct collection of solar energy would be impaired. Concrete shares many of the advantages of packed beds while being able to conduct heat to a greater degree due to higher density and fewer voids. Thus, it is better adaptable to the passive collector-storage wall concept.

There are advantages for using either water or concrete for storage. Water will store approximately one Btu per pound per degree Fahrenheit temperature change, when in the liquid phase, and one cubic foot weighs approximately 62.3 pounds. Therefore, 62.3 Btu per cubic foot per degree could be stored. Concrete will store approximately 0.2 Btu per pound per degree, depending on the make-up of the mix. If the concrete weighs 150 pounds per cubic foot, again depending on the mixture, then approximately 30 Btu per cubic foot per degree could be stored, roughly half that of water. Water, however, must be protected from freezing, either by emergency auxiliary heat or some type of antifreeze solution. It also must be stored in tight containers to prevent leaks. Still, the main advantage of using a concrete wall for storage and collection, as pointed out by Spillman (1975), is that it is structurally sound, and can replace the conventional south wall in new buildings. Because the conventional south wall is not needed, the cost of such a wall could be subtracted from the cost of building a collector-storage system. This would lower the net cost to the producer of building such a system.

## INVESTIGATION

### Objectives

With the preceding studies in mind, three objectives of this research were developed. First, it was desired to know if the nocturnal cooling of a solar collector-storage system for use in cooling daytime ventilating air would be a feasible venture. Specifically, would the ventilating air temperature be lowered enough to significantly affect the comfort of the sows inside a farrowing house, and could this be done economically?

Secondly, it was desired to draw conclusions relating the use of the unit during the spring and fall when cooling may be desired during the day, but heating needed at night. This knowledge could aid a producer in managing such a unit for the best year round performance.

Finally, it was desired to find at which combination of air flow rate and block spacing cooling of the blocks was optimal. Since cooling of ventilating air is dependent on cooling the storage, such an optimal situation should maximize the cooling potential of the system. Also, the optimal air flow rate should be dependent upon the need for ventilation inside the building and the cost of suppling the needed air flow. Higher rates would make a larger fan necessary, which could use more electric energy.

### Theory

In investigating the performance of a collector-storage system, one must compare the heat transfer from the storage to the air for different combinations of spacing and flow. For a system such as this, other considerations which need to be known include Reynold's number, length of passage, and passage diameter.

First of all, when making calculations, a value for the hydraulic diameter is needed to calculate Reynold's number (Re). ASHRAE (1977) suggests that the hydraulic diameter for non-circular ducts equals four times the ratio of cross-sectional area to perimeter of the duct. Also, for parallel plates, the value equals twice the distance between the plates. The question arises about which situation is represented here. As will be explained later, the model used for testing had aspect ratios, the ratio of length to width of the passage, ranging from 14.5 to 29.0. The full-sized unit monitored had an aspect ratio of approximately 32.0. Hartnett, et al. (1962) concluded that, when $6000 < Re < 5x10^5$, the correlation between circular tubes and rectangular tubes with equal hydraulic diameters held for all aspect ratios when finding friction coefficients for flow. Huebscher (1947) concluded that this relationship held only up to aspect ratios of 8, for mean air velocities of 300 to 9000 feet per minute. Preliminary calculations show that air speeds ranging from 50 to 200 feet per minute would be used, representing air flows of one to three cubic feet per minute per square foot of collector area. The corresponding Re would be between 100 and 830. Since this topic was not well researched or reported upon, it was concluded that the parallel plate treatment would probably be sufficient, but the passages were treated as rectangular ducts for calculations. For spacings of 0.25 inches, the difference in the calculated values for the hydraulic diameter is 6.8 percent, which could be significant.

Also of importance in making calculations is whether flow is laminar or turbulent. The Chemical Engineer's Handbook (1963) lists the transition from laminar to turbulent flow as ranging between Re of 2100 and 10,000 for tube flow. Likewise, McAdams (1954) gives values of 2100 to 2500 to the Re corresponding to the transition zone. Therefore, flow is considered to be laminar through the wall.

When considering heat transfer from the front of the wall as the air flows over it, Re between 3500 and 10500 are encountered. Holman (1972) uses the following relationship for calculating the heat transfer coefficient, h, from a flat plate:

$$Nu = 0.664\ Pr^{1/3}\ Re^{1/2} \tag{1}$$

where: $$Nu = \frac{hd_e}{k} \tag{2}$$

$$Pr = \frac{c_p\ \mu}{k} \tag{3}$$

$$Re = \frac{\rho\ V_o d_e}{\mu} \tag{4}$$

Norris and Streid (1940) point out that variations in viscosity of air with temperature are too small to justify its evaluation for each temperature, therefore, a constant value of 0.044 pound-mass per foot per hour is used in all calculations. Inspection of tables in Holman (1972) revealed that $c_p$, k, and Pr varied very little over the range of temperatures considered. Therefore, values of 0.240 Btu per pound-mass per Fahrenheit degree, 0.0154 Btu per hour per foot per Fahrenheit degree and 0.71 (dimensionless) were used, respectfully, for $c_p$, k and Pr.

When calculating the heat transfer in a duct, the length of passage is important. When the air enters into the duct, a certain length is necessary before the flow becomes fully developed. McAdams (1954) reported that h varied until the length to diameter ratio equalled 60. Such ratios for the tests reported here range from about 32 to 64. Knudsen and Katz (1958) reported that lengths of 50 tube diameters were necessary for the formation of fully developed turbulent flow, while a length of 30 diameters is necessary for h to become constant. Also, it was reported that the development length for laminar flow between parallel planes is represented by the relation:

$$\frac{L_e}{2b} = 0.00648\ Re \tag{5}$$

where $L_e$ = development length
$b$ = plate spacing

Knudsen and Katz (1958) and Holman (1972) both report on a formula for calculating the heat transfer in the entrance region of a duct for turbulent flow as:

$$Nu = 0.036\ Re^{.8} Pr^{1/3} \left(\frac{d_e}{L}\right)^{.055} \tag{6}$$

for $10 < L/d_e < 400$

Knudsen and Katz (1958) also report that for laminar flow, the entrance region heat transfer could be obtained using the following:

$$Nu = 1.077\ Pe^{1/3} \left(\frac{d_e}{x}\right)^{1/3} \tag{7}$$

for $100 < Pe\ d_e/L < 500$

where $Pe = RePr$ (8)

Also reported were three equations for the entrance region of circular ducts for use in a specified situations, where Pr = 0.7. They are:

For a constant wall temperature:

$$Nu = 3.66 + \frac{0.104\ \left\{\left(\frac{x}{d_e}\right)/Pe\right\}^{-1}}{1 + 0.016\ \left\{\left(\frac{x}{d_e}\right)/Pe\right\}^{-0.8}} \tag{9}$$

For a constant temperature difference between the air and the duct walls:

$$Nu = 4.36 + \frac{0.1 \quad \{(\frac{x}{d_e})/Pe\}^{-1}}{1 + 0.016 \quad \{(\frac{x}{d_e})/Pe\}^{-0.8}} \tag{10}$$

For a constant heat input:

$$Nu = 4.36 + \frac{0.036 \quad \{(\frac{x}{d_e})/Pe\}^{-1}}{1 + 0.0011 \quad \{(\frac{x}{d_e})/Pe\}^{-1}} \tag{11}$$

where x is the point at which h is evaluated.

However, none of these requirements are met exactly by the situation at hand. The wall temperature varies through the wall by as much as several degrees, while the temperature difference and heat input vary at each location.

At times, entrance factors may have added affect on the flow in a duct. However, Hartnett, et al. (1962) reported that for low Re, entrance configuration had little effect. Also, Knudsen and Katz (1958) reported that duct roughness has no effect on velocity distribution or friction loss.

For developed laminar flow in rectangular ducts little information had been reported. McAdams (1954) reported the following relation for such flow:

$$Nu = 0.5 \frac{d_e \; G \; c_p}{k} \frac{d_e}{L} \tag{12}$$

This relationship was developed from studies with ducts with aspect ratios of 6.43.

As shown, the basic equation for heat transfer is some function of the Nusselt, Prandtl and Reynolds numbers. By substituting the appropriate components for each term, h is found to be proportional to the air velocity raised to some power, and inversely proportional to the hydraulic diameter, also raised to some power. The powers depend upon which relationship is used. Therefore, theoretically, as the velocity increases or the spacing decreases, the convective heat transfer coefficient should increase until flow is restricted to the point that the coefficient is decreased rather than increased. This point, however, is not well defined.

## Materials and Equipment

Studies were made using two different units. First, a small unit, representing a portion of a full-sized unit, was used indoors. Secondly, a full-sized, existing unit located at the Kansas State University Experimental Swine Farm was monitored.

The small model was one described by Spillman (1975) with some modifications. It consisted of a storage wall composed of solid concrete blocks with dimensions 3 5/8 x 7 5/8 x 15 5/8 inches for large blocks, and several smaller blocks half as wide to facilitate stacking. The wall represented a two feet square section of a full-sized unit. The average weight of the 15 large blocks was 32.83 pounds and the average for the six smaller blocks was 16.08 pounds for a total of 588.95 pounds of storage. The blocks were dry stacked so that they could be moved. Their front surface was painted with 3-M Nextel* black paint, amounting to a 22 inches by 24 inches surface. In front of the blocks were two 4-mil TEDLAR transparent covers spaced one inch apart. Sheilding the

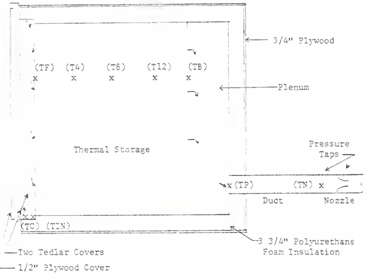
*Mention of specific items is for description only and does not imply endorsement.

front was a cover of 1/2-inch plywood, spaced approximately 3/4 of an inch from the TEDLAR covers. This represents a cover which could be placed over a full-sized unit during the summer to block solar radiation. Behind the blocks was a plenum. The sides were then covered with three layers of polyurethane foam insulation, each 1 1/4 inches thick, and a sheet of 3/4-inch plywood.

Flow through the model was measured using an ASME long-radius flow nozzle as outlined in the ASME Power Test Codes (1959). The nozzle was located in a duct, which opened from the lower center portion of the plenum and which had a fan attached to the end to provide air flow by sucking air through the model and out the duct. A micromanometer was used to measure pressure drop across the nozzle. See figure 1 for construction and operation details.

For this model, a low $\beta$ series nozzle with throat diameter (d) of 0.75 inches was installed in a plastic pipe with a nominal inside diameter (D) of three inches. A low $\beta$ series nozzle is one where $\beta < 0.5$. Here, $\beta = d/D = 0.25$. Following the guidelines mentioned above, the nozzle was firmly clamped and centered in the plastic tube. Pressure taps were located 1.5 d, or 1.125 inches, from the flange for the inlet pressure. The temperature of the air was measured by a thermocouple located at the same point as the inlet pressure tap.

The guidelines call for undisturbed pipe six diameters upstream and about 2.2 diameters downstream, or the use of straightening vanes. These lengths, 18 and 6.6 inches, respectively, were more than met by the equipment used, therefore, no straightening vanes were used. Also, the plastic pipe used had a very smooth inside surface, so the air stream was not unduly disturbed.




x: Denote placement of thermocouples
Arrows show flow path of air through system

Figure 1. Cut-Away View of Solar Collector-Storage Model
Used in Cooling Tests, Winter, 1977-1978

By using the dewpoint temperature of the room air, the measured nozzle temperature, and the barometric pressure obtained from the Department of Physics, Kansas State University, the specific weight of the air passing through the nozzle could be calculated. Although the dewpoint temperature was obtained for the room air, the heating of the air as it passes through the storage is a constant moisture process, therefore the dewpoint remains the same.

The standard ASME (1959) formula for use with compressible fluids for long-radius flow nozzles was used to obtain the flow. The formula is:

$$W_h = 359\ CFF_a d^2 Y_a \sqrt{h_w\ \gamma} \qquad (13)$$

where: $W_h$ = weight rate of flow, pounds per hour
$C$ = coefficient of discharge
$F$ = velocity of approach factor, $1/\sqrt{1-\beta^4}$
$d$ = nozzle throat diameter, inches
$F_a$ = thermal expansion factor
$Y_a$ = expansion factor
$h_w$ = pressure drop across the nozzle, inches of $H_2O$ at 68 F
$\gamma$ = specific weight at inlet side of nozzle, pounds per $ft^3$
$\beta$ = nozzle throat diameter divided by duct diameter

With a 0.75 inch nozzle and a three inch duct, $F = 1.002$. Values of $F_a = 1$ and $Y_a = 0.99$ were chosen from charts in ASME (1959). Therefore, equation (13) reduces to :

$$W_h = 200.32\ C\sqrt{h_w\ \gamma} \qquad (13a)$$

for this combination of duct and nozzle.

The value for $h_w$ was read from a micromanometer at the start of each cooling cycle. Values for C were calculated by iteration from the formula from Benedict (1965):

$$C = 0.19436 + 0.152884\ (\ln Re) - 0.0097785\ (\ln Re)^2 + 0.00020903\ (\ln Re)^3 \qquad (14)$$

An intial guess of C = 0.95 was used to begin the iteration. Reynolds number, Re, was calculated, for the nozzle flow, from the flow shown by the previous estimate for C. After obtaining the flow in pounds per hour, it was converted to cubic feet per minute. See Appendix C for the computer program statements to perform this process.

The full-scale model was added to an existing farrowing house at the Kansas State University Swine Farm during late 1975, see figure 2. It consists of a collector-storage unit eight feet high and 50 feet long, with a net collecting surface of approximately 380 square feet. The wall consists of solid concrete blocks, 6 x 8 x 16 inches, laid up without mortar approximately 16 inches in front of the existing south wall of the building. The front surface of the blocks are painted black. Vertical gaps averaging 0.1875 inches were left between blocks. The total weight of the storage wall is approximately 31 tons.

Two transparent covers, spaced 1.5 inches from each other and the blocks surface, were fastened over the south side of the wall. These covers were undergoing other tests and were changed several times in the course of the study. Moveable plywood panels were in place over the entire surface for most of the tests. These panels were to screen the surface from the sun during the cooling operation.

Two different means of measuring air flow were used. During late summer 1977, a 16 inch propeller fan, 16S728, and orifice purchased from Aerovent Corporation, were installed in the east end of the plenum. The fan was driven by a manually adjusted variable speed motor. When cooling was desired, inlets to the building were closed and the fan was manually turned on, blowing air into the plenum, through the blocks and out the covers, essentially the opposite of normal operation. Air

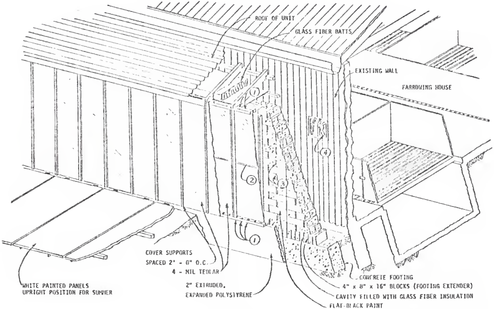



Figure 2. Construction details of the solar collector-storage unit. Ventilating air moving through the system enters at point 1, moves between the covers to point 2, then through the vertical gaps between blocks, point 3, and to the fan where it is moved into the environmental space. The wall is of 6- x 8- x 16-inch solid concrete blocks.

flow rates were interpolated from fan curves from measured fan speeds and static pressure in the plenum. This mode was called backflushing. During the fall of 1977, a grain bin was installed nearby which had a duct leading to the plenum. Therefore, for tests in the summer of 1978, the bin fan was operated to pull air out of the plenum. Air flow was measured by use of pitot tubes and a micromanometer. Flow from the plenum could be regulated by altering ambient inlets to the fan.

On the full-scale unit, the blocks were heated by the ventilating air being drawn through them during the day. The small model was located in a room with essentially a constant temperature, therefore, artifical means were necessary to heat the blocks. This was accomplished by ducting heated air to the inlets to the model. In other words, the model dealt only with air at the high and low points of its range, whereas, the full-scale unit dealt with fluctuating temperatures.

Data for all tests were read at one hour intervals. At the full-sized unit an Esterline Angus PD 2064 data logger supplied the data in Fahrenheit degrees. For the model, the first part of the data was recorded by a Campbell Scientific, Inc, CR5 system, in Celsius degrees. The rest of the data was complied by a Digitec 1268 data logger, which recorded a millivolt potential difference which was then converted to Fahrenheit degrees.

## Procedure

Preliminary tests were taken beginning in August 1977 with the full-scale unit. The cooling cycle would begin when the backflushing fan was turned on at some arbitrary time. Inspection of data showed that on hot days, the ambient temperature would not fall below the block temperature until 9:00 p.m. or later. Therefore, it was attempted to begin the cooling cycle at that time. Then, the duct into the building was closed off and the fan, on a variable-speed motor, was adjusted to a predetermined speed. Data was taken at air flow rates of six, five and three cubic feet per minute of air per square foot of collector surface. Data was also available from earlier in the summer when a constant two cfm per square foot of collector was being drawn through the wall into the building.

Data recorded, on an hourly basis, for cooling analysis, were temperatures of the concrete block storage, ambient temperature and plenum air temperature. Times were noted when the unit was in a storage cooling mode. When not in the cooling mode, ventilating air was being drawn through the storage into the building, at a rate of approximately two cfm per square foot of collector. This rate represented the ventilation required for the swine inside.

The cooling mode normally ran about twelve hours. Inspection of data showed that from 8:00 to 10:00 a.m. the ambient temperature would normally rise above the block storage temperature. Again, termination of the cooling mode was done at arbitrary times, though there was some consistancy.

There were several complications that arose in the data taking process. First, the summer of 1977 was fairly cool for Kansas, and beginning in mid-August, the temperatures were almost cold, with high ambient temperatures failing to rise much out of the 70's F range at times. This was coupled with a relatively steady temperature, lacking the large daily range common to summer weather. Therefore, data were not considered to be very representative of a normal year. Also, there were equipment problems, with three different variable-speed motors being used at one time or another. Finally, by obtaining air flow rates by interpolating from fan curves, it was thought that gross error might be present. Relative rates could certainly be obtained, however, to give an indication for future considerations.

By inspecting the data that was obtained, there was no noticeable difference in performance between the different air flow rates. On those days with normal temperature fluctuations, the blocks would be heated by the daytime ventilating air to within 5-8 F of the ambient daily high temperature. Similarly, the storage would cool to roughly the same 5-8 F above the ambient low temperature at night during the storage cooling mode.

Using these indications as guides, the testing continued during the winter using the model indoors. It was decided to look at the model performance for a relationship between spacing between the blocks, air flow rates and cooling of the storage. Since preliminary results of the summer had shown no noticeable advantage of using the higher air flow rates, lower rates were considered, specifically, one, two and three cfm per square foot of collector. These rates would be more suitable for ventilating rates for farrowing houses. Also, cooling was not done by backflushing the model, but drawing air through it in the normal outside-to-inside manner.

The full-sized unit was constructed with 3/16 inch spaces. Also, several other existing units had been constructed with this spacing. For these reasons, and projections of performance for the system, spacings tested were 1/8, 3/16 and 1/4 inch. This was considered to be the applicable range for spacing. As shown in the theory section of this report, the heat transfer rate should theoretically increase with a decrease in spacing. Conceiveably, there could be a point where flow is too restricted for good performance. Therefore, these spacings were investigated.

Air flow rates were set using the 1/4 inch spacing. Since the blocks used in the model wall were not the same size as the ones used in the full-sized unit, there was more area for air passages through the model per square foot of surface. Therefore, an average velocity of air passing through the wall was used to set the flow rates. With the 1/4 inch spaces, the velocity through the model wall was the same at total air flow rates of 5.21, 10.42 and 15.64 cfm as in the full-sized unit for air flow rates of one, two and three cfm per square foot of collector area, respectfully. So that the effect of changing the spacings could be studied, these air flow rates were used for all spacings, rather than keeping the same velocities.

The model was kept in a room with a fairly constant temperature, ranging in the low 70's F. To heat the blocks for subsequent cooling, heated air was ducted to the inlets of the collector system. It was drawn through the system by using a fan on a duct leading from the plenum. Air flow rates were measured by using an ASME long-radius nozzle, 3/4 of an inch in diameter. Pressure drop was measured across the nozzle by use of a micromanometer. Once daily, normally at the time when operating modes were changed, the pressure was read and readings for wet- and dry-bulb temperatures were taken with a sling psychrometer. When necessary, the flow was altered by restricting the outlet area of the fan. Two different fans were used to supply the necessary range of air flow.

Control of whether the system was in a heating or cooling mode was maintained by operation of the heated air source. During the first portion of tests, the heater was turned off after the blocks were heated. Room air, normally ranging from 70-75 F was then drawn through the system to provide cooling. However, the room temperature, which was independently controlled, became very erractic after a time, varying as much as five degrees in one hour. Therefore, inlet air was supplied solely by the heater. The blocks were heated to a higher level, then the heater was adjusted such that a 20 to 25 F temperature differential existed between the warmest air supplied and the coolest. In this way, a more steady temperature was maintained by the cooling air.

Following the observations from the previous summer's tests, the blocks were heated approximately 20 F above room conditions, when cooling was to be provided by room air. Cooling then continued until the blocks were within 5 F of the temperature of the ventilating air, or all night, depending upon the timing of the test. All the operation was manual, with no automatic switching of modes.

During the fall of 1977, a grain bin was installed near the full-sized unit. Included in the grain system was a duct which could draw off air from the solar collector-storage unit as part of some separate research. Therefore, the grain bin fan was used for moving the air through the unit for testing. The fan was set at a constant setting, the flow being altered by restricting the inlets for ambient air to the fan. Flow was determined by the use of a pitot tube in the rectangular duct leading from the collector unit. The tube measured the velocity pressure, which was used in the formula:

$$V_o = (\frac{2g \times Pv}{\rho})^{\frac{1}{2}} \tag{15}$$

where $V_o$ is the velocity of the air flow.

An average velocity pressure for the 12 observations made at each test was used to obtain the air velocity. The density was obtained by reading the dry- and wet-bulb temperatures with a psychrometer inserted into the duct. Then, by finding the product of the velocity and the area of the duct, flow was obtained. Data was taken at air flow rates of 2.07 and 2.95 cfm per square foot of collector area.

# RESULTS AND DISCUSSION

## Summer 1977

During the summer of 1977, the solar collector-storage unit at the Kansas State University Swine Experimental Farm was shielded from the sun by plywood panels while ventilation, approximately two cubic feet per minute per square foot of collector area, was being drawn through it. Temperatures for ambient air, plenum air and average block are shown in figure 3 for the period of July 20 through July 26. An inspection of this figure shows that there is a trend for the block temperature to rise to within 5 to 10 F of the high ambient temperature for the day, then fall to within 5 to 10 F of the nighttime low ambient temperature, when there was a normal diurnal temperature change. The plenum temperature, which reflects the temperature of the ventilating air into the building, closely follows the pattern of the block temperature. Even one day when the ambient temperature rose above 100 F, the plenum temperature only rose to about 88 F.

Two interesting observations can be made from this figure. First, one day of abnormal ambient temperature will not cause a huge fluctuation in the block temperature. The wall seems to react slowly to such fluctuations, requiring several days of hot weather, for example, to raise its average daily temperature very much. Secondly, on several occasions, the plenum temperature is higher than either the storage temperature or the ambient during nocturnal hours, while being cooler then either of the others during the daytime.

Both of these phenomena can be partially explained by the physical characteristics of the storage wall and calculation of its average temperature. The average storage temperature is based upon three

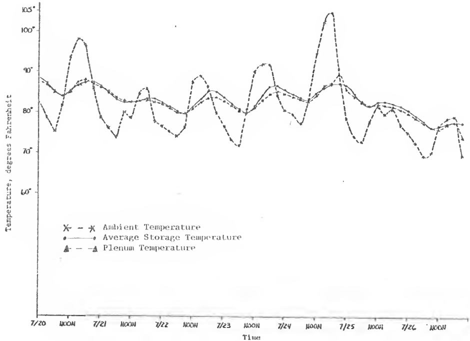



Figure 3. Performance with Continuous Ventilation of 2 cfm/ft$^2$ with Prototype Unit at the KSU Swine Experiment Farm, Summer, 1977

readings on the full-sized unit: the temperatures of the back surface, front surface and middle of the wall. The middle temperature is considered to be representative of the center half of the block, while the surface temperatures are each taken as the temperature of 1/4 of the block's volume. However, as it is heated and cooled, a temperature gradient, or "wave," moves through the storage. Therefore, if the wave peak occurs between points where the temperatures are read, the average storage temperature calculated may not be a true indication of the actual temperature. Also, this wave moves slowly within the wall, such that more than one day of high ambient temperatures are needed before the wave is moved far enough into the wall that most of it is not moved out of the wall as it cools at night. Similarly, a peak could occur such that the air, as it moves through the wall, could be cooled more towards the rear of the wall, so that the plenum temperature is cooler than the apparent average storage temperature. Also, heat could be conducted through the building wall from the interior, causing the plenum temperature to rise above the storage temperature. Finally, there are hygroscopic properties of concrete, which will be discussed later, which could cause such a situation.

While errors certainly occur due to the limited positions for reading temperatures of the storage, such errors are not considered excessive. Although this situation does create results which are not readily apparent, observing the storage as the temperature wave is moved back and forth illustrates another adaptation for the unit. During the spring and fall seasons, there are many days which are warm during the day, but cool during the night. A farrowing house kept at 60 to 70 F may have need for cooling during the day, yet require heating at night to maintain comfort. By using the collector-storage unit, the incoming ventilating air could be maintained at a more constant temperature, eliminating the need for much, if not all, of the heating and cooling which otherwise would be needed.

Later in the summer of 1977, the experimental unit was fitted with a fan, so that, at night, outside air would be blown into the plenum and out through the storage and covers. Air flow rates used were six, five and three cubic feet per minute per square foot of collector area. During the day, air was drawn through the storage into the building at the rate of about two cubic feet per minute per square foot of collector area. Equipment problems with fan motors delayed operation until mid-August when Manhattan experienced unseasonable cool, wet weather. Often the daily high ambient temperature was only in the mid-80 F range, with 10 to 15 F diurnal fluctuations. Therefore, the data collected during this period were somewhat suspect, and used mainly for indicating which tests were needed in the future.

Overall, the unit did not show a great deal of added response to increased ventilating rates. Though not enough data could be obtained to justify a strong conclusion, the unit seemed to follow a similar response as for the two cfm rate; that is, the storage would cool to within 5 to 10 F of the nightly low, then heat up to 5 to 10 F of the daily high. This pattern roughly held for all air flow rates investigated. See figures 3 and 4.

During mid-September, the panels which had been covering the front of the collector were lowered. The unit was backflushed at night with approximately five cfm per square foot of collector area, then had normal ventilation of about two cfm per square foot during the day.

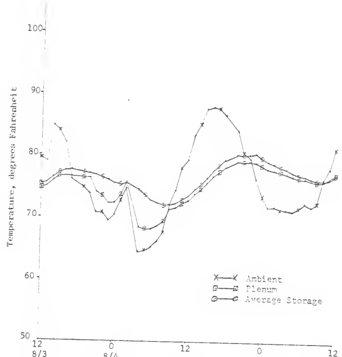



Figure 4. Performance with Nocturnal Backflushing at 6 $cfm/ft^2$ with Prototype Unit at the KSU Swine Experiment Farm, Summer, 1977

See figure 5 for a graph of the system performance. The storage, now exposed to direct sunlight, heated up to around 100 F during the day, and then cooled down by as much as 30 F at night. This cooling was partly due to increased radiation losses from the front of the wall, where the storage temperature was greatest, as well as greater differences in temperature between the storage and cooling air. Meanwhile, ambient temperatures ranged from highs of 70 to 80 F to lows of 50 to 60 F. The plenum temperature fluctuated quite a bit, though never dropping much below 60 F at night. Though the plenum temperature did rise to near 90 F one day, it normally fluctuated between 80 and 60 F each day. This would suggest that heating would have been unnecessary for the comfort of the sows, while any additional cooling needed would have been minimal. While this plenum temperature represents a backflushing mode, if the normal ventilation was drawn through the storage at night, the plenum temperature at night should be higher than that found here. Thus heating should be even less necessary.

## Winter, 1977-1978

With these preliminary results in mind, tests were begun on the model housed in the laboratory area of the Agricultural Engineering Department in Seaton Hall, Kansas State University. Since there appeared to be no advantage to larger air flow rates, air flows relating to one, two and three cubic feet per minute per square foot of collector area were used. The concrete blocks which comprised the storage were spaced at 1/4, 3/16 and 1/8 inches at each air flow rate.

The storage was heated by ducting heated air to the inlets of the model, to be drawn through the system by the fan. During the cooling

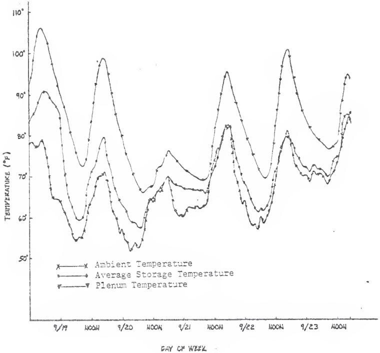



Figure 5. Performance with Nocturnal Backflushing at 5 cfm/ft$^2$ with No Radiation Shield, Using Prototype Unit at the KSU Swine Experiment Farm, Summer, 1977

mode, the heated air supply was stopped and room air was drawn through the system. Storage temperature was obtained by measuring one block at five points: front and back surfaces, and points four, eight and twelve inches from the front. A method similar to the one previously discussed for the full-sized unit was used to obtain an average temperature. Each combination of spacing and flow rate was monitored on an hourly basis through three cooling cycles.

The data was analyzed using the Kansas State University Itel computer system. Flow rates were calculated using the dewpoint temperature, atmospheric pressure and pressure drop across the nozzle taken at the start of each test and hourly readings for the nozzle temperature. Since fans basically deliver a constant volume, the flow rate calculated at the beginning was delivered throughout the test. The dewpoint temperature was read for room conditions rather than at the nozzle. However, the heating process was one of constant water content of the air, therefore, the dewpoint temperature at the nozzle should equal that of the room. It was found that the fan used was inadequate to deliver 3 cfm per square foot as planned, but delivered only about 2.8 cfm per square foot.

Temperature data was recorded as degrees Celsius and millivolt equivalents, therefore, conversion was made to degrees Fahrenheit. Temperatures read in were five storage temperatures, ambient, plenum, nozzle, inlet to the system and between the storage wall and inside cover.

After the tests were completed, it was decided that a temperature was needed at the inlet into the storage wall. Several tests were made recording the air temperature as it entered the storage wall.

A relationship was then found concerning the difference between the air temperatures entering the storage (T1) and entering the system (TC) to the difference between the temperatures at the front storage surface (TF) and the air entering the system (TC). Using data from tests run at two and three cubic feet per minute per square foot of collector surface, the following relationship was found with the Least Square Method:

$$Y = -0.1152 + 0.5797\ X \tag{16}$$

where: Y = Temperature entering storage -temperature entering system, (T1 -TC)
X = Temperature of front storage surface - temperature entering system, (TF - TC).

The correlation coefficient, R, was found to be 0.9894 and inspection of data showed a strong linear trend. A value of 1.0 for R is a perfect fit. The temperature of the air could therefore be found by calculating Y, then adding it to the air temperature entering the system. See Appendix B for details of the derivation of the above equation.

To calculate the h value for the cooling process, many formulas were inspected and rejected. The flow rates were so low that no analytical relationships for h were found for these rates. Velocities in the passages through the storage wall ranged only from 55 to 310 feet per minute, with subsequent Reynolds numbers of 226.5 to 635.9. Therefore, a formula was used which deals with transient heat flow in systems with negligible internal resistance. The formula, from Kreith (1973) is:

$$\ln \frac{T - T_\infty}{T_o - T_\infty} = - \frac{h\ A_s}{c_p\ \rho\ V}\ \theta \tag{17}$$

where: $T$ = average temperature of the system, here the storage
$T_\infty$ = average temperature of the air in the passages
$T_o$ = initial temperature of the system
$h$ = convective heat transfer coefficient
$A_s$ = surface area
$c_p$ = specific heat of the system
$\rho$ = density of the system
$V$ = volume of the system
$\theta$ = time

This relation assumes that the temperature of the system, or, in this case, the storage wall, will be uniform at any instant. Inspection of the data in Appendix D shows that this is not entirely true for this system, temperature differences between the front and back surfaces of the wall may be as high as ten degrees. Kreith (1973) states that for bodies whose shape resembles a plate, a cylinder, or a sphere, the error introduced by assuming uniform temperature at any instant is less than 5 percent when the Biot number, $\frac{hL}{k}$, is less than 0.1, where L is a characteristic dimension of the body and k is the thermal conductivity of the body. Here, L is considered to be the ratio of the volume of the large blocks divided by the exposed surface area, or 0.25 $ft^3$/ 0.787 $ft^2$ =0.318 ft. Biot numbers for projected values of h range from about 0.32 to 0.95. Though the response of the storage wall does not fully meet the guidelines for use of the formula, less error was thought to be associated with this formula than with others available.

The h value desired is that dealing with the surface area in the passages through the wall. Equation 17 deals with an h value over the entire exposed surface, therefore its use is justified by several means. First, the $A_s$ in the formula is only the surface area exposed in the passages. Secondly, $T_\infty$, the temperature of the surroundings, is taken as the average of the calculated air temperature into the wall, which was discussed earlier, and the plenum temperature. This is then the average temperature of the air in the passages. Finally, the plenum temperature is very close to the temperature of the back surface of the wall, therefore little heat is transferred at that surface. Use of these two temperatures to average for $T_\infty$ minimizes the effect of heat transfer from the front surface on the calculated h value. Other heat losses, such as radiation and conduction through the side walls to the room, raise h somewhat through their effects.

To derive an average value of h, the data from each test was studied to determine when the plot of $\ln \frac{T - T_\infty}{T_o - T_\infty}$ versus time was linear during the first 10 to 15 hours. This time period was chosen because that is the length of time available for cooling under true outside conditions. The linear relationship was desired to find an h value that was indeed indicative of the system performance. The values of $\ln \frac{T - T_\infty}{T_o - T_\infty}$ which seemed linear were then analyzed using the Least Squared Method to derive the slope of the line, which was then equal to $-\frac{h\ A_s}{c_p\ \rho\ V}$ in equation (17). Therefore:

$$h = -\ \text{slope}\ \frac{c_p\ \rho\ V}{A_s} \tag{18}$$

The product $\rho V$ is the mass of the storage wall, measured as 589 pounds, while the specific heat of the concrete was assumed to be 0.185 Btu/lb-F. Therefore, $c_p \rho V$ was taken as a constant equal to 108.965 Btu/F for this study. The value for $A_s$ varied with the spacing used, but was a product of the number of passages, 15, the length of the passages, 15.625 inches, and twice the sum of the height of the passage, 3.625 inches, and the spacing between blocks. Substituting these constant values into the calculation of $A_s$ leaves the following relationship:

$$A_s = 468.75\ (3.625 + \text{Spacing})/\ 144\ \text{in}^2/\text{ft}^2 \tag{19}$$

Therefore, the calculations yielded an average h value for each test, along with calculated values for slope of the $\ln \frac{T - T_\infty}{T_o - T_\infty}$ versus time line, the intercept of the line, and a correlation coefficient, R. The intercept is an indication of how well the data fit, because, by the form of the equation, the intercept should be 1.0 for all cases. The equation is really in the form:

$$\ln \frac{T - T_\infty}{T_o - T_\infty} = \ln \text{ (intercept)} - \frac{h\ A_s}{c_p \rho \ \ V} \tag{20}$$

Though there is no quantitative measure for what an acceptable experimental value for the intercept would be, the calculated values seemed to be within reasonable error of 1.0.

Appendix C shows the computer program used. Appendix D lists calculated temperatures and values used in the analysis.

The following is a list of the h values calculated:

Table 3. Average Convective Heat Transfer Coefficients

| Block Spacing inches | Air Flow Rates, Equivalent cubic feet per minute per square foot collector surface: 1 | Average | 2 | Average | 3* | Average |
|---|---|---|---|---|---|---|
| 0.250 | 1.026+<br>0.936<br>1.030 | 0.997 | 1.516<br>1.608<br>1.561 | 1.562 | 1.741<br>1.679<br>1.865 | 1.762 |
| 0.1875 | 1.048<br>1.101<br>1.226 | 1.125 | 1.815<br>1.634<br>1.550 | 1.666 | 2.118<br>1.951<br>2.399 | 2.156 |
| 0.125 | 1.036<br>1.081<br>0.910 | 1.010 | 1.319<br>1.485<br>1.404 | 1.403 | 1.879<br>1.799<br>1.526 | 1.735 |

*Actually 2.8

+Units are Btu/(ft$^2$-hr-F)

Statisical analysis of these values showed that spacing and flow rate both had significant effect on h, tested at the $\alpha = 0.05$ level, see Appendix E for detailed analysis. The effect of the interaction between spacing and flow rate did not show significance at this level. The average values for each combination were compared, using the Least Significant Difference (LSD) test, to find which ones were significantly different from the rest. These were found to be as follows:

Table 4. LSD Comparisons of Average "h" Values

| Spacing | Flow Rate $cfm/ft^2$ collector | Average "h" | |
|---|---|---|---|
| 1/4 | 1 | 0.997 | a* |
| 1/8 | 1 | 1.010 | a |
| 3/16 | 1 | 1.125 | a |
| 1/8 | 2 | 1.403 | b |
| 1/4 | 2 | 1.562 | bc |
| 3/16 | 2 | 1.666 | c |
| 1/8 | 3 | 1.735 | c |
| 1/4 | 3 | 1.762 | c |
| 3/16 | 3 | 2.156 | d |

*like letters indicate values which are not significantly different at the $\alpha = 0.05$ level.

Similar tests done on the averages of each spacing and each flow rate revealed that there was significant difference between each level.

What Table 4 shows is that, at the low flow rates, spacing has little or no significant effect, while the effect is increased with increasing flow rates. Also, it shows that the values for the 1/4 and 1/8 spacing are never significantly different from each other at equal flow rates, while the 3/16 spacing shows increasing significance with increasing flow rate.

The effects of the changes in flow rates are logical when considering the formulas reported on earlier. These formulas have h as a function of the velocity of air through the passage. The velocity is, of course, a direct function of the flow rate. The Reynolds number for each flow rate is the same for each spacing, the Reynolds number being a function of velocity and spacing. When considering the formulas, one could conclude that the h values could actually be the same for each flow rate regardless of spacing, due to the equality of the Reynolds numbers. However, this was not the case.

The tests with 1/4 and 1/8 inch spacings followed this prediction, while the 3/16 inch spacing test did not. Several possibilities exist to explain this behavior, besides the possibility of an error in the tests.

Due to the fact that the storage was heated more for the tests with 1/8 spacings than for the other tests, it was thought that possibly more heat could have been lost through conduction, convection and radiation to the surroundings. To test this hypothesis, an energy balance was made of the model. The means by which the storage can lose heat are by conduction through the sides, top and bottom of the model, convection to the air being moved through the storage, conduction and convection to the incoming air before it enters the storage wall, and radiation to the surroundings.

Conduction can be calculated from "R" values relating to the resistance to heat flow through the components of the model. Where the storage blocks were in contact with the model, R values of 25.05 hr-ft$^2$-F/Btu for the sides and top, and 25.67 hr-ft$^2$-F/Btu for the bottom were calculated. These values assume still air on the outside of the model.

Some air movement was present due to room ventilation, but this was not considered to be significant. If moving air would be considered, the R values would be decreased approximately two percent. Heat flow can then be calculated from the difference between the average storage or plenum temperature and the room temperature.

The h values represent the convection to the air as it moves through the storage. This convected heat can be calculated by first finding the rise in the air temperature as it moves through the passages in the storage wall. Then, by finding the pounds of air moving through the system, the amount of heat needed to cause the temperature rise can be calculated.

Similarly, the temperature rise in the air from the point where it enters the system until it enters the storage can be used to find how much heat was added to the air due to convection from the covers and the front storage surface and due to conduction to the air through the covers. The covers and storage front should be at or above the air temperature surrounding them. Heat is radiated through the transparent covers, between the covers, and from the covers to the storage and plywood cover. Though their mass is quite small, and therefore, their heat storage capacity, the covers can absorb some heat, which is then either convected to the air or thermally stored in the cover.

Radiation loss would occur from the plywood cover over the front, as heat reaching it by radiation and convection from the storage and transparent covers is radiated to the room. Also, some natural convection from this cover and the outside surface of the outside transparent cover would occur.

These loses are very hard to calculate directly, especially since no temperatures for the three covers were recorded. Therefore, in the analysis of the model, these losses were lumped together as the difference between the total heat loss of the storage and the other means of heat loss mentioned earlier. The loss by the storage was calculated using a value of 0.185 Btu/lb-F for the specific heat of the concrete blocks.

In actual applications, one more source of heat loss or gain is present, the result of hygroscopic properties of the concrete storage wall. The heat of absorption for water into concrete is approximately 1300 Btu per pound of water absorbed, this being heat given off as the water is absorbed. Tests reported by Spillman, et al. (1978) showed that one half pound of water could be absorbed per cubic foot of storage when the temperature fell from 120 F to 60 to 70 F in a period of one to two days. Though such temperature extremes are more common for winter heating uses than summer cooling, a 10 F drop in storage temperature could be accompanied by the absorption of 0.0833 pounds of water per cubic foot of storage, or the release of about 108 Btu in addition to other cooling.

The effects of this reaction may be significant under certain conditions. Should a period of cool, humid weather be followed by a hot, dry period, cooling performance may exceed expectations. The storage could absorb a large amount of moisture, increasing the cooling potential of the storage. Then, when hot, dry air is circulated through the storage, the moisture would be transferred to the air from the storage. This would also occur on a smaller scale during most daily cycles, when cool, moist night air would add moisture to the storage, while the warm, dryer daytime air would pick the moisture back up from the storage.

For the tests run indoors, the hygroscopic properties should not have had any significant effect. Relative humidities were fairly constant for all tests, and at low levels between 20 and 30 percent. It was assumed that under these conditions, hygroscopic effects would be minimal, and what effects were present would be relatively constant for all tests made. Therefore, no direct provision was made to analyze this reaction.

Using these assumptions, the contribution of the various means of heat loss relative to the total heat lost by the storage was found to remain basically the same regardless of spacing, though some differences occur between different air flow rates. This difference probably relates to the time involved, as the low flow rate test took about twice as long as the high rate test.

The results at one cfm per square foot air flow rate ranged as follows: heat gained by air as it moved through the model, 45-55% of the total lost by the storage; conduction losses, 15-20%; convection and conduction to the air before it enters the storage wall, 20-25%; radiation and other losses, 10-15%. At two cfm per square foot air flow rate, the contribution of each means of heat loss were 55-60%, 5-15%, 20-25%, 5-10%, respectively. The results for three cfm per square foot were 65-70%, 5-10%, around 25%, and less than 5%, respectively. Though this adds validity to the methods used to collect data and shows an increased effect of the ventilating air, this analysis does little to explain the higher h values found with the 3/16 inch spacing.

Probably the best explanation of these results has to do with the air flow behavior in the passages. As with the formulas for h calculations, the topic of flow patterns has not been very well defined for low Reynolds numbers encountered in these tests. The effects of entrance regions might have some significance, though, using equation 5, the flow development lengths are never much over two inches, and the difference between different spacings at the same flow rate is at most about one inch, or 6.25% of the total flow length. Also, if the entrance region would be a significant variable, the h values for the 1/4 and 1/8 inch spacings should show the greatest discrepancy, when in fact, they were not significantly different.

The action of the air in these passages could be such that the true area available for heat transfer is lower than the area used in the calculations. The effective heat transfer area may be, in fact, only the large surfaces, here the 3.625 by 16 inches area on either side of the passage, and not including the small area of width equal to the spacing at the top and bottom of each passage. A decrease in the effective area would, as shown by equation 18, increase the h value calculated. However, this change still would not explain the trend in the h values calculated.

It is possible that some kind of flow separation may take place, which could lower the h value for 1/8 inch tests. Flow separation may occur in entrance regions and result in little or no air flow next to the surface, and may continue for an indeterminate distance. This would result in poor mixing of the air as it passed through the storage, and therefore, lower heat transfer. The air next to the walls of the passage would heat up quickly, and, due to a smaller temperature gradient, less heat would be convected to that air further along in the passage. The air flowing in a stream in the center of the passage would not show much effect of the heat transfer. One would expect to see this occur more in the larger spaced passages, though, as surface roughness and other considerations would tend to keep the air better mixed in the smaller passage.

The smaller passage, due to its greater restriction of flow, may have less effective area than the larger passages. Perhaps its small dimensioned surfaces are not involved in the heat transfer process, while those of the larger passages are. The literature does not provide any indication of a reaction such as this, so until more specific studies are done on this phenomenon, the conjecture is that the flow pattern is altered by the 1/8 inch spacing, making the 3/16 inch spacing preferable to obtain a maximum value for h.

Summer, 1978

Further tests were made in the summer of 1978 using the full-sized unit discussed earlier. Three complete cycles of operation were monitered for each of two air flow rates: 2.07 and 2.95 cfm per square foot of collector surface. See figures 6 and 7 for graphs of the system's performance.

From the graphs, it is apparent that the higher air flow rate cools the storage more than the lower flow rate. Under similar ambient conditions, the line representing the average storage temperature shows more fluctuation under the higher air flow rate. The plenum temperature, which represents the air temperature entering the building, peaks at a temperature about 10 F below the high ambient temperature of the

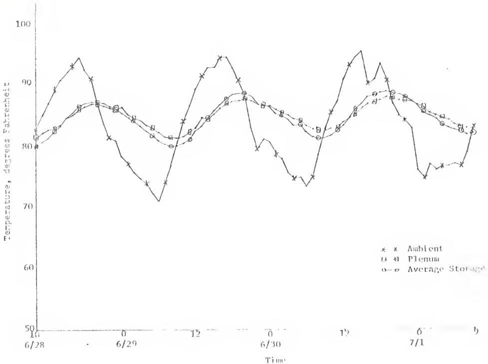



Figure 6. Performance with Continuous Ventilation of 2.95 cfm/ft$^2$ with Prototype Unit at the KSU Swine Experiment Farm, Summer, 1978

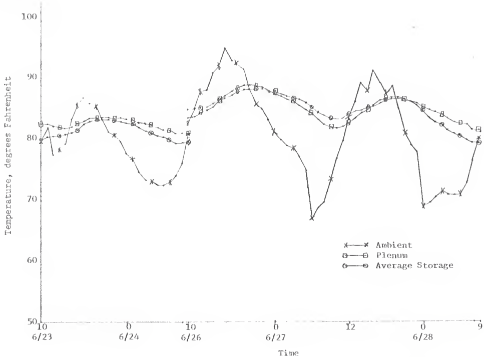



Figure 7. Performance with Continuous Ventilation of 2.07 cfm/Ft$^2$ with Prototype Unit at the KSU Swine Experiment Farm, Summer, 1978

day with air flow rates of 2.95 cfm per square foot. Meanwhile, with the 2.07 cfm per square foot rate, the plenum temperature raises nearer to 5 F of the high ambient temperature. Similarly, the storage temperature lowers to about 10 F of the ambient low with the 2.95 cfm rate, while the storage temperature remains closer to 15 F above the ambient low for the lower air flow rate. In other words, the higher air flow apparently cools the storage more, and therefore provides more cooling to the ventilating air being drawn through it.

To quantitatively analyze the performance under these conditions, some parameter had to be chosen for comparison. The h value used for the model is not easily calculated for this unit because, due to the arrangement of the blocks in the storage wall, the area for heat transfer is not well defined. The wall was laid up containing transverse air spaces, as well as spaces through the wall.

Therefore, an (hA) value was calculated for each test. This is a value of dimension Btu per hour per degree Fahrenheit which comes from the equation

$$Q = h\ A_s\ \Delta t \tag{21}$$

To calculate this value, Q was obtained by calculating the heat necessary to raise the air temperature as it was drawn through the storage. Then, $\Delta$ t was found as the difference between the storage temperature and the average of the temperature of the air as it entered the storage and the plenum temperature. The quantity hA was then simply $Q/\Delta t$.

The hA values calculated were then 3072.2, 1910.8 and 2630.3 (average, 2537.8) for the 2.07 cfm/ft$^2$ air flow and 3041.5, 2963.4 and 2917.0 (average, 2974.0) for 2.95 cfm/ft$^2$. There is quite a spread in the values for the lower air flow. The daily fluctuation in ambient temperature was not as great as the average for the first test, which may partially explain the high value for that test. Due to the limited data and the downward trend in ambient temperature for the tests at the lower air flow rate, the comparisons between the two flow rates may not be very good. However, the data does show how the system would work. The average value for the low air flow rate is 14.67 percent lower than the value for the higher air flow, indicating that the higher air flow does provide for better cooling performance of the system.

Some conclusions may be made here about the system by observing its performance. First, by using this system, the temperature of the ventilating air may be lowered by 10 F from the peak ambient temperature. Even if no other cooling were provided, this drop in temperature could provide a much more comfortable environment inside the building for the swine. Secondly, observing the curves in figures 3 and 6, it seems that, for maximum comfort for sows in the building, when the storage is being cooled at night, ventilating air should be drawn into the building by means other than passing through the storage first. In the spring and fall, when nights may be too cool while the days are too warm for swine comfort, the air may come through the wall at night into the building to provide a more even temperature of the ventilating air. As illustrated by figure 3, the storage wall has a strong moderating effect on the plenum air temperature. During the summer, however, this heat at night is not needed, and so should be flushed to the outside of the building. Also, it should be noted that other means of heat, such as heat lamps, may be needed for baby pigs should the inside temperature drop too low.

Additional storage cooling may be obtained by lowering the outside cover at night to allow more radiation loss from the storage. Though such methods have been used in residential applications, a farmer with a system such as this would probably rather not be bothered with the task of lowering covers at night and raising them in the morning. Therefore, until some means are devised to mechanically raise and lower them, the covers will probably have to remain up during the entire cooling season.

The common method presently used to rate the efficiency of cooling equipment is with a "Coefficient of Performance" (COP), which is the ratio of energy removed by the system to energy used by the system. The main source of energy used by this system is electricity to power the fans. A small amount of energy may be used in actual applications to open and close shutters, but this would be an insignificant amount when compared to the total.

The COP may be maximized by increasing the cooling or by decreasing the energy input. To increase the cooling, either an increase in cooling potential or a decrease in heat gain through the system's structure is desired.

Present units are fairly well insulated on the ends and top, such that increases in thickness may not be economical, especially considering the small area of the ends. The front surface is shielded by the plywood cover from radiation. There is still transfer of heat into the storage through the front, however. This could be decreased, to a degree, by installing temporary insulation panels on the front rather than the plywood covers. This would, probably, result in higher materials costs, increased labor and more maintenance. The added bother and cost would not seem to justify the use of insulating panels.

An increase in cooling potential could be accomplished by the addition of more storage. However, at the present time, the collector unit covers the entire south side of the building. As mentioned earlier, Forbes (1977) provided storage inside the conditioned room. This would necessitate more air being moved, however, as well as occupying space inside the building. The block wall could be made thicker to provide more storage capacity. However, results seem to show that the present 16 inches is already thick enough that the temperature wave does not penetrate the entire wall during operation under normal conditions. More storage capacity, therefore, may not result in more usable storage.

Since the air drawn through the wall to be cooled is the necessary ventilation for the building, the manipulation of air flow to obtain maximum cooling is minimal. Velocities of air will depend on the ventilating rates rather than storage unit performance. It is, however, possible to regulate the cooling of the storage unit so that the storage is cooled as much as possible. This remains the best method to increase the COP due to maximizing the cooling done.

To get the minimum energy input, fan sizing is important. A separate fan is normally used for cooling the storage than is used to provide building ventilation. Thus, this fan can be more carefully sized to provide the air flow needed. The cooling system has been separated from the ventilating system as described by Murphy, et al. (1977). If the systems are not separated, then a variable speed fan could be used to change flow rates for different situations, though a steady flow rate could probably be used. The use of separate systems has been estimated by Spillman (1978) to add about $1 per square foot of collector to the cost of the solar collector-storage system. This cost would not be prohibitively high.

Normally, fans with larger outputs are more efficient than those with less output. Also, variable speed fans have lower efficiencies than single speed fans, and their efficiency goes down when operated below capacity. One manufacturer offers fans with efficiencies ranging from 9.1 to 17.8 cfm per Watt of input. Variable speed fans adjusted below capacity may be even lower.

Also, due to flow patterns and the dynamics of the system, one fan can service about 60 feet of length. If longer, too much air passes through the wall before the flow reaches the ends, preventing uniform air movement. Therefore, for maximum COP, the unit should be designed such that one large, efficient fan, or some multiple of efficient fans, can provide the air flow for the entire unit. This could limit the building to something smaller or larger than desired, however.

As an example of the COP possible with this system, consider a hypothetical collector 60 feet long and 8 feet high. If three cfm per square foot of collector surface area is used to cool the storage, a fan delivering 1440 cfm is needed. An Aerovent VF 145 with a 14 inch propeller will deliver 1570 cfm with power input of 170 Watts. This is more air flow than needed, actually 3.27 cfm per square foot of collector surface, but it is the closest fan size to this situation.

If it is assumed that the cooling period is 12 hours, that the storage heats up 10 F while cooling the ventilating air, and that the storage provides 160 pounds of concrete per square foot of surface with a specific heat of 0.185 But/lb-F, then:

Heat removed from ventilating air = 8 ft x 60 ft x 160 $lb/ft^2$ x 0.185 Btu/lb-F x 10 F/day = 142,080 Btu/day

Energy input = 3413 Btu/Kwhr x 0.17 Kw x 12 hr/day = 6963 Btu/day

$$COP = \frac{142,080}{6963} = 20.4 \tag{22}$$

Another common ratio used to rate efficiencies of cooling equipment is the "Energy Efficiency Ratio" (EER), the ratio of Btu's of cooling to Watt-hours of power input. Here, the EER would be:

$$EER = \frac{142,080}{170 \text{ W} \times 12 \text{ hr}} = 69.65 \tag{23}$$

Conventional cooling equipment normally has COP in the range of 2 to 3 with EER ranging to a maximum of approximately 12 to 14. So, it is obvious that this system is superior in efficiency to conventional systems. As mentioned earlier, a passive solar cooler has a COP greater than 50, therefore, this case is not passive. Changes could be made in the system to improve the COP, but values greater than 35 would be hard to achieve with this system. Still, the system is much more efficient than conventional units.

# CONCLUSIONS

Several conclusions may be drawn from the previous discussion. First, the nocturnal cooling of a solar collector-storage system is feasible to provide a more comfortable environment inside a swine building. The research showed that ventilating air temperature may be reduced up to 10 F below the ambient temperature. Also, this cooling can be obtained much more economically than by conventional methods, with a Coefficient of Performance of 20 to more, compared to conventional systems with values of two to three.

Secondly, in the fall and spring, such a system can provide cooling during the day when the ambient temperature rises above comfort levels for the sows inside the building, as well as add heat to the ventilating air at night when ambient temperatures drop below the comfort level. This capability, coupled with summertime cooling, should help a producer decide whether to construct such a system, and how to manage it once constructed.

Thirdly, the optimal combination of flow rate and block spacing was found to be three cubic feet per minute per square foot of collector surface and 3/16 inch, respectively. Though an exact reason was not found for the superiority of the 3/16 inch spacing, the performance in the test showed it to be superior to either 1/4 or 1/8 inch spacings. Closely monitored tests were made with the model with air flow rates of one, two and three cfm per square foot, and increasing the flow rate each time was found to give better performance. As only a limited time is available to cool the storage each night, the flow rate must provide that cooling in a 10 to 12 hour period. Though tests with high air flow rates were not conducted with the model, preliminary indications using the full-sized unit were that air flow rates as high as six cfm per square foot did not increase the cooling performance beyond that at three cfm per square foot.

Finally, inspection of graphs shows that a separate system should be installed to cool the storage such that the air is flushed to the outside after picking up heat from the storage, rather than being drawn into the building all night. This will prevent heated air entering the building when the nighttime ambient temperature does not go below comfort conditions for the sows. Though some cost is added for additional controls and equipment, sow comfort would be increased by their use.

# SUMMARY

This study dealt with the performance of a solar collector-storage system under summertime conditions when it is desired for the storage to be at a temperature below ambient conditions so that ventilating air is cooled as it is drawn through the storage into the building. It was desired to know if such nocturnal cooling would be feasible, both economically and practically. Also, the performance of the system during fall and spring when days are warm and nights are cool was researched. Finally, an optimal combination of air flow rate through the storage and spacing between concrete blocks in the storage wall was found.

Tests were made in a unit comprised of a dry stacked concrete block wall, 50 by 8 feet. Two transparent covers and a plywood cover covered the outside of the storage wall. Air flow was supplied by a fan located at one end of the space between the block wall and the existing building wall. Preliminary tests showed that the temperature of the ventilating air could be cooled about 10 F below the high ambient temperature. Little difference was found between performance for air flow rates of three, five and six cubic feet per square foot of collector.

Further tests were conducted using a small model representing a section of storage wall two feet square. Using the previous study as a guide, air flow rates of one, two and three cfm per square foot and spacings of 1/8, 3/16 and 1/4 inch were monitored through three cooling cycles for each of the nine combinations of air flow rate and spacing tested. Air flow rates were set such that the velocity through the wall with 1/4 inch spacing was the same as the velocity through the full-sized unit with one, two and three cfm per square foot. The same total air flow was then used for the other spacings.

It was desired to calculate an average value for the heat transfer coefficient, h, for each test. Existing relationships to calculate h under forced flow deal with greater flow rates and velocities than encountered here. Also, the small dimensions of the width of the passages creates a situation not well researched. Therefore a method was used which was developed to analyze a billet placed in a different environment.

With this method of analysis, it was found that the highest h value was for the 3/16 inch spacing with three cfm per square foot flow rate. A statistical analysis showed that for all air flow rates, the 1/8 and 1/4 inch spacings performed without significant difference from each other. The reasons that the 3/16 inch spacing was found to be superior were not clear, but was thought to relate to different flow characteristics through the wall.

Following these results, further studies were conducted with the full-sized unit, which was constructed with nominal 3/16 inch spacings. Studying the performance with air flow rates of two and three cfm per square foot, the higher air flow rate was again found to be superior. Because of the method of construction, the area available for heat transfer was not well defined, therefore, rather than calculate an h value, a value for the product of h and the heat transfer surface area was calculated. This value was found to be 14.7 percent higher for the higher flow rate.

Inspection of graphs of the system's performance showed that at three cfm per square foot, the storage cooled to within about 10 F of the low ambient temperature at night, while the air temperature into the building was lowered about 10 F from the peak ambient temperature during the day. At the lower flow rate, the storage cooled only to 10-15 F of the low ambient temperature while the ventilating air temperature was lowered only about 5 F from the high ambient temperature.

The conclusions drawn from these results are that the combination of 3/16 inch spacing with three cfm per square foot air flow was optimal. The system produced cooling which can help maintain comfort levels for swine with a Coefficient of Performance of 20 or more, much higher than conventional cooling equipment. The use of this system in the fall and spring could help provide ventilating air with less temperature fluctuation than ambient conditions, providing cooling during warm days and heating during cool nights. Such uses can expand the time that the system can be used, thus offsetting some added expense and management which the system requires.

## SUGGESTIONS FOR FUTURE RESEARCH

Several aspects of this topic could be researched to a greater extent.

First, more research is needed to better show the effect of block spacing on cooling. Though this study showed that a 3/16 inch spacing was best, the theory involved did not substantiate this outcome. A more controlled test is needed to better solve this problem.

Also, in conjunction with the above, possibly a better means of calculating the convective heat transfer coefficient could be found. Some simplifying assumptions made in this study may have affected the accuracy of the calculations. By using a more controlled study, perhaps better values could be obtained.

Some study could also be made of the heating properties of the block storage wall. This study dealt only with the reaction of the wall as it was cooled. Though similar results with various air flow rates and spacings would be expected, verification of the results would be informative.

Appendix A. Cooling Load for Farrowing House as Outlined by Bond, et al. (1959)

Construction details: 30 sow farrowing house, with the length dimension on the east-west axis.

Walls: Steel siding, 2 x 4 studs, $3\frac{1}{2}$ inches batt fiberglas insulation, 3/4 inch plywood interior.
$R = 11$
$U = 1/R = 0.0909$ Btu/(hr-ft$^2$-F)

Ceiling and roof: $R = 19$, $U = 0.0526$ Btu/(hr-ft$^2$-F)

Inside dimensions: 80 x 20 x 8 feet

Inside design conditions: 70 F, 50% relative humidity

Outside design conditions: 95 F, 40% relative humidity from ASHRAE (1977)

Average daily sol-air temperatures, from ASHRAE (1977)

North wall: 86 F
South wall: 89
East wall: 91
West wall: 91
Horizontal surface: 91

Heat gain North wall: 80 x 8 x 0.0909 x (86 - 70) = 931 Btu/hr

| | |
|---|---|
| Similarly, East wall: | 305 |
| West wall: | 305 |
| South wall: | 1105 |
| Roof: | 1767 |
| Total construction heat gain: | 4413 Btu/hr |

Sensible animal load: 1000 But/hr per sow and litter
1000 x 30 = 30,000 Btu/hr

Sensible heat gain from 30, 200 Watt heat lamps for pigs: 20,500 Btu/hr

Room sensible load = 4413 + 30,000 + 20,500 = 54913 Btu/hr

Animal latent load: About 1000 Btu/hr/sow and litter
1000 x 30 = 30,000 Btu/hr

Heat gain from ventilating air: ventilating at 100 cfm/sow and litter:
100 x 30 = 3000 cfm = 180,000 cfh

Outside conditions: $h_e = 38.4$ Btu/pound dry air
$v = 14.26$ ft$^3$/pound dry air

Inside conditions: $h_e = 25.4$ Btu/pound dry air

Appendix A. continued

$$\frac{180000\ ft^3/hr}{14.26\ ft^3/lb\ d.a.} = 12623 \text{ pounds dry air/hr}$$

$$12623\ lb\ d.a./hr \times (38.4 - 25.4)\ Btu/lb\ d.a. = 164{,}100\ Btu/hr$$

Therefore, total cooling load = 54913 + 30,000 + 164,100 = 249013 Btu/hr

## Appendix B. Derivation of Equation to Estimate Air Temperature Entering Storage Wall

Data used to find the relationship for estimating air temperature entering the passages through the storage wall:

Table 5. Temperatures for Air Flow Rate of 3 cfm/ft$^2$, 3/16" Spacing

| TF* Front Block Surface | TA Ambient | TC Air Entering System | TT Air Between Blocks and Inside Cover, Top | TIN Air Between Blocks and Inside Cover, Bottom | T1 Air Into Wall Passages |
|---|---|---|---|---|---|
| 91.49 F | 75.70 F | 83.02 F | 85.70 F | 85.61 F | 88.01 F |
| 88.88 | 75.44 | 78.05 | 81.09 | 80.21 | 84.34 |
| 87.27 | 78.40 | 79.19 | 81.22 | 80.25 | 83.81 |
| 86.09 | 80.69 | 80.60 | 81.75 | 80.96 | 83.55 |
| 85.39 | 80.47 | 80.65 | 81.66 | 80.91 | 83.20 |
| 84.86 | 81.75 | 81.31 | 82.01 | 81.35 | 83.20 |
| 84.43 | 81.22 | 80.82 | 81.79 | 81.18 | 82.98 |
| 84.12 | 79.50 | 80.03 | 81.40 | 80.56 | 82.63 |
| 83.77 | 79.46 | 80.25 | 81.22 | 80.47 | 82.32 |
| 83.59 | 78.13 | 79.59 | 81.09 | 80.16 | 82.10 |
| 82.85 | 75.75 | 77.25 | 78.53 | 77.87 | 80.25 |
| 82.10 | 75.04 | 75.97 | 77.38 | 76.81 | 79.28 |
| 81.62 | 75.35 | 76.19 | 77.47 | 76.90 | 79.19 |
| 81.04 | 75.08 | 75.57 | 76.76 | 76.14 | 78.49 |
| 80.60 | 74.73 | 75.48 | 76.45 | 75.83 | 78.13 |
| 80.12 | 74.95 | 75.17 | 76.23 | 75.61 | 77.83 |
| 79.72 | 74.24 | 75.13 | 75.97 | 75.39 | 77.43 |
| 79.37 | 74.24 | 74.50 | 75.66 | 75.08 | 77.16 |

*Intial names of columns refer to the names used in computer analysis for these temperatures and in the following discussion.

Appendix B. continued

Table 6. Temperatures for Air Flow Rate of 2 cfm/ft$^2$, 3/16" Spacing

| TF Front Block Surface | TA Ambient | TC Air Entering System | TT Air Between Blocks and Inside Cover, Top | TIN Air Between Blocks and Inside Cover, Bottom | T1 Air Into Wall Passages |
|---|---|---|---|---|---|
| 84.91 F | 72.15 F | 78.49 F | 81.13 F | 81.75 F | 82.63 F |
| 83.38 | 72.37 | 74.90 | 77.12 | 76.99 | 79.63 |
| 81.44 | 72.37 | 73.62 | 75.61 | 75.04 | 78.09 |
| 80.38 | 72.37 | 73.62 | 75.04 | 74.33 | 77.34 |
| 79.46 | 74.28 | 74.19 | 75.35 | 74.59 | 77.29 |

After some inspection and trial calculations, it was decided to obtain a relationship between the quantities (T1 - TC) and (TF - TC). It was felt that the value for T1 was a function of both TC and TF, and by using temperature differences a good approximation for T1 could be found. Plots of (TF - TC) versus (T1 - TC) appeared fairly linear, except for the first point obtained with the two cfm/ft$^2$ air flow rate. This point was considered to be inaccurate due to recording procedures, so it was not used for the analysis. The following points were used:

Appendix B. continued

Table 7. Points Used in Derivation of Equation 18.

| | TF - TC | T1 - TC |
|---|---|---|
| 2 $cfm/ft^2$ | 8.48 F | 4.73 F |
| | 7.82 | 4.47 |
| | 6.76 | 3.72 |
| | 5.27 | 3.10 |
| 3 $cfm/ft^2$ | 8.47 | 4.99 |
| | 10.83 | 6.29 |
| | 8.08 | 4.62 |
| | 5.49 | 2.95 |
| | 4.74 | 2.55 |
| | 3.55 | 1.89 |
| | 3.61 | 2.16 |
| | 4.09 | 2.60 |
| | 3.52 | 2.07 |
| | 4.00 | 2.51 |
| | 5.60 | 3.00 |
| | 6.13 | 3.31 |
| | 5.43 | 3.00 |
| | 5.47 | 2.92 |
| | 5.12 | 2.65 |
| | 4.95 | 2.66 |
| | 4.59 | 2.30 |
| | 4.87 | 2.66 |

Using the Least Square Method to obtain an equation for the line, where (TF - TC) = X and (T1 - TC) = Y:

$\Sigma X = 126.87$ $\quad \Sigma X^2 = 808.05$ $\quad \overline{X} = 5.77$
$\Sigma Y = 71.15$ $\quad \Sigma Y^2 = 256.34$ $\quad \overline{Y} = 3.23$
$\Sigma XY = 454.61$ $\quad n = 22$

$$\text{slope:} \quad b = \frac{454.61 - \frac{(126.87)(71.15)}{22}}{808.05 - \frac{(126.87)^2}{22}} = .5797$$

intercept: $a = \overline{Y} - b\overline{X} = 3.23 - (.5797)\ 5.77 = -.1152$

$R = .9894$, where 1 is perfect fit

Appendix B. continued

therefore, the equation of the line is:

$$Y = -0.1152 + 0.5797\ X \qquad (18)$$

where $X = TF - TC$
$Y = T1 - TC$

With TF and TC known, T1 can then be calculated from this relationship.

Appendix C. Computer Program Used in Analyzing Oata from Model Tests, Winter, 1977-1978

```
      DIMENSION TA(100),TF(100),T4(100),T8(100),T12(100),TB(100),
     ,TP(100),TN(100),TC(100),TIN(100),NOHR(100),ABT(100),AT(100),
     ,Y(100),X(50),SUMY(50),SUMX(50),SUMXY(50),SUMXX(50),SUMYY(50),
     ,SLOPE(50),H(50),SPAC(50),A(50),R(50),T1(50),DELP(100),TOP(100),
     ,PB(100),AREA(100),BOTM(100),FLOW(100)
      INTEGER HOUR(50)
C NOHR=NUMBER OF HOURS/RUN, DELP=STATIC PRESSURE ACROSS NOZZLE,IN.H2O,
C TOP=DEW POINT, PB=BAROMETRIC PRESSURE,AREA=AREA OF OPENINGS THRU WALL,
C SPAC=SPACING BETWEEN BLOCKS. 5.198 CONVERTS IN.H2O TO P/SQ.FT.,
C 0.491 CONVERTS IN.HG.TO PSI
C TF,T4,T8,T12,TB ARE BLOCK TEMPS, TA=AMBIENT TEMP,TP=PLENUM TEMP,TN=TEMP AT
C NOZZLE, TC=AIR TEMP BETWEEN COVERS, TIN= AIR TEMP BETWEEN COVER AND BLOCKS
C ABT= AVG BLOCK TEMP, AAT=AVG.AIR TEMP, ATIN= AVG AIR TEMP ENTERING BLOCKS,
C DELTA= DIFF IN TEMP OF AIR IN TO AIR IN PLENUM, DELBA=DIFF TEMP ABT TO AAT
C ATF= AVG TEMP BLOCK FRONT
      DO 1 J=1,27
      READ,NOHR(J),DELP(J),TOP(J),PB(J),AREA(J),SPAC(J)
      P=DELP(J)*5.198
      PS=EXP(54.633-(12301.7/(TOP(J)+459.67))-5.169*ALOG(TOP(J)+459.67))
      W=0.622*PS/((PB(J)*0.491)-PS)
      SUMY(J)=0.0
      SUMX(J)=0.0
      SUMXY(J)=0.0
      SUMXX(J)=0.0
      SUMYY(J)=0.0
      KK=NOHR(J)
      DO 2 I=1,KK
      IF(J.GT.13) GO TO 102
      READ(5,13) HOUR(I),TF(I),T4(I),T8(I),T12(I),TB(I),TA(I),TP(I),
     -TN(I),TC(I),TIN(I)
   13 FORMAT(I2,10F6.3)
      TF(I)=31.991634+46.307766*TF(I)-1.3819562*TF(I)*TF(I)+0.054120312
     ,*TF(I)*TF(I)*TF(I)
      T4(I)=31.991634+46.307766*T4(I)-1.3819562*T4(I)*T4(I)+0.054120312
     ,*T4(I)*T4(I)*T4(I)
      T8(I)=31.991634+46.307766*T8(I)-1.3819562*T8(I)*T8(I)+0.054120312
     ,*T8(I)*T8(I)*T8(I)
      TB(I)=31.991634+46.307766*TB(I)-1.3819562*TB(I)*TB(I)+0.054120312
     ,*TB(I)*TB(I)*TB(I)
      TP(I)=31.991634+46.307766*TP(I)-1.3819562*TP(I)*TP(I)+0.054120312
     ,*TP(I)*TP(I)*TP(I)
      TC(I)=31.991634+46.307766*TC(I)-1.3819562*TC(I)*TC(I)+0.054120312
     ,*TC(I)*TC(I)*TC(I)
      T12(I)=31.991634+46.307766*T12(I)-1.3819562*T12(I)*T12(I)+
     ,0.054120312*T12(I)*T12(I)*T12(I)
      TIN(I)=31.991634+46.307766*TIN(I)-1.3819562*TIN(I)*TIN(I)+
     ,0.054120312*TIN(I)*TIN(I)*TIN(I)
      TA(I)=31.991634+46.307766*TA(I)-1.3819562*TA(I)*TA(I)+0.054120312
     ,*TA(I)*TA(I)*TA(I)
      TN(I)=31.991634+46.307766*TN(I)-1.3819562*TN(I)*TN(I)+0.054120312
     ,*TN(I)*TN(I)*TN(I)
      ABT(I)=(0.5*(TF(I)+TB(I))+T4(I)+T8(I)+T12(I))*0.25
      INT=TF(I)-TC(I)
      XXT=(0.5797*INT)-0.1152
      T1(I)= XXT+TC(I)
      AT(I)=(T1(I)+TP(I))*0.5
      TOP=ABT(I)-AT(I)
      BOTTOM=ABT(I)-AT(I)
```

Appendix C. Continued

```
      Y(I)=(ALOG(ABS(TOP/BCTTCM)))
      IF(HOUR(I).EQ.)) GO TO 2
      X(I)=HOUR(I)
      SUMY(J)=SUMY(J)+Y(I)
      SUMX(J)=SUMX(J)+X(I)
      SUMXY(J)=SUMXY(J)+X(I)*Y(I)
      SUMXX(J)=SUMXX(J)+X(I)*X(I)
      SUMYY(J)=SUMYY(J)+Y(I)*Y(()
C V= SPECIFIC VOLUME, C=NCZZLE CISCHARGE CCEF, ODTM=MASS FLOW THRU NOZZLE,
C 200.32= 359*F*D*C*FA*YA, CN=C CALCULATED FRCM TRIAL C, VL=VELCCITY THRU
C NOZZLE, RE=REYNCLDS NC. FOR NCZZLE,=VEL*D/MU, 0.003C68= AREA CF NOZZLE,
      V=((TN(()+TN(I-1))*0.5+459.67)*(53.35+W*85.78)/(PB(J)*70.704)
C 70.704 CONVERTS IN. HG. TO PSF.
      C=0.950
    4 DOTM(I)=C*200.32*SQRT(DELP(J)/V)
      FLCW(I)=DCTM(I)*V/60.
      VL=FLCW(I)*60./0.003068
      RE=VL*0.75/(12.*V*0.044)
      ALN=ALOG(RE)
      CN=0.19436+0.152884*ALN-0.0097785*ALN*ALN+0.00020903*ALN*ALN*ALN
      IF(ABS(CN-C).LT.0.002) GO TC 3
      C=CN
      GO TO 4
    3 CCNT(NUE
    2 CONTINUE
      YBAR=SUMY(J)/(KK-1)
      XBAR=SUMX(J)/(KK-1)
      TOP2=SUMXY(J)-(SUMX(J)*SUMY(J)/(KK-1))
      BOTOM=SUMXX(J)-(SUMX(J)*SUMX(J)/(KK-1))
      SLOPE(J)=TCP2/BOTOM
      A(J)=EXP(YBAR-SLCPE(J)*XBAR)
      BTTM=SQRT(BOTOM*(SUMYY(J)-(SUMY(J)*SUMY(J)/(KK-1))))
      R(J)=TOP2/BTTM
      AREA2=468.75*(3.625+SPAC(J))/144.
      H(J)=108.965*ABS(SLOPE(J))/AREA2
      WRITE(6.100) DELF(J),TCP(J),PE(J)
  100 FORMAT('0',2X,'PRESSURE DROP ACROSS NOZZLE=',F7.4,', DEWPCINT TEMP
     -ERATURE= ',F5.2,', BARCMETRIC PRESSURE= ',F5.2)
      WRITF(6,30)
   30 FORMAT('0',2X,'HOUR',2X,'TA',5X,'TF',5X,'TB',5X,'T4',5X,'T8',5X,
     ,'T12',4X,'ABT',4X,'TP',5X,'TN',5X,'TC',5X,'TIN',5X,'T1',6X,'T',
     ,4X,'ABT-T/ABT(1)-T')
      DO 5 I=I,KK
      WRITE(6,40) HOUR(I),TA(I),TF(I),T8(I),T4(I),TB(I),T12(I),ABT(I),
     ,TP(I),TN(I),TC(I),TIN(I),T1(I),AT(I),Y(I)
   40 FORMAT(' ',3X,I2,13F7.2,6X,F6.3)
    5 CONTINUE
      WRITE(6,99)
   99 FORMAT('0',2X,'SPACE',6X,'"R"',6X,'INTERCEPT',4X,'SLCPE',4X,'H')
      WRITE(6,97) SPAC(J),R(J).A(J),SLOPE(J),H(J)
   97 FCRMAT(' ',2X,F6.4,5X,F6.3,7X,F6.3,4X,F6.3,2X,F5.3)
    I CONT(NUE
      STOP
      END
```

## Appendix D. Data Used in "h" Calculations

Table 8. Data for "h" and Flow Calculation, 1/8" Spacing, 1 cfm/ft$^2$ Ventilation

PRESSURE DROP ACROSS NOZZLE= 0.1970, DEWPOINT TEMPERATURE= 45.50, BAROMETRIC PRESSURE= 28.99

| HOUR | TA | TF | T8 | T4 | T8 | T12 | ABT | TP | TN | TC | TIN | T1 | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 77.83 | 97.21 | 93.71 | 96.30 | 95.09 | 94.40 | 95.31 | 95.74 | 91.19 | 92.71 | 98.54 | 94.91 | 95.33 | 0.000 |
| 1 | 79.77 | 92.54 | 93.75 | 95.18 | 95.05 | 94.53 | 94.48 | 93.92 | 90.10 | 79.68 | 83.24 | 86.52 | 90.22 | -0.180 |
| 2 | 79.68 | 90.06 | 93.84 | 93.32 | 94.53 | 94.57 | 93.59 | 93.19 | 90.23 | 78.80 | 80.82 | 85.36 | 85.12 | -0.326 |
| 3 | 73.35 | 88.06 | 93.71 | 91.58 | 93.49 | 94.14 | 92.52 | 92.45 | 88.84 | 75.13 | 78.71 | 81.97 | 87.21 | -0.422 |
| 4 | 74.46 | 87.27 | 93.79 | 90.49 | 92.84 | 94.05 | 91.98 | 92.02 | 89.10 | 76.06 | 79.15 | 82.32 | 87.17 | -0.526 |
| 5 | 79.33 | 86.05 | 93.19 | 89.19 | 91.80 | 93.36 | 90.99 | 91.28 | 89.01 | 77.60 | 78.62 | 82.13 | 86.70 | -0.657 |
| 6 | 78.88 | 85.26 | 92.67 | 88.19 | 90.84 | 92.71 | 90.18 | 90.63 | 88.71 | 77.29 | 78.40 | 81.24 | 85.93 | -0.793 |
| 7 | 79.11 | 84.56 | 91.97 | 87.27 | 89.93 | 91.93 | 89.35 | 89.75 | 88.27 | 77.74 | 78.53 | 81.10 | 85.43 | -0.925 |
| 8 | 79.02 | 84.03 | 91.28 | 86.53 | 89.15 | 91.15 | 88.62 | 89.01 | 87.62 | 77.60 | 78.31 | 80.97 | 84.99 | -1.046 |
| 9 | 78.84 | 83.46 | 90.41 | 85.74 | 88.27 | 90.28 | 87.81 | 88.19 | 86.88 | 77.60 | 78.22 | 80.35 | 84.29 | -1.142 |
| 10 | 78.58 | 82.98 | 89.62 | 85.08 | 87.49 | 89.49 | 87.09 | 87.40 | 86.31 | 77.65 | 78.18 | 80.43 | 83.92 | -1.278 |
| 11 | 78.14 | 82.58 | 88.93 | 84.51 | 86.83 | 88.80 | 86.47 | 86.79 | 85.65 | 77.65 | 78.27 | 79.85 | 83.32 | -1.336 |
| 12 | 78.18 | 82.23 | 88.23 | 84.03 | 86.22 | 88.10 | 85.90 | 86.18 | 85.21 | 77.60 | 78.31 | 79.81 | 82.99 | -1.446 |

| SPACE | "R" | INTERCEPT | SLOPE | H |
|---|---|---|---|---|
| 0.1250 | -0.998 | 0.917 | -0.116 | 1.038 |

Table 9. Data for "h" and Flow Calculation, 1/8" Spacing, 1 cfm/ft$^2$ Ventilation

PRESSURE DROP ACROSS NOZZLE= 0.1890, DEWPOINT TEMPERATURE= 46.00, BAROMETRIC PRESSURE= 28.78

| HOUR | TA | TF | T8 | T4 | T8 | T12 | ABT | TP | TN | TC | TIN | T1 | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 75.30 | 100.44 | 99.36 | 102.88 | 101.98 | 100.44 | 101.30 | 100.95 | 95.74 | 84.21 | 88.93 | 93.37 | 97.16 | 0.000 |
| 1 | 77.96 | 97.21 | 99.71 | 100.65 | 101.55 | 100.74 | 100.35 | 100.35 | 95.65 | 84.03 | 86.40 | 91.45 | 95.90 | -0.194 |
| 2 | 76.28 | 95.31 | 100.05 | 98.59 | 100.44 | 100.70 | 99.35 | 99.58 | 95.35 | 84.51 | 86.92 | 90.20 | 94.89 | -0.363 |
| 3 | 75.35 | 93.92 | 100.18 | 97.21 | 99.58 | 100.61 | 98.61 | 99.06 | 95.78 | 83.02 | 85.65 | 88.70 | 93.88 | -0.450 |
| 4 | 74.28 | 92.62 | 99.84 | 95.83 | 98.46 | 99.96 | 97.62 | 98.20 | 94.31 | 82.41 | 85.13 | 88.05 | 93.14 | -0.600 |
| 5 | 74.33 | 91.97 | 99.28 | 94.83 | 97.60 | 99.41 | 96.86 | 97.47 | 93.84 | 82.80 | 85.13 | 87.90 | 92.69 | -0.723 |
| 6 | 73.79 | 90.84 | 98.42 | 93.66 | 96.35 | 98.37 | 95.75 | 96.26 | 93.23 | 82.06 | 84.38 | 86.58 | 91.42 | -0.824 |
| 7 | 77.78 | 90.19 | 97.55 | 92.75 | 95.44 | 97.51 | 94.89 | 95.44 | 92.67 | 82.58 | 84.43 | 86.92 | 91.18 | -1.003 |
| 8 | 79.33 | 89.58 | 96.73 | 91.97 | 94.57 | 96.65 | 94.09 | 94.44 | 92.36 | 83.46 | 84.47 | 86.82 | 90.63 | -1.128 |
| 9 | 79.19 | 88.97 | 95.91 | 91.15 | 93.71 | 95.78 | 93.27 | 93.66 | 91.80 | 83.72 | 84.60 | 86.51 | 90.09 | -1.259 |
| 10 | 77.74 | 88.58 | 95.14 | 90.54 | 93.01 | 95.01 | 92.60 | 92.88 | 91.10 | 83.81 | 84.65 | 86.02 | 89.45 | -1.324 |
| 11 | 75.08 | 88.19 | 94.31 | 89.93 | 92.23 | 94.23 | 91.91 | 92.23 | 90.15 | 83.55 | 84.60 | 85.75 | 88.99 | -1.440 |
| 12 | 75.44 | 87.88 | 93.53 | 89.41 | 91.54 | 93.45 | 91.27 | 91.41 | 89.54 | 83.55 | 84.60 | 85.75 | 88.58 | -1.552 |
| 13 | 77.87 | 87.49 | 92.80 | 88.88 | 90.89 | 92.71 | 90.66 | 90.80 | 88.97 | 84.16 | 84.69 | 85.79 | 88.29 | -1.706 |
| 14 | 77.65 | 87.23 | 92.10 | 88.49 | 90.36 | 92.06 | 90.14 | 90.28 | 88.40 | 84.43 | 84.78 | 85.47 | 87.87 | -1.777 |
| 15 | 77.34 | 87.01 | 91.49 | 88.19 | 89.84 | 91.45 | 89.68 | 89.71 | 87.84 | 83.94 | 84.38 | 85.57 | 87.64 | -1.900 |

| SPACE | "R" | INTERCEPT | SLOPE | H |
|---|---|---|---|---|
| 0.1250 | -0.999 | 0.893 | -0.121 | 1.081 |

Table 10. Data for "h" and Flow Calculation. 1/8" Spacing. 1 cfm/ft$^2$ Ventilation

PRESSURE DROP ACROSS NOZZLE= 0.1850, DEWPOINT TEMPERATURE= 40.50, BAROMETRIC PRESSURE= 28.70

| HOUR | TA | TF | T8 | T4 | T8 | T12 | ABT | TP | TN | TC | TIN | T1 | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 77.87 | 104.30 | 102.97 | 107.11 | 105.89 | 104.17 | 105.20 | 104.30 | 98.93 | 86.53 | 90.97 | 96.27 | 100.28 | 0.000 |
| 1 | 74.95 | 101.82 | 103.65 | 104.77 | 105.53 | 104.77 | 104.33 | 103.61 | 97.38 | 84.38 | 80.10 | 93.54 | 98.58 | -0.141 |
| 2 | 77.47 | 98.67 | 103.74 | 102.45 | 104.42 | 104.59 | 103.17 | 102.88 | 98.29 | 85.83 | 88.40 | 92.67 | 97.76 | -0.319 |
| 3 | 77.29 | 96.91 | 103.70 | 100.70 | 103.23 | 104.21 | 102.11 | 102.24 | 97.77 | 85.56 | 87.71 | 91.83 | 97.03 | -0.475 |
| 4 | 77.12 | 95.65 | 103.35 | 99.19 | 102.07 | 103.65 | 101.10 | 101.38 | 97.94 | 85.35 | 87.31 | 91.03 | 96.20 | -0.607 |
| 5 | 77.87 | 94.70 | 102.71 | 97.90 | 100.91 | 102.84 | 100.09 | 100.57 | 97.51 | 85.70 | 87.31 | 90.80 | 95.68 | -0.770 |
| 6 | 78.09 | 93.71 | 101.85 | 96.69 | 99.71 | 101.81 | 99.00 | 99.41 | 96.82 | 86.00 | 87.36 | 89.94 | 94.68 | -0.850 |
| 7 | 77.65 | 93.06 | 100.95 | 95.70 | 98.67 | 100.91 | 98.07 | 98.54 | 96.13 | 86.22 | 87.49 | 89.58 | 94.06 | -1.022 |
| 8 | 73.13 | 92.45 | 100.22 | 95.01 | 97.90 | 100.14 | 97.34 | 97.64 | 94.88 | 84.86 | 86.88 | 88.81 | 93.22 | -1.067 |
| 9 | 75.21 | 91.71 | 99.19 | 94.14 | 96.86 | 99.06 | 96.38 | 96.56 | 93.53 | 84.38 | 86.13 | 88.32 | 92.44 | -1.176 |
| 10 | 77.43 | 91.19 | 98.24 | 93.40 | 96.00 | 98.11 | 95.56 | 95.74 | 93.40 | 85.65 | 86.61 | 88.44 | 92.09 | -1.329 |
| 11 | 77.65 | 90.76 | 97.38 | 92.75 | 95.18 | 97.25 | 94.81 | 95.01 | 92.54 | 85.65 | 86.61 | 88.44 | 91.72 | -1.472 |
| 12 | 77.12 | 90.19 | 96.52 | 92.15 | 94.44 | 96.39 | 94.08 | 94.23 | 91.84 | 85.04 | 86.05 | 87.62 | 91.02 | -1.533 |
| 13 | 77.21 | 89.89 | 95.65 | 91.58 | 93.71 | 95.61 | 93.42 | 93.45 | 91.23 | 85.26 | 86.22 | 87.46 | 90.45 | -1.605 |
| 14 | 76.41 | 89.45 | 95.09 | 91.19 | 93.19 | 94.96 | 92.90 | 92.93 | 90.36 | 84.78 | 85.65 | 86.98 | 69.95 | -1.643 |
| 15 | 76.94 | 89.15 | 94.23 | 90.63 | 92.49 | 94.18 | 92.25 | 92.15 | 89.97 | 85.04 | 85.91 | 87.24 | 89.69 | -1.804 |
| 16 | 76.72 | 89.88 | 93.58 | 90.23 | 91.97 | 93.58 | 91.75 | 91.67 | 89.80 | 85.08 | 85.67 | 86.71 | 89.19 | -1.831 |
| 17 | 76.81 | 88.58 | 93.01 | 89.84 | 91.45 | 92.97 | 91.26 | 91.23 | 89.41 | 85.13 | 85.83 | 86.75 | 68.99 | -1.964 |
| 18 | 76.59 | 88.32 | 92.36 | 89.45 | 90.97 | 92.41 | 90.79 | 90.67 | 88.71 | 65.13 | 85.74 | 86.75 | 88.71 | -2.069 |
| 19 | 77.29 | 88.32 | 92.06 | 89.32 | 90.76 | 92.10 | 90.59 | 90.49 | 88.58 | 85.35 | 85.91 | 86.39 | 88.44 | -2.054 |
| 20 | 77.34 | 88.14 | 91.67 | 89.10 | 90.36 | 91.71 | 90.27 | 90.19 | 88.32 | 85.52 | 86.00 | 86.56 | 68.38 | -2.184 |
| 21 | 76.63 | 87.84 | 91.06 | 88.67 | 89.89 | 91.10 | 89.78 | 89.67 | 87.92 | 85.35 | 85.83 | 86.39 | 88.03 | -2.285 |
| 22 | 77.43 | 87.62 | 90.54 | 88.36 | 89.45 | 90.63 | 85.38 | 89.32 | 87.40 | 85.35 | 85.74 | 86.39 | 87.65 | -2.432 |

| SPACE | "R" | INTERCEPT | SLOPE | H |
|---|---|---|---|---|
| 0.1250 | -0.993 | 0.801 | -0.102 | 0.910 |

Table 11. Data for "h" and Flow Calculation, 1/8" Spacing, 2 cfm/ft$^2$ Ventilation

PRESSURE DROP ACROSS NOZZLE= 0.7460, DEWPOINT TEMPERATURE= 37.50, BAROMETRIC PRESSURE= 28.95

| HOUR | TA | TF | T8 | T4 | T8 | T12 | ABT | TP | TN | TC | TIN | T1 | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 73.79 | 108.82 | 107.67 | 111.41 | 110.01 | 108.77 | 105.61 | 107.67 | 103.23 | 85.96 | 91.71 | 98.60 | 103.13 | 0.000 |
| 3 | 79.06 | 96.04 | 107.11 | 100.74 | 104.98 | 107.37 | 103.67 | 103.31 | 100.91 | 87.53 | 80.14 | 92.05 | 97.68 | -0.690 |
| 4 | 74.90 | 94.83 | 106.00 | 98.72 | 103.01 | 105.52 | 102.02 | 102.03 | 99.84 | 86.75 | 87.97 | 91.27 | 96.65 | -0.881 |
| 5 | 72.73 | 93.45 | 104.59 | 97.12 | 101.17 | 104.34 | 100.41 | 100.27 | 97.77 | 85.13 | 86.53 | 89.65 | 94.96 | -0.988 |
| 6 | 79.37 | 92.19 | 102.67 | 95.35 | 99.19 | 102.37 | 98.58 | 98.54 | 96.65 | 86.79 | 87.09 | 89.57 | 94.66 | -1.234 |
| 7 | 75.04 | 91.32 | 101.04 | 94.10 | 97.68 | 100.74 | 97.17 | 97.04 | 95.22 | 85.78 | 86.83 | 88.57 | 92.80 | -1.346 |
| 8 | 73.26 | 90.58 | 99.58 | 93.14 | 96.39 | 99.20 | 95.97 | 95.70 | 93.71 | 84.60 | 86.09 | 87.36 | 91.54 | -1.405 |
| 9 | 77.07 | 89.32 | 97.81 | 91.76 | 94.75 | 97.47 | 94.38 | 94.18 | 92.49 | 84.60 | 85.43 | 86.80 | 90.49 | -1.592 |
| 10 | 78.44 | 88.37 | 96.43 | 90.93 | 93.62 | 96.17 | 93.36 | 93.14 | 91.80 | 85.78 | 86.09 | 67.41 | 90.28 | -1.637 |
| 11 | 75.52 | 88.62 | 95.27 | 90.23 | 92.71 | 95.01 | 92.47 | 92.28 | 90.76 | 85.56 | 86.22 | 87.19 | 89.73 | -1.981 |
| 12 | 73.35 | 88.06 | 94.01 | 89.49 | 91.67 | 93.79 | 91.50 | 91.32 | 89.71 | 84.16 | 85.21 | 85.79 | 88.55 | -1.968 |
| 13 | 76.72 | 07.58 | 92.88 | 88.93 | 90.84 | 92.75 | 90.69 | 90.41 | 88.75 | 84.86 | 85.26 | 85.91 | 88.16 | -2.137 |
| 14 | 77.16 | 87.40 | 91.93 | 88.49 | 90.19 | 91.89 | 90.04 | 89.84 | 88.40 | 85.61 | 85.70 | 86.07 | 87.96 | -2.343 |
| 15 | 77.34 | 87.18 | 91.15 | 88.14 | 89.58 | 91.15 | 89.51 | 89.32 | 87.84 | 85.74 | 85.74 | 86.20 | 87.76 | -2.526 |
| 16 | 77.25 | 86.96 | 90.41 | 87.75 | 89.01 | 90.45 | 88.98 | 88.75 | 87.14 | 85.61 | 85.65 | 86.07 | 87.41 | -2.654 |

| SPACE | "R" | INTERCEPT | SLOPE | H |
|---|---|---|---|---|
| 0.1250 | -0.996 | 0.755 | -0.148 | 1.319 |

Appendix O. Continued

Table 12. Data for "h" and Flow Calculation, 1/8" Spacing, 2 cfm/ft$^2$ Ventilation

PRESSURE DROP ACROSS NOZZLE= 0.7460, DEWPOINT TEMPERATURE= 40.00, BAROMETRIC PRESSURE= 28.82

| HOUR | TA | TF | T8 | T4 | T0 | T12 | ABT | TP | TN | TC | TIN | T1 | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 79.37 | 116.14 | 112.34 | 114.07 | 114.07 | 113.31 | 113.92 | 114.33 | 107.67 | 121.98 | 120.39 | 118.96 | 116.65 | 0.000 |
| 3 | 76.41 | 98.67 | 111.53 | 104.47 | 109.28 | 111.87 | 107.68 | 107.11 | 104.12 | 85.52 | 87.09 | 92.94 | 100.03 | -0.596 |
| 4 | 73.08 | 96.78 | 110.22 | 101.77 | 106.86 | 110.18 | 105.57 | 105.41 | 102.50 | 85.26 | 87.39 | 91.52 | 98.46 | -0.777 |
| 5 | 77.74 | 95.09 | 108.43 | 99.45 | 104.38 | 108.09 | 103.42 | 103.40 | 101.12 | 86.57 | 87.62 | 91.09 | 97.24 | -0.953 |
| 6 | 77.69 | 94.14 | 106.73 | 97.90 | 102.50 | 106.22 | 101.76 | 101.73 | 99.45 | 86.70 | 87.62 | 90.64 | 96.18 | -1.157 |
| 7 | 77.65 | 93.23 | 104.81 | 96.52 | 100.70 | 104.34 | 100.14 | 100.09 | 97.81 | 86.66 | 87.58 | 90.02 | 95.06 | -1.311 |
| 8 | 77.56 | 92.41 | 103.01 | 95.27 | 99.10 | 102.50 | 98.64 | 98.59 | 96.73 | 86.66 | 87.58 | 89.44 | 94.01 | -1.459 |
| 9 | 77.29 | 91.58 | 101.17 | 94.10 | 97.60 | 100.74 | 97.20 | 96.99 | 95.05 | 86.57 | 87.66 | 89.35 | 93.17 | -1.639 |
| 10 | 77.29 | 90.89 | 99.58 | 93.14 | 96.30 | 99.19 | 95.97 | 95.70 | 94.18 | 86.53 | 87.36 | 88.73 | 92.21 | -1.755 |

| SPACE | "R" | INTERCEPT | SLOPE | H |
|---|---|---|---|---|
| 0.1250 | -0.998 | 0.378 | -0.166 | 1.485 |

Table 13. Data for "h" and Flow Calculation, 1/8" Spacing, 2 cfm/ft$^2$ Ventilation

PRESSURE DROP ACROSS NOZZLE= 0.7230, DEWPOINT TEMPERATURE= 48.00, BAROMETRIC PRESSURE= 28.67

| HOUR | TA | TF | T8 | T4 | T8 | T12 | ABT | TP | TN | TC | TIN | T1 | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 78.00 | 110.60 | 106.69 | 109.75 | 108.52 | 107.62 | 108.63 | 109.59 | 104.17 | 118.46 | 115.97 | 114.28 | 111.93 | 0.000 |
| 3 | 77.69 | 97.12 | 106.77 | 101.55 | 105.19 | 107.11 | 103.95 | 103.87 | 100.65 | 86.83 | 87.57 | 92.51 | 98.19 | -0.595 |
| 4 | 77.87 | 95.44 | 105.88 | 99.41 | 103.27 | 105.88 | 102.30 | 102.50 | 99.58 | 87.36 | 88.06 | 91.08 | 97.19 | -0.806 |
| 5 | 78.84 | 94.18 | 104.59 | 97.64 | 101.47 | 104.34 | 100.71 | 101.08 | 98.46 | 87.58 | 88.01 | 90.94 | 96.01 | -0.988 |
| 6 | 77.07 | 93.19 | 103.27 | 96.35 | 99.96 | 102.97 | 99.38 | 99.75 | 96.95 | 86.79 | 87.36 | 90.15 | 94.55 | -1.129 |
| 7 | 74.10 | 92.32 | 101.81 | 95.14 | 98.50 | 101.47 | 98.04 | 98.37 | 95.48 | 86.09 | 86.96 | 89.45 | 93.91 | -1.271 |
| 8 | 74.68 | 91.76 | 100.44 | 94.18 | 97.29 | 100.09 | 96.92 | 97.17 | 94.62 | 86.14 | 87.05 | 88.96 | 93.06 | -1.396 |
| 9 | 73.31 | 91.10 | 99.06 | 93.32 | 96.13 | 98.72 | 95.61 | 96.00 | 93.27 | 86.00 | 86.83 | 88.79 | 92.39 | -1.558 |
| 10 | 77.21 | 90.36 | 97.51 | 92.78 | 94.79 | 97.25 | 94.56 | 94.83 | 92.36 | 86.75 | 87.01 | 88.37 | 91.60 | -1.749 |

| SPACE | "R" | INTERCEPT | SLOPE | H |
|---|---|---|---|---|
| 0.1250 | -0.998 | 0.848 | -0.157 | 1.404 |

Appendix O. Continued

Table 14. Data for "h" and Flow Calculation, 1/8" Spacing, 3 cfm/ft$^2$ Ventilation

PRESSURE DROP ACROSS NOZZLE= 1.3090, DEWPOINT TEMPERATURE= 40.00, BAROMETRIC PRESSURE= 28.95

| HOUR | TA | TF | TB | T4 | T8 | T12 | ABT | TP | TN | TC | TIN | T1 | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 75.79 | 110.30 | 110.51 | 112.30 | 111.79 | 111.32 | 111.45 | 110.64 | 107.24 | 89.49 | 98.72 | 100.97 | 105.81 | 0.000 |
| 1 | 75.13 | 102.41 | 110.60 | 108.31 | 110.73 | 111.28 | 109.20 | 108.05 | 105.28 | 86.18 | 89.49 | 95.34 | 101.69 | -0.262 |
| 2 | 76.23 | 98.16 | 109.79 | 103.91 | 108.13 | 110.22 | 106.56 | 106.00 | 103.14 | 86.09 | 88.43 | 92.93 | 99.47 | -0.525 |
| 3 | 79.50 | 95.57 | 108.18 | 100.39 | 105.06 | 108.18 | 103.88 | 103.87 | 101.60 | 87.75 | 89.01 | 91.69 | 97.78 | -0.008 |
| 4 | 75.92 | 93.88 | 106.30 | 97.90 | 102.45 | 106.00 | 101.61 | 101.73 | 99.53 | 87.27 | 88.67 | 90.63 | 96.18 | -1.034 |
| 5 | 75.61 | 92.67 | 104.42 | 96.13 | 100.31 | 103.87 | 95.71 | 99.88 | 97.81 | 86.22 | 87.84 | 89.58 | 94.73 | -1.211 |
| 6 | 74.77 | 91.49 | 102.24 | 94.49 | 98.24 | 101.68 | 97.82 | 97.58 | 96.00 | 86.00 | 87.27 | 88.79 | 93.39 | -1.405 |
| 7 | 74.95 | 90.63 | 100.22 | 93.19 | 96.52 | 99.66 | 96.20 | 96.26 | 94.27 | 85.83 | 86.92 | 88.13 | 92.14 | -1.561 |
| 8 | 75.17 | 89.97 | 98.37 | 92.23 | 95.14 | 97.98 | 94.88 | 94.92 | 93.14 | 85.74 | 86.61 | 87.94 | 91.43 | -1.758 |
| 9 | 78.44 | 89.15 | 96.39 | 91.02 | 93.53 | 96.09 | 93.35 | 93.36 | 91.67 | 86.79 | 86.92 | 87.83 | 90.60 | -2.025 |

| SPACE | "R" | INTERCEPT | SLOPE | H |
|---|---|---|---|---|
| 0.1250 | -0.997 | 0.833 | -0.210 | 1.879 |

Table 15. Data for "h" and Flow Calculation, 1/8" Spacing, 3 cfm/ft$^2$ Ventilation

PRESSURE DROP ACROSS NOZZLE= 1.3650, DEWPOINT TEMPERATURE= 35.00, BAROMETRIC PRESSURE= 28.43

| HOUR | TA | TF | TO | T4 | T8 | T12 | ABT | TP | TN | TC | TIN | T1 | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 77.25 | 107.75 | 111.70 | 112.46 | 112.80 | 112.59 | 111.90 | 110.43 | 107.11 | 87.01 | 91.93 | 98.45 | 104.46 | 0.000 |
| 1 | 77.56 | 101.25 | 111.32 | 107.41 | 110.81 | 112.00 | 109.13 | 108.09 | 105.11 | 86.61 | 88.40 | 94.61 | 101.35 | -0.305 |
| 2 | 77.60 | 97.51 | 110.43 | 103.27 | 108.01 | 110.68 | 106.48 | 106.26 | 103.65 | 86.44 | 88.01 | 92.70 | 99.48 | -0.573 |
| 3 | 77.52 | 95.18 | 108.86 | 100.18 | 105.15 | 108.69 | 104.01 | 104.00 | 101.38 | 86.31 | 87.75 | 90.83 | 97.41 | -0.787 |
| 4 | 77.12 | 93.40 | 106.81 | 97.73 | 102.58 | 106.39 | 101.70 | 101.85 | 99.23 | 86.00 | 87.31 | 89.94 | 95.90 | -1.014 |
| 5 | 77.12 | 92.15 | 104.59 | 95.78 | 100.22 | 104.08 | 99.61 | 99.62 | 97.42 | 86.05 | 87.09 | 89.41 | 94.51 | -1.226 |
| 6 | 77.07 | 91.15 | 102.33 | 94.23 | 98.20 | 101.81 | 97.74 | 97.90 | 95.61 | 86.05 | 86.83 | 88.83 | 93.36 | -1.443 |
| 7 | 77.47 | 90.32 | 100.18 | 92.88 | 96.39 | 99.71 | 96.06 | 96.13 | 94.18 | 86.18 | 86.79 | 88.38 | 92.26 | -1.642 |
| 8 | 77.43 | 89.59 | 98.24 | 91.80 | 94.68 | 97.90 | 94.62 | 94.53 | 92.80 | 86.26 | 86.66 | 87.89 | 91.21 | -1.802 |
| 9 | 76.68 | 89.01 | 96.60 | 90.93 | 93.58 | 96.26 | 93.39 | 93.36 | 91.54 | 86.00 | 86.31 | 87.63 | 90.49 | -1.999 |
| 10 | 77.60 | 88.62 | 94.88 | 90.06 | 92.28 | 94.66 | 92.19 | 92.28 | 90.67 | 86.57 | 86.83 | 87.61 | 89.55 | -2.282 |
| 11 | 77.03 | 88.19 | 93.71 | 89.45 | 91.41 | 93.49 | 91.32 | 91.28 | 89.89 | 86.40 | 86.53 | 86.86 | 89.07 | -2.315 |
| 12 | 77.16 | 87.79 | 92.36 | 88.75 | 90.41 | 92.28 | 90.38 | 90.41 | 88.88 | 86.22 | 86.35 | 86.69 | 88.55 | -2.545 |

| SPACE | "R" | INTERCEPT | SLOPE | H |
|---|---|---|---|---|
| 0.1250 | -0.997 | 0.832 | -0.202 | 1.799 |

Appendix D. Continued

Table 16. Data for "h" and Flow Calculation, 1/8" Spacing, 3 cfm/ft$^2$ Ventilation

PRESSURE DROP ACROSS NOZZLE= 1.3560, DEWPOINT TEMPERATURE= 55.50, BAROMETRIC PRESSURE= 28.52

| HOUR | TA | TF | TB | T4 | T8 | T12 | ABT | TP | TN | TC | TIN | T1 | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 77.56 | 110.60 | 114.58 | 114.83 | 115.25 | 115.34 | 114.50 | 113.27 | 109.67 | 89.45 | 95.61 | 101.51 | 107.39 | 0.000 |
| 4 | 77.29 | 95.78 | 109.37 | 103.14 | 105.06 | 108.94 | 104.18 | 104.38 | 101.94 | 88.01 | 89.67 | 91.95 | 98.17 | -1.000 |
| 5 | 77.12 | 94.40 | 107.16 | 98.20 | 102.75 | 106.64 | 102.09 | 102.20 | 99.92 | 87.57 | 89.32 | 91.33 | 96.76 | -1.202 |
| 6 | 76.76 | 93.27 | 104.94 | 96.52 | 100.65 | 104.38 | 100.16 | 100.27 | 98.29 | 87.79 | 88.88 | 90.58 | 95.42 | -1.392 |
| 7 | 76.45 | 92.19 | 102.71 | 95.35 | 98.72 | 102.15 | 98.34 | 98.42 | 96.56 | 87.62 | 88.53 | 89.82 | 94.12 | -1.574 |
| 8 | 76.37 | 91.41 | 100.70 | 93.84 | 97.12 | 100.27 | 96.82 | 96.78 | 94.79 | 87.49 | 88.23 | 89.11 | 92.94 | -1.716 |
| 9 | 76.72 | 90.80 | 99.02 | 92.88 | 95.78 | 98.59 | 95.54 | 95.39 | 93.62 | 87.44 | 88.06 | 89.07 | 92.23 | -1.907 |
| 10 | 75.75 | 89.93 | 97.21 | 91.80 | 94.31 | 96.91 | 94.15 | 94.01 | 92.15 | 86.96 | 87.53 | 88.01 | 91.01 | -2.013 |
| 11 | 75.75 | 87.41 | 95.74 | 90.97 | 93.19 | 95.48 | 93.05 | 93.01 | 91.23 | 86.96 | 87.31 | 88.01 | 90.51 | -2.244 |
| 12 | 75.83 | 88.93 | 94.44 | 90.23 | 92.19 | 94.27 | 92.09 | 92.02 | 90.36 | 87.18 | 87.40 | 87.65 | 89.83 | -2.389 |

| SPACE | "R" | INTERCEPT | SLOPE | H |
|---|---|---|---|---|
| 0.1250 | -0.999 | 0.706 | -0.171 | 1.526 |

Appendix D. Continued

Table 17. Data for "h" and Flow Calculation, 3/16" Spacing, 1 cfm/ft$^2$ Ventilation

PRESSURE ORCP ACROSS NOZZLE= 0.1980, DEWPOINT TEMPERATURE= 39.00, BARCMETRIC PRESSURE= 28.95

| HOUR | TA | TF | T8 | T4 | T8 | T12 | ABT | TP | TN | TC | TIN | T1 | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 73.08 | 97.29 | 94.23 | 96.78 | 55.70 | 95.05 | 95.82 | 94.96 | 88.93 | 91.36 | 95.14 | 94.15 | 94.55 | 0.000 |
| 1 | 72.64 | 92.10 | 94.31 | 95.09 | 95.35 | 95.05 | 94.68 | 94.68 | 09.45 | 75.66 | 81.75 | 84.82 | 89.85 | -0.213 |
| 2 | 73.04 | 89.32 | 94.36 | 92.93 | 94.53 | 94.92 | 93.55 | 94.70 | 89.36 | 74.24 | 79.24 | 82.82 | 88.76 | -0.387 |
| 3 | 73.48 | 87.36 | 94.10 | 90.93 | 93.27 | 94.49 | 92.35 | 94.27 | 89.45 | 74.37 | 78.49 | 81.21 | 87.74 | -0.561 |
| 4 | 73.97 | 86.05 | 93.66 | 89.36 | 92.06 | 93.79 | 91.27 | 93.62 | 88.93 | 74.55 | 78.13 | 80.81 | 87.22 | -0.753 |
| 5 | 73.88 | 84.95 | 92.97 | 88.10 | 90.93 | 92.93 | 90.23 | 92.71 | 88.45 | 74.50 | 77.83 | 80.19 | 86.45 | -0.908 |
| 6 | 74.10 | 84.08 | 92.15 | 86.92 | 89.71 | 91.97 | 89.18 | 91.80 | 87.53 | 74.68 | 77.60 | 79.78 | 85.79 | -1.086 |
| 7 | 74.46 | 83.42 | 91.19 | 85.96 | 88.67 | 90.97 | 88.23 | 90.71 | 87.01 | 74.95 | 77.52 | 79.47 | 85.09 | -1.231 |
| 8 | 74.37 | 82.71 | 90.19 | 85.04 | 87.66 | 89.93 | 87.27 | 89.71 | 86.31 | 74.81 | 77.34 | 78.76 | 84.23 | -1.339 |
| 9 | 74.28 | 82.10 | 89.28 | 84.25 | 86.75 | 89.01 | 86.42 | 88.67 | 85.39 | 74.68 | 77.03 | 78.62 | 83.65 | -1.477 |
| 10 | 73.66 | 81.44 | 88.32 | 83.46 | 85.87 | 88.06 | 35.57 | 87.75 | 84.60 | 74.28 | 76.59 | 78.23 | 02.99 | -1.635 |
| 11 | 73.17 | 80.87 | 87.31 | 82.76 | 85.00 | 87.14 | 84.75 | 86.79 | 83.77 | 73.75 | 76.06 | 77.69 | 82.24 | -1.690 |
| 12 | 73.17 | 80.25 | 86.40 | 82.06 | 84.25 | 86.31 | 84.00 | 85.91 | 82.80 | 73.39 | 75.61 | 76.76 | 81.34 | -1.695 |
| 13 | 72.68 | 79.55 | 85.56 | 81.26 | 83.33 | 85.35 | 83.12 | 84.91 | 82.14 | 72.91 | 75.13 | 76.27 | 80.59 | -1.793 |
| 14 | 72.55 | 79.15 | 84.78 | 80.69 | 82.67 | 84.60 | 82.48 | 84.21 | 81.70 | 72.91 | 74.90 | 76.27 | 80.24 | -1.938 |
| 15 | 72.51 | 78.71 | 84.08 | 00.12 | 82.01 | 83.90 | 81.86 | 83.51 | 81.26 | 72.86 | 74.77 | 75.64 | 79.57 | -1.963 |

| SPACE | "R" | INTERCEPT | SLOPE | H |
|---|---|---|---|---|
| 0.1875 | -0.985 | 0.787 | -0.125 | 1.101 |

Table 18. Data for "h" and Flow Calculation, 3/16" Spacing, 1 cfm/ft$^2$ Ventilation

PRESSURE DROP ACROSS NOZZLE= 0.1920, DEWPOINT TEMPERATURE= 43.00, BAROMETRIC PRESSURE= 28.67

| HOUR | TA | TF | T8 | T4 | T8 | T12 | ABT | TP | TN | TC | TIN | T1 | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 95.14 | 102.37 | 100.91 | 102.07 | 101.64 | 101.51 | 101.71 | 101.38 | 94.56 | 101.55 | 102.75 | 101.44 | 101.41 | 0.000 |
| 1 | 72.73 | 96.99 | 100.87 | 100.48 | 101.34 | 101.47 | 100.55 | 101.12 | 94.27 | 78.18 | 86.70 | 88.50 | 94.81 | -0.184 |
| 2 | 72.64 | 93.88 | 100.61 | 97.98 | 100.14 | 101.04 | 99.10 | 100.65 | 94.49 | 76.32 | 02.36 | 86.06 | 93.36 | -0.375 |
| 3 | 73.17 | 91.97 | 100.31 | 96.09 | 98.98 | 100.48 | 97.92 | 100.18 | 94.10 | 76.14 | 81.26 | 84.72 | 92.45 | -0.527 |
| 4 | 73.53 | 90.41 | 99.58 | 94.31 | 97.51 | 99.62 | 96.61 | 99.23 | 93.45 | 76.37 | 80.78 | 84.37 | 91.80 | -0.723 |
| 5 | 74.15 | 89.01 | 98.72 | 92.80 | 96.13 | 98.59 | 95.34 | 98.11 | 92.54 | 75.97 | 79.81 | 83.39 | 90.75 | -0.870 |
| 6 | 73.44 | 87.75 | 97.68 | 91.41 | 94.79 | 97.42 | 94.08 | 97.17 | 91.84 | 75.70 | 79.37 | 82.54 | 89.85 | -1.031 |
| 7 | 74.02 | 86.83 | 96.52 | 90.10 | 93.49 | 96.26 | 92.88 | 95.78 | 90.63 | 75.75 | 79.06 | 82.01 | 88.50 | -1.168 |
| 8 | 74.06 | 86.00 | 95.44 | 89.01 | 92.28 | 95.05 | 91.76 | 94.75 | 90.02 | 75.75 | 79.19 | 81.43 | 88.09 | -1.310 |
| 9 | 74.15 | 85.35 | 94.31 | 88.10 | 91.19 | 93.88 | 90.75 | 93.58 | 88.80 | 75.61 | 78.75 | 80.71 | 87.15 | -1.357 |
| 10 | 78.84 | 84.86 | 93.10 | 87.23 | 90.15 | 92.71 | 89.77 | 92.36 | 88.80 | 77.60 | 79.50 | 81.55 | 86.95 | -1.658 |
| 11 | 72.95 | 84.29 | 91.97 | 86.48 | 89.15 | 91.62 | 88.85 | 91.28 | 87.40 | 76.06 | 78.75 | 80.58 | 85.93 | -1.688 |
| 12 | 77.21 | 83.72 | 91.06 | 85.83 | 88.36 | 90.76 | 88.08 | 90.45 | 86.61 | 76.32 | 78.62 | 00.26 | 85.36 | -1.792 |
| 13 | 78.22 | 83.20 | 89.93 | 85.00 | 87.40 | 89.62 | 87.15 | 89.32 | 86.44 | 77.29 | 78.84 | 80.08 | 84.70 | -1.939 |
| 14 | 72.95 | 82.71 | 39.06 | 84.43 | 86.61 | 88.75 | 86.42 | 88.40 | 85.26 | 76.01 | 78.22 | 79.37 | 83.89 | -1.952 |
| 15 | 71.04 | 82.28 | 88.32 | 83.94 | 86.00 | 88.06 | 85.82 | 87.71 | 84.21 | 74.64 | 77.69 | 78.58 | 83.14 | -1.935 |

| SPACE | "R" | INTERCEPT | SLOPE | H |
|---|---|---|---|---|
| 0.1875 | -0.987 | 0.836 | -0.132 | 1.226 |

Appendix D. Continued

Table 19. Data for "h" and Flow Calculation, 3/16" Spacing, 1 cfm/ft$^2$ Ventilation

PRESSURE DROP ACROSS NOZZLE= 0.1890, DEWPOINT TEMPERATURE= 45.50, BARCMETRIC PRESSURE= 28.93

| HOUR | TA | TF | T8 | T4 | T8 | T12 | ABT | TP | TN | TC | TIN | TI | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 73.13 | 98.89 | 94.62 | 97.68 | 96.26 | 95.44 | 96.53 | 95.61 | 89.75 | 102.20 | 101.17 | 100.34 | 97.98 | 0.000 |
| 1 | 71.53 | 94.10 | 94.88 | 96.52 | 96.30 | 95.70 | 95.75 | 95.70 | 89.93 | 78.44 | 84.69 | 87.02 | 91.36 | -0.164 |
| 2 | 72.82 | 91.45 | 95.05 | 94.44 | 95.57 | 95.70 | 94.74 | 95.61 | 90.02 | 76.99 | 82.06 | 84.95 | 90.30 | -0.339 |
| 3 | 76.01 | 89.28 | 94.79 | 92.49 | 94.44 | 95.31 | 93.57 | 95.18 | 90.32 | 76.90 | 81.00 | 83.74 | 89.46 | -0.543 |
| 4 | 77.25 | 87.92 | 94.36 | 90.89 | 93.23 | 94.62 | 92.47 | 94.57 | 90.10 | 77.38 | 80.74 | 83.06 | 88.82 | -0.749 |
| 5 | 73.97 | 86.96 | 93.97 | 89.67 | 92.23 | 94.01 | 91.59 | 93.97 | 89.49 | 76.54 | 80.08 | 82.62 | 88.29 | -0.915 |
| 6 | 72.02 | 85.91 | 93.27 | 88.53 | 91.10 | 93.14 | 90.59 | 93.10 | 88.49 | 75.57 | 79.15 | 81.25 | 87.10 | -1.007 |
| 7 | 71.44 | 85.13 | 92.41 | 87.58 | 90.15 | 92.28 | 89.69 | 92.19 | 87.44 | 74.55 | 78.62 | 80.63 | 86.41 | -1.127 |
| 8 | 76.45 | 84.29 | 91.54 | 86.61 | 89.10 | 91.32 | 88.74 | 91.23 | 67.09 | 75.66 | 78.36 | 80.18 | 85.71 | -1.273 |
| 9 | 78.13 | 83.68 | 90.54 | 85.78 | 88.19 | 90.32 | 87.85 | 90.23 | 86.92 | 76.37 | 78.36 | 80.31 | 85.27 | -1.474 |
| 10 | 79.15 | 33.33 | 89.71 | 85.08 | 87.36 | 89.49 | 87.11 | 89.32 | 86.61 | 77.47 | 78.84 | 80.25 | 84.79 | -1.619 |
| 11 | 73.93 | 82.67 | 88.93 | 84.21 | 86.66 | 88.62 | 86.32 | 88.53 | 65.78 | 76.54 | 78.53 | 79.91 | 84.22 | -1.768 |
| 12 | 74.06 | 82.14 | 88.10 | 83.59 | 85.91 | 87.75 | 85.59 | 87.62 | 85.17 | 76.63 | 78.40 | 79.41 | 83.52 | -1.835 |
| 13 | 77.12 | 81.62 | 87.27 | 82.98 | 85.17 | 86.96 | 84.89 | 86.83 | 84.65 | 76.68 | 78.18 | 78.88 | 82.86 | -1.906 |
| 14 | 76.99 | 81.53 | 86.66 | 82.58 | 84.65 | 86.35 | 84.42 | 86.22 | 84.08 | 76.81 | 78.40 | 79.01 | 82.62 | -2.044 |
| 15 | 76.10 | 81.26 | 86.22 | 82.41 | 84.29 | 85.91 | 84.09 | 85.78 | 83.59 | 76.14 | 78.35 | 78.93 | 82.36 | -2.101 |
| 16 | 75.97 | 80.74 | 85.52 | 81.92 | 83.77 | 85.26 | 83.52 | 85.13 | 83.11 | 75.44 | 77.38 | 78.22 | 81.67 | -2.085 |
| 17 | 77.29 | 80.56 | 84.06 | 81.44 | 83.20 | 84.65 | 83.00 | 84.43 | 82.89 | 76.85 | 77.96 | 78.48 | 81.45 | -2.277 |
| 18 | 77.12 | 80.34 | 84.21 | 81.04 | 82.67 | 83.99 | 82.49 | 83.81 | 82.45 | 77.12 | 78.13 | 78.74 | 81.28 | -2.529 |
| 19 | 77.12 | 80.21 | 83.72 | 80.82 | 82.28 | 83.55 | 82.15 | 83.33 | 82.06 | 77.21 | 78.13 | 78.83 | 81.08 | -2.667 |
| 20 | 77.25 | 80.16 | 83.29 | 80.65 | 82.01 | 83.15 | 81.88 | 82.93 | 81.79 | 77.25 | 78.13 | 78.29 | 80.61 | -2.528 |
| 21 | 77.16 | 79.99 | 82.89 | 80.43 | 81.66 | 82.76 | 81.57 | 82.54 | 81.48 | 77.21 | 78.09 | 78.25 | 80.39 | -2.619 |

| SPACE | "R" | INTERCEPT | SLOPE | H |
|---|---|---|---|---|
| 0.1875 | -0.989 | 0.781 | -0.123 | 1.048 |

Appendix D. Continued

Table 20. Data for "h" and Flow Calculation, 3/16" Spacing, 2 cfm/ft$^2$ Ventilation

PRESSURE DROP ACROSS NOZZLE= 0.7460, DEWPOINT TEMPERATURE= 42.00, BAROMETRIC PRESSURE= 28.96

| HOUR | TA | TF | T8 | T4 | T8 | T12 | ABT | TP | TN | TC | TIN | TI | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 75.38 | 100.76 | 99.14 | 100.58 | 99.86 | 99.32 | 99.93 | 100.04 | 95.18 | 100.04 | 102.02 | 99.92 | 99.98 | 0.000 |
| 2 | 74.66 | 89.42 | 97.88 | 94.28 | 96.98 | 98.06 | 95.74 | 97.34 | 92.66 | 75.38 | 80.60 | 83.38 | 90.36 | -0.575 |
| 3 | 74.84 | 86.72 | 96.80 | 91.22 | 94.82 | 96.80 | 93.65 | 95.54 | 91.22 | 75.56 | 79.52 | 81.82 | 88.68 | -0.817 |
| 4 | 75.02 | 84.92 | 95.18 | 89.24 | 92.48 | 95.18 | 91.74 | 94.10 | 90.32 | 75.56 | 78.80 | 80.66 | 87.38 | -1.058 |
| 5 | 75.20 | 83.66 | 93.56 | 87.26 | 90.68 | 93.20 | 89.94 | 92.12 | 88.70 | 75.56 | 78.44 | 80.08 | 86.10 | -1.282 |
| 6 | 75.74 | 82.58 | 91.94 | 86.00 | 89.06 | 91.58 | 88.47 | 90.50 | 87.80 | 76.10 | 78.26 | 79.46 | 84.98 | -1.454 |
| 7 | 75.92 | 82.04 | 90.50 | 84.56 | 87.44 | 89.96 | 87.06 | 88.88 | 86.18 | 75.92 | 78.08 | 79.28 | 84.08 | -1.672 |
| 8 | 75.56 | 81.14 | 88.70 | 83.66 | 86.18 | 88.34 | 85.77 | 87.26 | 85.10 | 75.92 | 77.54 | 78.70 | 82.98 | -1.803 |

| SPACE | "R" | INTERCEPT | SLOPE | H |
|---|---|---|---|---|
| 0.1875 | -0.997 | 0.816 | -0.207 | 1.815 |

Table 21. Data for "h" and Flow Calculation, 3/16" Spacing, 2 cfm/ft$^2$ Ventilation

PRESSURE DROP ACROSS NOZZLE= 0.7550, DEWPOINT TEMPERATURE= 40.00, BAROMETRIC PRESSURE= 28.77

| HOUR | TA | TF | T8 | T4 | T8 | T12 | ABT | TP | TN | TC | TIN | TI | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 76.64 | 100.04 | 97.70 | 99.32 | 98.78 | 98.24 | 98.80 | 98.60 | 93.38 | 98.96 | 101.84 | 99.42 | 99.01 | 0.000 |
| 2 | 74.66 | 88.52 | 96.80 | 93.56 | 96.08 | 96.98 | 94.82 | 96.08 | 91.94 | 75.38 | 80.96 | 82.80 | 89.44 | -0.554 |
| 3 | 74.48 | 86.00 | 95.72 | 90.68 | 94.10 | 95.72 | 92.84 | 94.46 | 90.68 | 75.20 | 79.52 | 80.88 | 87.67 | -0.767 |
| 4 | 75.02 | 84.20 | 94.28 | 88.52 | 91.94 | 94.28 | 90.99 | 93.02 | 89.42 | 75.20 | 70.62 | 80.30 | 86.66 | -1.030 |
| 5 | 75.20 | 82.94 | 92.84 | 86.72 | 89.56 | 92.46 | 89.26 | 91.22 | 87.98 | 75.20 | 78.08 | 79.14 | 85.18 | -1.205 |
| 6 | 75.02 | 82.04 | 91.04 | 85.10 | 88.34 | 90.86 | 87.71 | 89.42 | 86.90 | 75.38 | 77.54 | 78.74 | 84.08 | -1.400 |
| 7 | 75.02 | 81.14 | 89.60 | 84.02 | 86.90 | 89.06 | 86.34 | 87.80 | 85.28 | 75.20 | 77.36 | 77.98 | 82.89 | -1.530 |
| 8 | 74.84 | 80.24 | 87.98 | 82.94 | 85.64 | 87.62 | 85.06 | 86.18 | 84.02 | 75.02 | 77.00 | 77.80 | 81.99 | -1.695 |
| 9 | 74.66 | 79.34 | 86.36 | 81.86 | 84.20 | 86.18 | 83.77 | 84.92 | 83.12 | 75.02 | 76.46 | 77.22 | 81.07 | -1.882 |

| SPACE | "R" | INTERCEPT | SLOPE | h |
|---|---|---|---|---|
| 0.1875 | -0.996 | 0.791 | -0.186 | 1.634 |

Appendix O. Continued

Table 22. Data for "h" and Flow Calculation, 3/16" Spacing, 2 cfm/ft$^2$ Ventilation

PRESSURE DROP ACROSS NOZZLE= 0.7560, DEWPOINT TEMPERATURE= 39.00, BAROMETRIC PRESSURE= 28.68

| HOUR | TA | TF | T8 | T4 | T8 | T12 | ABT | TP | TN | TC | TIN | TI | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 75.20 | 99.86 | 97.88 | 99.50 | 98.78 | 98.06 | 98.80 | 98.78 | 93.38 | 103.10 | 101.84 | 101.25 | 100.01 | 0.000 |
| 2 | 74.12 | 88.16 | 96.80 | 93.38 | 95.90 | 97.16 | 94.73 | 96.08 | 91.76 | 74.66 | 80.06 | 82.06 | 89.08 | -0.543 |
| 3 | 74.12 | 85.82 | 95.72 | 90.32 | 93.74 | 95.72 | 92.64 | 94.64 | 90.50 | 74.66 | 78.80 | 80.92 | 87.78 | -0.820 |
| 4 | 74.30 | 83.84 | 94.28 | 88.16 | 91.58 | 94.10 | 90.72 | 92.84 | 89.06 | 74.66 | 77.90 | 79.76 | 86.30 | -1.039 |
| 5 | 71.24 | 82.58 | 92.48 | 86.36 | 89.78 | 92.12 | 88.95 | 91.04 | 87.44 | 72.68 | 76.64 | 77.78 | 84.41 | -1.154 |
| 6 | 73.58 | 81.14 | 90.68 | 84.56 | 87.98 | 90.50 | 87.24 | 89.36 | 86.18 | 73.76 | 76.10 | 77.70 | 83.38 | -1.386 |
| 7 | 73.94 | 80.06 | 89.24 | 83.30 | 86.36 | 88.70 | 85.75 | 87.26 | 84.74 | 74.12 | 75.92 | 76.90 | 82.08 | -1.516 |
| 8 | 73.22 | 79.16 | 87.62 | 82.04 | 84.92 | 87.26 | 84.40 | 85.82 | 83.48 | 73.58 | 75.38 | 76.36 | 81.09 | -1.677 |
| 9 | 72.68 | 78.44 | 86.00 | 80.96 | 83.66 | 85.46 | 83.07 | 84.56 | 82.22 | 73.04 | 75.02 | 75.82 | 80.19 | -1.865 |
| 10 | 72.32 | 77.72 | 84.56 | 79.88 | 82.40 | 84.02 | 81.86 | 83.12 | 80.96 | 72.50 | 74.48 | 75.28 | 79.20 | -1.998 |

| SPACE | "R" | INTERCEPT | SLOPE | H |
|---|---|---|---|---|
| 0.1875 | -0.996 | 0.761 | -0.177 | 1.550 |

Table 23. Data for "h" and Flow Calculation, 3/16" Spacing, 3 cfm/ft$^2$ Ventilation

PRESSURE DROP ACROSS NOZZLE= 1.4430, DEWPOINT TEMPERATURE= 33.00, BAROMETRIC PRESSURE= 28.88

| HOUR | TA | TF | TB | T4 | T8 | T12 | ABT | TP | TN | TC | TIN | TI | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 76.46 | 98.96 | 99.32 | 99.68 | 99.68 | 99.50 | 99.50 | 100.40 | 96.26 | 102.92 | 102.74 | 101.07 | 100.73 | 0.000 |
| 1 | 75.92 | 90.86 | 98.42 | 96.62 | 98.42 | 98.78 | 97.11 | 97.88 | 93.92 | 76.64 | 82.04 | 84.64 | 91.26 | -0.342 |
| 2 | 75.92 | 87.44 | 97.16 | 92.84 | 96.08 | 97.34 | 94.64 | 95.90 | 92.48 | 76.46 | 80.24 | 82.14 | 89.02 | -0.623 |
| 3 | 76.46 | 85.10 | 95.54 | 89.96 | 93.56 | 95.54 | 92.34 | 93.74 | 90.86 | 76.82 | 79.52 | 81.34 | 87.54 | -0.912 |
| 4 | 76.82 | 83.84 | 93.74 | 87.62 | 51.40 | 93.56 | 90.34 | 91.94 | 89.24 | 77.00 | 79.16 | 80.36 | 86.15 | -1.158 |
| 5 | 77.72 | 82.76 | 91.76 | 86.18 | 89.24 | 91.76 | 88.61 | 89.56 | 87.80 | 77.72 | 79.34 | 80.50 | 85.23 | -1.441 |
| 6 | 78.26 | 82.22 | 90.14 | 84.92 | 87.62 | 89.78 | 87.12 | 88.16 | 86.36 | 78.44 | 79.52 | 80.06 | 84.11 | -1.631 |
| 7 | 78.44 | 82.04 | 88.34 | 84.02 | 86.18 | 88.16 | 85.89 | 86.54 | 85.28 | 78.26 | 79.34 | 79.88 | 83.21 | -1.806 |
| 8 | 78.08 | 81.32 | 86.90 | 83.30 | 85.10 | 86.90 | 84.85 | 85.46 | 84.38 | 78.26 | 70.98 | 79.88 | 82.67 | -2.043 |

| SPACE | "R" | INTERCEPT | SLOPE | H |
|---|---|---|---|---|
| 0.1875 | -0.997 | 0.853 | -0.241 | 2.118 |

Appendix D. Continued

Table 24. Data for "h" and Flow Calculation, 3/16" Spacing, 3 cfm/ft$^2$ Ventilation

PRESSURE DROP ACROSS NOZZLE= 1.4290, DEWPOINT TEMPERATURE= 36.50, BAROMETRIC PRESSURE= 28.56

| HOUR | TA | TF | TB | T4 | T8 | T12 | ABT | TP | TN | TC | T1A | T1 | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 77.36 | 97.70 | 96.98 | 98.06 | 97.70 | 97.34 | 97.61 | 98.78 | 94.82 | 98.78 | 101.48 | 98.09 | 98.43 | 0.000 |
| 1 | 76.64 | 90.68 | 96.44 | 95.18 | 96.62 | 96.62 | 95.49 | 96.62 | 92.84 | 77.54 | 82.58 | 84.96 | 90.79 | -0.371 |
| 2 | 76.28 | 87.44 | 95.54 | 91.94 | 94.64 | 95.54 | 93.40 | 94.82 | 91.58 | 77.18 | 80.78 | 82.86 | 88.84 | -0.654 |
| 3 | 76.82 | 85.46 | 94.10 | 89.42 | 92.48 | 94.10 | 91.44 | 93.20 | 89.96 | 77.00 | 79.52 | 81.52 | 87.36 | -0.920 |
| 4 | 77.18 | 84.02 | 92.84 | 87.44 | 90.32 | 92.48 | 89.67 | 91.04 | 88.70 | 77.36 | 79.16 | 80.72 | 85.88 | -1.131 |
| 5 | 77.90 | 83.12 | 91.04 | 86.00 | 88.70 | 90.86 | 88.16 | 89.42 | 87.44 | 77.00 | 78.98 | 80.68 | 85.05 | -1.396 |
| 6 | 78.62 | 82.58 | 89.42 | 84.74 | 87.26 | 89.24 | 86.81 | 87.80 | 86.18 | 78.44 | 79.34 | 80.64 | 84.22 | -1.643 |
| 7 | 79.16 | 82.04 | 87.98 | 83.84 | 86.00 | 87.80 | 85.66 | 86.54 | 85.10 | 78.98 | 79.52 | 80.60 | 83.57 | -1.904 |
| 8 | 78.80 | 81.68 | 86.90 | 83.30 | 84.92 | 86.36 | 84.72 | 85.10 | 84.20 | 78.80 | 79.70 | 79.84 | 82.47 | -1.968 |
| 9 | 78.62 | 81.32 | 85.64 | 82.76 | 84.02 | 85.46 | 83.93 | 84.20 | 83.48 | 78.98 | 79.34 | 80.02 | 82.11 | -2.143 |

| SPACE | "R" | INTERCEPT | SLOPE | H |
|---|---|---|---|---|
| 0.1875 | -0.992 | 0.794 | -0.222 | 1.951 |

Table 25. Data for "h" and Flow Calculation, 3/16" Spacing, 3 cfm/ft$^2$ Ventilation

PRESSURE DROP ACROSS NOZZLE= 1.4190, DEWPOINT TEMPERATURE= 56.00, BAROMETRIC PRESSURE= 28.90

| HOUR | TA | TF | TB | T4 | T8 | T12 | ABT | TP | TN | TC | TIN | T1 | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 73.17 | 96.39 | 96.95 | 97.77 | 97.51 | 97.51 | 97.36 | 97.51 | 95.01 | 79.06 | 84.51 | 88.80 | 93.16 | 0.000 |
| 1 | 74.64 | 89.49 | 95.83 | 93.92 | 95.65 | 96.43 | 94.67 | 95.83 | 93.08 | 75.48 | 78.40 | 83.48 | 89.65 | -0.430 |
| 2 | 74.77 | 86.35 | 94.70 | 90.54 | 93.49 | 95.18 | 92.43 | 94.31 | 91.49 | 75.35 | 77.60 | 81.61 | 87.96 | -0.743 |
| 3 | 74.95 | 84.12 | 92.84 | 87.62 | 90.84 | 94.14 | 90.02 | 92.15 | 89.45 | 75.21 | 76.94 | 79.74 | 85.94 | -1.030 |
| 4 | 74.99 | 82.45 | 91.10 | 85.52 | 88.67 | 91.19 | 88.04 | 90.19 | 87.84 | 74.95 | 76.45 | 78.89 | 84.54 | -1.299 |
| 5 | 75.13 | 81.48 | 89.36 | 83.99 | 86.92 | 89.45 | 86.45 | 88.49 | 86.35 | 75.21 | 76.54 | 78.58 | 83.53 | -1.558 |
| 6 | 76.90 | 80.65 | 87.53 | 82.63 | 85.17 | 87.53 | 84.85 | 86.66 | 85.04 | 76.59 | 77.38 | 78.79 | 82.72 | -1.927 |
| 7 | 77.29 | 80.56 | 86.09 | 81.92 | 83.99 | 86.09 | 83.83 | 85.43 | 84.08 | 77.29 | 78.13 | 78.92 | 82.18 | -2.216 |
| 8 | 77.21 | 80.52 | 85.04 | 81.57 | 83.24 | 85.08 | 83.17 | 84.38 | 83.37 | 77.38 | 78.27 | 79.01 | 81.69 | -2.364 |
| 9 | 77.91 | 80.16 | 83.94 | 81.04 | 82.41 | 83.99 | 82.37 | 83.42 | 82.58 | 77.78 | 78.44 | 78.83 | 81.12 | -2.563 |

| SPACE | "R" | INTERCEPT | SLOPE | H |
|---|---|---|---|---|
| 0.1875 | -0.997 | 0.815 | -0.273 | 2.399 |

Appendix D. Continued

Table 26. Data for "h" and Flow Calculation, 1/4" Spacing, 1 cfm/ft$^2$ Ventilation

PRESSURE DROP ACROSS NOZZLE= 0.1700, DEWPOINT TEMPERATURE= 30.00, BAROMETRIC PRESSURE= 28.70

| HOUR | TA | TF | TB | T4 | T8 | T12 | ABT | TP | TN | TC | TIN | T1 | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 75.20 | 96.80 | 92.84 | 95.90 | 94.28 | 93.20 | 94.55 | 93.20 | 89.78 | 101.84 | 99.86 | 98.83 | 96.01 | 0.000 |
| 2 | 75.02 | 89.60 | 92.84 | 93.02 | 93.74 | 93.38 | 92.84 | 93.02 | 89.60 | 75.92 | 79.16 | 83.34 | 88.18 | -0.313 |
| 3 | 75.38 | 87.98 | 92.84 | 91.04 | 92.84 | 93.20 | 91.87 | 92.84 | 89.60 | 75.74 | 78.26 | 82.58 | 87.71 | -0.457 |
| 4 | 75.92 | 86.54 | 92.48 | 89.60 | 91.76 | 92.66 | 90.88 | 92.48 | 88.88 | 76.10 | 78.26 | 81.78 | 87.13 | -0.682 |
| 5 | 75.92 | 85.64 | 91.94 | 88.34 | 90.86 | 91.94 | 89.98 | 91.40 | 88.70 | 76.10 | 78.26 | 81.20 | 86.30 | -0.867 |
| 6 | 75.74 | 84.74 | 91.22 | 87.26 | 89.60 | 91.04 | 88.97 | 90.68 | 87.62 | 76.10 | 77.90 | 80.62 | 85.65 | -0.986 |
| 7 | 75.20 | 84.02 | 90.68 | 86.54 | 88.70 | 90.32 | 88.23 | 89.96 | 87.26 | 75.92 | 77.36 | 80.44 | 85.20 | -1.128 |
| 8 | 75.38 | 83.30 | 89.78 | 85.64 | 87.98 | 89.24 | 87.35 | 89.24 | 86.36 | 75.74 | 77.36 | 79.68 | 84.40 | -1.251 |
| 9 | 74.84 | 82.76 | 89.06 | 84.92 | 87.08 | 88.70 | 86.65 | 88.34 | 86.00 | 75.56 | 76.82 | 79.50 | 83.92 | -1.359 |
| 10 | 75.56 | 82.22 | 88.16 | 84.20 | 86.36 | 87.80 | 85.89 | 87.44 | 85.46 | 75.74 | 77.00 | 79.10 | 83.27 | -1.461 |
| 11 | 75.38 | 81.86 | 87.44 | 83.48 | 85.64 | 87.08 | 85.21 | 86.72 | 84.56 | 75.38 | 76.82 | 78.74 | 82.73 | -1.561 |
| 12 | 75.38 | 81.14 | 86.72 | 83.12 | 04.92 | 86.36 | 84.58 | 86.00 | 84.20 | 75.38 | 76.64 | 78.16 | 82.08 | -1.637 |
| 13 | 75.20 | 80.78 | 86.00 | 82.58 | 84.30 | 85.82 | 84.04 | 85.28 | 83.48 | 75.38 | 76.46 | 78.16 | 81.72 | -1.710 |
| 14 | 75.02 | 80.60 | 85.46 | 82.22 | 83.84 | 85.10 | 83.55 | 84.74 | 82.76 | 75.38 | 76.10 | 78.16 | 81.45 | -1.833 |
| 15 | 74.84 | 80.24 | 84.74 | 81.68 | 83.30 | 84.38 | 82.96 | 84.38 | 82.58 | 75.20 | 76.10 | 77.98 | 81.18 | -2.016 |
| 16 | 74.84 | 79.88 | 84.02 | 81.14 | 82.76 | 83.84 | 82.42 | 83.66 | 81.86 | 75.20 | 75.92 | 77.40 | 80.53 | -2.003 |

| SPACE | "R" | INTERCEPT | SLOPE | H |
|---|---|---|---|---|
| 0.2500 | -0.991 | 0.809 | -0.119 | 1.026 |

Table 27. Data for "h" and Flow Calculation, 1/4" Spacing, 1 cfm/ft$^2$ Ventilation

PRESSURE DROP ACROSS NOZZLE= 0.1700, DEWPOINT TEMPERATURE= 30.00, BAROMETRIC PRESSURE= 28.70

| HOUR | TA | TF | TB | T4 | T8 | T12 | ABT | TP | TN | TC | TIN | T1 | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 73.76 | 93.20 | 90.32 | 92.30 | 91.40 | 90.50 | 91.49 | 90.50 | 87.26 | 99.32 | 95.18 | 95.73 | 93.11 | 0.000 |
| 2 | 73.40 | 86.90 | 90.32 | 89.60 | 90.68 | 90.50 | 89.85 | 90.50 | 87.08 | 74.66 | 77.00 | 81.50 | 86.00 | -0.356 |
| 3 | 74.12 | 85.28 | 89.96 | 87.98 | 89.60 | 90.14 | 88.83 | 89.96 | 87.08 | 74.84 | 77.18 | 80.52 | 85.24 | -0.553 |
| 4 | 75.02 | 84.20 | 89.60 | 86.72 | 88.88 | 09.78 | 88.07 | 89.42 | 86.54 | 75.20 | 77.18 | 80.30 | 84.86 | -0.726 |
| 5 | 75.38 | 83.66 | 89.06 | 86.00 | 87.98 | 89.06 | 87.35 | 88.88 | 86.00 | 75.56 | 77.36 | 80.08 | 84.48 | -0.893 |
| 6 | 75.20 | 82.76 | 88.70 | 85.10 | 87.08 | 88.34 | 86.56 | 88.16 | 84.74 | 75.56 | 77.18 | 79.50 | 83.83 | -1.031 |
| 7 | 75.02 | 82.40 | 87.80 | 84.38 | 86.36 | 87.80 | 85.91 | 87.62 | 85.10 | 75.20 | 76.82 | 79.14 | 83.38 | -1.165 |
| 8 | 75.56 | 82.04 | 07.26 | 83.84 | 85.82 | 87.08 | 85.35 | 86.72 | 83.66 | 75.74 | 77.18 | 79.10 | 82.91 | -1.259 |
| 9 | 75.02 | 81.68 | 86.72 | 83.30 | 84.92 | 06.36 | 84.69 | 86.00 | 83.84 | 75.56 | 76.64 | 78.92 | 82.46 | -1.397 |
| 10 | 74.84 | 80.78 | 85.82 | 02.58 | 84.30 | 85.64 | 83.97 | 85.46 | 83.12 | 75.38 | 76.46 | 78.16 | 81.81 | -1.458 |
| 11 | 74.66 | 80.60 | 85.28 | 82.04 | 83.84 | 85.10 | 83.48 | 85.10 | 82.76 | 75.20 | 76.10 | 77.98 | 81.54 | -1.636 |
| 12 | 74.48 | 80.24 | 84.74 | 61.68 | 83.30 | 04.38 | 82.96 | 84.20 | 82.22 | 75.02 | 75.92 | 77.80 | 81.00 | -1.677 |
| 13 | 74.66 | 79.88 | 84.02 | 81.32 | 82.76 | 84.02 | 82.51 | 83.66 | 81.86 | 74.84 | 75.92 | 77.62 | 80.64 | -1.758 |
| 14 | 74.48 | 79.34 | 83.66 | 80.96 | 82.22 | 83.48 | 82.04 | 83.12 | 81.32 | 74.84 | 75.74 | 77.04 | 80.08 | -1.762 |
| 15 | 74.48 | 79.34 | 83.12 | 80.42 | 81.68 | 82.94 | 81.57 | 82.58 | 80.78 | 74.66 | 75.56 | 76.86 | 79.72 | -1.853 |
| 16 | 74.66 | 78.80 | 82.58 | 80.24 | 81.32 | 82.40 | 81.16 | 82.22 | 80.78 | 74.84 | 75.56 | 76.46 | 79.34 | -1.898 |

| SPACE | "R" | INTERCEPT | SLOPE | H |
|---|---|---|---|---|
| 0.2500 | -0.980 | 0.725 | -0.108 | 0.936 |

Table 28. Data for "h" and Flow Calculation, 1/4" Spacing, 1 $cfm/ft^2$ Ventilation

PRESSURE DROP ACROSS NOZZLE= 0.1700, DEWPOINT TEMPERATURE= 30.00, BARCMETRIC PRESSURE= 28.70

| HOUR | TA | TF | TB | T4 | T8 | T12 | ABT | TP | TN | TC | TIN | T1 | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 73.04 | 93.38 | 91.04 | 92.66 | 91.94 | 91.40 | 92.05 | 91.58 | 88.16 | 99.14 | 95.72 | 96.13 | 93.85 | 0.000 |
| 2 | 71.96 | 86.54 | 90.86 | 89.60 | 90.86 | 91.04 | 90.05 | 90.86 | 87.44 | 73.22 | 75.92 | 80.64 | 85.75 | -0.382 |
| 3 | 72.14 | 84.56 | 90.50 | 87.80 | 90.14 | 90.50 | 88.99 | 90.14 | 86.18 | 72.68 | 75.02 | 78.94 | 84.54 | -0.523 |
| 4 | 71.78 | 83.30 | 89.96 | 86.36 | 88.88 | 87.96 | 87.96 | 89.60 | 86.90 | 72.86 | 75.02 | 78.54 | 84.07 | -0.720 |
| 5 | 72.32 | 82.40 | 89.24 | 85.28 | 87.80 | 89.24 | 87.03 | 88.70 | 86.00 | 72.68 | 74.66 | 77.78 | 83.24 | -0.843 |
| 6 | 72.86 | 81.68 | 88.52 | 84.20 | 86.72 | 88.34 | 86.09 | 87.98 | 83.66 | 73.04 | 74.66 | 77.56 | 82.77 | -1.028 |
| 7 | 72.50 | 80.96 | 87.62 | 83.30 | 85.64 | 87.44 | 85.17 | 86.90 | 82.94 | 72.68 | 74.66 | 77.20 | 82.05 | -1.166 |
| 8 | 72.32 | 80.24 | 86.90 | 82.58 | 84.92 | 86.54 | 84.40 | 86.18 | 83.12 | 72.86 | 74.12 | 76.80 | 81.49 | -1.289 |
| 9 | 71.78 | 79.70 | 86.00 | 81.86 | 84.02 | 85.64 | 83.59 | 85.28 | 82.22 | 72.50 | 73.94 | 76.44 | 80.86 | -1.410 |
| 10 | 72.14 | 79.16 | 85.10 | 81.14 | 83.30 | 84.92 | 82.87 | 84.56 | 81.68 | 72.32 | 73.58 | 75.68 | 80.12 | -1.467 |
| 11 | 71.96 | 78.62 | 84.38 | 80.60 | 82.58 | 84.20 | 82.22 | 83.66 | 80.78 | 72.50 | 73.40 | 75.86 | 79.76 | -1.609 |
| 12 | 71.60 | 78.08 | 83.66 | 79.88 | 82.04 | 83.48 | 81.57 | 83.12 | 80.96 | 72.50 | 73.40 | 75.26 | 79.20 | -1.692 |
| 13 | 72.14 | 77.54 | 82.76 | 79.34 | 81.14 | 82.58 | 80.80 | 82.22 | 80.06 | 72.14 | 73.22 | 74.92 | 78.57 | -1.799 |
| 14 | 71.42 | 77.36 | 82.22 | 78.80 | 80.60 | 81.86 | 80.26 | 81.50 | 78.80 | 71.96 | 73.04 | 74.74 | 78.12 | -1.873 |
| 15 | 71.42 | 77.00 | 81.68 | 78.44 | 79.88 | 81.32 | 79.74 | 81.14 | 78.44 | 71.96 | 72.86 | 74.74 | 77.94 | -2.057 |
| 16 | 71.60 | 76.64 | 80.78 | 77.90 | 79.52 | 80.78 | 79.23 | 80.42 | 78.80 | 71.96 | 72.50 | 74.16 | 77.29 | -2.031 |

| SPACE | "R" | INTERCEPT | SLOPE | H |
|---|---|---|---|---|
| 0.2500 | -0.992 | 0.777 | -0.119 | 1.030 |

Table 29. Data for "h" and Flow Calculation, 1/4" Spacing, 2 $cfm/ft^2$ Ventilation

PRESSURE DROP ACROSS NOZZLE= 0.7460, DEWPOINT TEMPERATURE= 24.00, BARCMETRIC PRESSURE= 28.70

| HOUR | TA | TF | TB | T4 | T8 | T12 | ABT | TP | TN | TC | TIN | T1 | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 74.30 | 94.10 | 95.90 | 96.62 | 96.98 | 96.44 | 96.26 | 94.10 | 87.26 | 77.00 | 84.20 | 86.74 | 90.42 | 0.000 |
| 2 | 75.20 | 86.72 | 94.10 | 90.68 | 93.38 | 94.46 | 92.23 | 92.48 | 88.88 | 75.38 | 77.00 | 81.64 | 87.06 | -0.576 |
| 3 | 74.66 | 85.10 | 92.84 | 88.34 | 91.40 | 92.84 | 90.39 | 91.04 | 87.44 | 75.02 | 76.64 | 80.70 | 85.87 | -0.833 |
| 4 | 74.84 | 83.66 | 91.40 | 86.54 | 89.42 | 91.22 | 88.68 | 89.42 | 86.18 | 75.38 | 76.28 | 79.90 | 84.66 | -1.061 |
| 5 | 75.02 | 82.22 | 89.78 | 85.10 | 87.98 | 89.78 | 87.21 | 88.16 | 85.64 | 75.02 | 76.10 | 78.96 | 83.56 | -1.246 |
| 6 | 74.84 | 81.50 | 88.34 | 83.84 | 86.36 | 88.34 | 85.86 | 86.72 | 84.20 | 75.20 | 75.92 | 78.56 | 82.64 | -1.441 |
| 7 | 74.30 | 80.78 | 87.08 | 82.94 | 85.28 | 86.90 | 84.76 | 85.28 | 82.94 | 74.66 | 75.38 | 78.02 | 81.65 | -1.547 |
| 8 | 74.30 | 80.06 | 85.82 | 81.86 | 84.02 | 85.64 | 83.61 | 84.20 | 82.22 | 74.48 | 75.20 | 77.26 | 80.73 | -1.684 |
| 9 | 74.30 | 79.16 | 84.56 | 80.96 | 83.12 | 84.38 | 82.58 | 83.30 | 81.50 | 74.48 | 74.84 | 76.68 | 79.99 | -1.838 |

| SPACE | "R" | INTERCEPT | SLOPE | H |
|---|---|---|---|---|
| 0.2500 | -0.991 | 0.731 | -0.176 | 1.516 |

Appendix D. Continued

Table 30. Data for "h" and Flow Calculation, 1/4" Spacing, 2 cfm/ft$^2$ Ventilation

PRESSURE DROP ACROSS NOZZLE= 0.7500, DEWPOINT TEMPERATURE= 30.00, BAROMETRIC PRESSURE= 28.70

| HOUR | TA | TF | T8 | T4 | T8 | T12 | ABT | TP | TN | TC | TIN | T1 | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 77.54 | 100.76 | 99.50 | 100.76 | 100.22 | 99.50 | 100.15 | 100.04 | 97.16 | 56.80 | 99.32 | 98.42 | 99.23 | 0.000 |
| 2 | 76.46 | 90.68 | 97.52 | 94.64 | 96.98 | 97.88 | 95.90 | 96.26 | 93.02 | 77.00 | 78.98 | 84.42 | 90.34 | -0.568 |
| 3 | 76.64 | 88.16 | 96.26 | 91.94 | 95.00 | 96.44 | 93.90 | 94.46 | 90.50 | 77.00 | 78.44 | 83.26 | 88.86 | -0.807 |
| 4 | 76.64 | 86.54 | 94.82 | 89.60 | 92.84 | 94.64 | 91.94 | 92.84 | 89.78 | 77.36 | 78.26 | 82.46 | 87.65 | -1.070 |
| 5 | 76.64 | 85.10 | 93.20 | 88.16 | 91.22 | 93.20 | 90.43 | 91.22 | 88.70 | 77.18 | 78.08 | 81.12 | 86.17 | -1.188 |
| 6 | 76.82 | 84.20 | 91.76 | 86.54 | 89.42 | 91.40 | 88.83 | 89.78 | 87.44 | 77.18 | 78.08 | 81.12 | 85.45 | -1.469 |
| 7 | 77.00 | 83.30 | 90.14 | 85.82 | 87.98 | 90.14 | 87.66 | 88.34 | 86.54 | 77.18 | 78.38 | 83.54 | 84.44 | -1.584 |
| 8 | 77.00 | 82.58 | 88.88 | 84.56 | 86.90 | 88.52 | 86.43 | 87.08 | 85.10 | 77.18 | 77.72 | 79.96 | 83.52 | -1.745 |
| 9 | 79.52 | 81.86 | 87.62 | 83.66 | 85.82 | 87.44 | 85.41 | 86.18 | 84.74 | 77.90 | 78.80 | 79.52 | 82.85 | -1.910 |
| 10 | 77.54 | 81.68 | 86.54 | 83.12 | 84.92 | 86.18 | 84.58 | 85.10 | 83.66 | 78.08 | 78.44 | 79.70 | 82.40 | -2.057 |

| SPACE | "R" | INTERCEPT | SLOPE | H |
|---|---|---|---|---|
| 0.2500 | -0.996 | 0.767 | -0.186 | 1.608 |

Table 31. Data for "h" and Flow Calculation, 1/4" Spacing, 2 cfm/ft$^2$ Ventilation

PRESSURE DROP ACROSS NOZZLE= 0.7500, DEWPOINT TEMPERATURE= 33.20, BAROMETRIC PRESSURE= 28.70

| HOUR | TA | TF | T8 | T4 | T8 | T12 | ABT | TP | TN | TC | TIN | T1 | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 77.54 | 100.94 | 100.04 | 100.94 | 100.40 | 100.04 | 100.47 | 100.76 | 97.52 | 104.18 | 102.92 | 102.33 | 101.54 | 0.000 |
| 2 | 76.64 | 92.12 | 98.42 | 95.90 | 98.24 | 98.78 | 97.05 | 97.34 | 94.10 | 77.54 | 79.52 | 85.54 | 91.44 | -0.476 |
| 3 | 76.82 | 89.24 | 97.16 | 93.02 | 96.08 | 97.34 | 94.91 | 95.54 | 91.58 | 77.36 | 78.80 | 83.62 | 89.58 | -0.714 |
| 4 | 77.18 | 87.44 | 95.72 | 90.86 | 93.92 | 95.72 | 93.02 | 93.92 | 90.50 | 77.36 | 78.62 | 83.04 | 88.48 | -0.971 |
| 5 | 77.54 | 86.00 | 94.10 | 89.24 | 92.12 | 94.10 | 91.38 | 92.30 | 89.24 | 77.36 | 78.62 | 81.88 | 87.09 | -1.138 |
| 6 | 77.00 | 84.74 | 92.66 | 87.62 | 90.50 | 92.48 | 89.82 | 90.68 | 88.34 | 77.36 | 78.44 | 81.30 | 65.59 | -1.329 |
| 7 | 76.82 | 83.84 | 91.04 | 86.18 | 88.88 | 90.86 | 88.34 | 89.24 | 86.72 | 76.82 | 77.72 | 80.76 | 85.00 | -1.533 |
| 8 | 76.46 | 82.76 | 89.60 | 85.28 | 87.62 | 89.24 | 87.08 | 87.80 | 85.82 | 77.00 | 77.72 | 79.78 | 83.79 | -1.624 |
| 9 | 76.46 | 82.22 | 88.16 | 84.02 | 86.36 | 88.16 | 85.93 | 86.54 | 84.92 | 76.82 | 77.36 | 79.60 | 83.07 | -1.865 |
| 10 | 76.46 | 81.50 | 87.08 | 83.12 | 85.10 | 86.90 | 84.85 | 85.46 | 83.84 | 76.82 | 77.36 | 79.02 | 82.24 | -1.943 |

| SPACE | "R" | INTERCEPT | SLOPE | H |
|---|---|---|---|---|
| 0.2500 | -0.994 | 0.821 | -0.181 | 1.561 |

Table 32. Oata for "h" and Flow Calculation, 1/4" Spacing, 3 cfm/ft$^2$ Ventilation

PRESSURE ORDP ACROSS NOZZLE= 1.4590, OEWPOINT TEMPERATURE= 36.00, OARUMETRIC PRESSURE= 29.05

| HOUR | TA | TF | T8 | T4 | T8 | T12 | ABT | TP | TN | TC | TIN | T1 | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 76.28 | 99.14 | 98.78 | 99.50 | 99.32 | 98.96 | 95.10 | 99.68 | 96.8C | 59.5C | 100.22 | 99.38 | 99.53 | 0.000 |
| 3 | 74.66 | 85.64 | 94.28 | 89.60 | 93.20 | 94.82 | 91.89 | 91.94 | 89.06 | 74.66 | 76.10 | 0).34 | 86.14 | -0.818 |
| 4 | 74.84 | 84.02 | 92.66 | 87.44 | 90.86 | 93.02 | 89.91 | 89.56 | 87.08 | 75.02 | 76.10 | 80.12 | 85.04 | -1.065 |
| 5 | 75.02 | 82.50 | 90.60 | 85.64 | 88.70 | 90.86 | 87.96 | 88.16 | 85.82 | 75.20 | 76.10 | 79.14 | 83.65 | -1.283 |
| 6 | 75.56 | 81.68 | 89.06 | 84.20 | 87.26 | 89.)6 | 86.47 | 86.54 | 84.56 | 75.56 | 76.20 | 78.92 | 82.73 | -1.481 |
| 7 | 74.48 | 80.96 | 87.44 | 83.12 | 85.46 | 07.26 | 85.01 | 65.10 | 83.48 | 75.02 | 75.74 | 77.8C | 81.45 | -1.606 |
| 8 | 75.02 | 80.06 | 85.82 | 82.22 | 84.20 | 85.82 | 83.79 | 83.84 | 82.22 | 75.02 | 75.74 | 77.80 | 83.02 | -1.821 |
| 9 | 75.38 | 79.16 | 84.74 | 81.14 | 83.12 | 84.56 | 82.69 | 82.58 | 81.14 | 75.20 | 75.56 | 76.82 | 79.70 | -1.874 |

| SPACE | "R" | INTERCEPT | SLOPE | H |
|---|---|---|---|---|
| 0.2500 | -0.990 | 0.705 | -0.179 | 1.543 |

Table 33. Data for "h" and Flow Calculation, 1/4" Spacing, 3 cfm/ft$^2$ Ventilation

PRESSURE ORDP ACROSS NOZZLE= 1.4590, OEWPOINT TEMPERATURE= 36.00, BAROMETRIC PRESSURE= 29.03

| HOUR | TA | TF | TB | T4 | T8 | T12 | ABT | TP | TN | TC | TIN | T1 | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 75.74 | 95.90 | 91.40 | 94.82 | 93.02 | 91.40 | 93.22 | 93.74 | 91.58 | 101.12 | 100.58 | 98.11 | 95.92 | 0.000 |
| 3 | 74.30 | 83.12 | 89.24 | 86.36 | 88.70 | 89.60 | 87.71 | 87.62 | 85.1C | 74.30 | 75.33 | 78.82 | 83.22 | -0.801 |
| 4 | 74.3) | 81.68 | 87.98 | 84.38 | 86.90 | 88.34 | 86.11 | 86.18 | 83.84 | 74.30 | 75.20 | 78.24 | 82.21 | -1.038 |
| 5 | 73.76 | 80.24 | 86.54 | 82.76 | 85.10 | 86.90 | 84.54 | 84.56 | 82.58 | 74.12 | 74.66 | 77.46 | 81.02 | -1.244 |
| 6 | 73.40 | 79.34 | 85.28 | 81.50 | 83.64 | 85.28 | 83.23 | 83.12 | 01.50 | 73.76 | 74.30 | 76.54 | 79.83 | -1.371 |
| 7 | 73.04 | 78.44 | 83.84 | 80.42 | 82.40 | 84.02 | 81.99 | 81.86 | 80.42 | 73.58 | 73.94 | 75.78 | 78.82 | -1.513 |
| 8 | 73.76 | 77.90 | 82.58 | 79.52 | 81.32 | 82.76 | 80.96 | 80.78 | 79.52 | 73.58 | 73.94 | 75.78 | 78.28 | -1.719 |
| 9 | 73.94 | 77.18 | 81.50 | 78.62 | 80.42 | 81.50 | 79.97 | 80.06 | 78.00 | 73.76 | 74.12 | 75.38 | 77.72 | -1.931 |
| 10 | 73.58 | 76.64 | 80.42 | 77.72 | 79.52 | 80.42 | 79.05 | 78.98 | 78.08 | 73.58 | 73.94 | 75.2C | 77.09 | -2.110 |
| 11 | 73.22 | 76.28 | 79.70 | 77.36 | 78.44 | 79.70 | 78.37 | 78.26 | 77.36 | 73.40 | 73.76 | 74.44 | 76.35 | -2.122 |
| 12 | 73.22 | 75.92 | 78.80 | 77.00 | 78.08 | 78.98 | 77.85 | 77.72 | 76.82 | 73.40 | 73.58 | 74.44 | 76.C8 | -2.269 |

| SPACE | "R" | INTERCEPT | SLOPE | h |
|---|---|---|---|---|
| 0.2500 | -0.993 | 0.681 | -0.164 | 1.414 |

Appendix D. Continued

Table 34. Data for "h" and Flow Calculation, 1/4" Spacing, 3 cfm/ft$^2$ Ventilation

PRESSURE DROP ACROSS NOZZLE= 1.4600. DEWPOINT TEMPERATURE= 34.50. BAROMETRIC PRESSURE= 29.00

| HOUR | TA | TF | T3 | T4 | T8 | T12 | ABT | TP | TN | TC | TIN | T1 | T | ABT-T/ABT(1)-T |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | 75.32 | 97.34 | 97.16 | 97.70 | 97.52 | 97.16 | 97.41 | 97.88 | 95.36 | 101.30 | 99.86 | 99.45 | 98.66 | 0.000 |
| 1 | 73.76 | 90.68 | 95.90 | 94.02 | 96.08 | 96.44 | 95.16 | 94.64 | 91.22 | 75.02 | 76.82 | 83.60 | 89.12 | -0.317 |
| 2 | 73.76 | 86.90 | 94.46 | 91.22 | 93.92 | 94.82 | 92.66 | 92.48 | 89.42 | 74.12 | 75.74 | 80.96 | 86.72 | -0.587 |
| 3 | 73.94 | 84.38 | 92.66 | 88.16 | 91.40 | 93.02 | 90.27 | 90.32 | 87.62 | 73.94 | 75.38 | 79.62 | 84.97 | -0.852 |
| 4 | 74.12 | 82.76 | 90.68 | 86.00 | 89.24 | 91.04 | 88.25 | 88.34 | 85.82 | 73.94 | 75.20 | 78.46 | 83.40 | -1.061 |
| 5 | 73.94 | 81.50 | 89.24 | 84.38 | 87.26 | 89.24 | 86.56 | 86.54 | 84.56 | 73.94 | 74.84 | 77.88 | 82.21 | -1.251 |
| 6 | 74.12 | 80.24 | 87.26 | 82.94 | 85.46 | 87.44 | 84.90 | 85.10 | 83.30 | 73.94 | 74.84 | 77.30 | 81.20 | -1.478 |
| 7 | 75.74 | 79.70 | 85.82 | 81.86 | 84.29 | 85.82 | 83.66 | 83.84 | 82.40 | 75.38 | 75.92 | 77.58 | 80.71 | -1.734 |
| 8 | 75.74 | 79.52 | 84.74 | 80.96 | 82.94 | 84.56 | 82.65 | 82.76 | 81.50 | 76.10 | 76.28 | 77.72 | 80.24 | -1.965 |
| 9 | 75.92 | 78.98 | 83.48 | 80.24 | 82.22 | 83.30 | 81.75 | 81.68 | 80.78 | 75.92 | 76.28 | 77.54 | 79.61 | -2.120 |
| 10 | 75.20 | 78.44 | 82.40 | 79.70 | 81.14 | 82.40 | 80.91 | 80.96 | 80.06 | 75.20 | 75.74 | 76.82 | 78.89 | -2.214 |

| SPACE | "R" | INTERCEPT | SLOPE | H |
|---|---|---|---|---|
| 0.2500 | -0.996 | 0.843 | -0.216 | 1.865 |

Appendix E. Statistical Analysis of Calculated "h" Values

Tests: I. $H_o$: $\mu_{1/4} = \mu_{3/16} = \mu_{1/8} = \mu_{space}$ vs. $H_A$: some $\mu_j \neq \mu_{space}$

II. $H_o$: $\mu_1 = \mu_2 = \mu_3 = \mu_{flow}$ vs. $H_A$: some $\mu_i \neq \mu_{flow}$

III. $H_o$: $\mu_{ij} = \mu_{combination}$ vs. $H_A$: some $\mu_{ij} \neq \mu_{combination}$

$H_o$ is the null hypothesis, $H_A$ is the alternate hypotheses, $\mu$ is mean of values

From the data presented in Table 3, page 44:

Uncorrected SS total = 63.9525
Uncorrected SS spacing = 60.3419
Uncorrected SS flow = 63.2014
Uncorrected SS combination = 63.6706
Uncorrected SS error = 63.9525 − 63.6706 = 0.2819
Correction factor = 59.9874

Table 35. Analysis of Variance for Calculated h Values for the Model Tests

| Source | df | SS | MS | F* | F.05 | $\hat{\alpha}$ |
|---|---|---|---|---|---|---|
| Spacing | 2 | .3545 | .17725 | 11.318 | 3.55 | < .5% |
| Flow | 2 | 3.2140 | 1.6070 | 102.611 | 3.55 | < .5% |
| Interaction | 4 | .1147 | .02868 | 1.831 | 2.93 | $10 < \hat{\alpha} < 25$ |
| Error | 18 | .2819 | .01566 | | | |
| Total | 26 | 3.9651 | | | | |

*$MS_{error}$ was used for the denominator in all F-tests.

Reject $H_{oI}$ and $H_{oII}$ at the $\alpha = .05$ level.
Fail to reject $H_{oIII}$ at the $\alpha = .05$ level.

Conclude that spacing and flow rates have significant effect on "h" values, but that their interaction does not at the 5% level.

Least Significant Difference (LSD) test:

$$LSD = t_{(.05/2,18)}\sqrt{\frac{2\ MSerror}{n}}$$

from tables: $t_{(.05/2.18)} = 2.101$

for interaction, n = 3: $LSD = 2.101\sqrt{\frac{2(.01566)}{3}} = .2147$

for spacing and flow, n = 9

$$LSD = 2.101\sqrt{\frac{2(.01566)}{9}} = .0281$$

Appendix E. continued

Interactions

| Spacing | Flow Rate cfm/f$^2$ collector | Average "h" | |
|---|---|---|---|
| 1/4" | 1 | .997 | a |
| 1/8 | 1 | 1.010 | a |
| 3/16 | 1 | 1.125 | a |
| 1/8 | 2 | 1.403 | b |
| 1/4 | 2 | 1.562 | bc |
| 3/16 | 2 | 1.666 | c |
| 1/8 | 3 | 1.735 | c |
| 1/4 | 3 | 1.762 | c |
| 3/16 | 3 | 2.156 | d |

| Spacing | | | Flow Rate | | |
|---|---|---|---|---|---|
| 1/8" | $\bar{h}$ = 1.382 | e | 1 cfm/f$^2$ | $\bar{h}$ = 1.044 | i |
| 1/4 | 1.440 | f | 2 | 1.544 | j |
| 3/16 | 1.649 | g | 3 | 1.884 | k |

Means with like letters are not significantly different at the $\alpha$= .05 level.

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# NOCTURNAL COOLING OF A SOLAR COLLECTOR-STORAGE UNIT

by

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submitted in partial fulfillment of the

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ABSTRACT

This paper reports on research done to study the performance of a solar collector-storage unit managed for summertime use. It was desired to know if the storage could be cooled at night by moving ambient air, at a lower temperature than the storage, through the storage. Then, by passing the ventilating air through the storage during the day, cool air could be drawn into the building. Economic considerations were included in deciding whether this would be feasible. Also, it was desired to know which combination of air flow rate and spacing of the dry-stacked concrete blocks that make up the storage was optimal. Finally, could the operation of this unit in the spring and fall, when daytime temperatures are above swine comfort levels while nighttime temperatures are below these levels, result in a more comfortable environment inside the building.

Tests were conducted on a full-sized unit located at the Kansas State University Swine Experimental Farm and with a model representing a section of collector-storage wall two feet square. Preliminary tests showed little effect of air flow rates on cooling performance with the full-sized unit.

Tests were then run with the model, using air flow rates representing one, two and three cubic feet per minute per square foot of collector surface, and block spacings of 1/8, 3/16 and 1/4 inch. A value for the convective heat transfer coefficient, h, was calculated for each combination and used for comparison. A high h value was desired because only 10 to 12 hours is available each night for cooling purposes, therefore the cool down should be as quick as possible.

The analysis showed that the best combination tested was 3/16 inch spacing with three cfm per square foot flow rate. There was no statistical difference between the 1/8 and 1/4 inch spacings at each flow rate. Higher flow rates had not been used due to the indications gathered from previous research with the full-sized unit.

Further tests with the full-sized unit, which was constructed with 3/16 inch spacings, showed that the cooling performance with three rather than two cfm per square foot was better. It was found that the ventilating air temperature could be reduced up to ten degrees Fahrenheit from the peak ambient temperature. Such a decrease in temperature could create a much more comfortable environment for swine housed in the building. Also, if the ventilating air was passed through the storage continuously, the nighttime air temperature in the building could be raised above the ambient conditions. This would be important in fall and spring operation.

The cooling capabilities of the system are secondary to its primary use of heating the ventilating air during the heating season. Thus, the system's cost does not need to be justified solely by its cooling performance. In judging the efficiency of the cooling, a hypothetical Coefficient of Performance (COP) was calculated as 20.4. This compares favorably with conventional air cooling equipment whose COP normally fall in the range of two to three. Some changes could be made in the system, through more efficient fans for example, so that the COP could be raised.