DESIGN OF A WIND-POWERED COOLING SYSTEM FOR AN APPLE STORAGE FACILITY

by

John Clisby Blanton

Thesis submitted to the Graduate Faculty of the

Virginia Polytechnic Institute and State University

in partial fulfillment of the requirements for the degree of

MASTER OF SCIENCE

in

Mechanical Engineering

APPROVED:

H. L. Moses, Chairman

W. F. O'Brien, Jr.

D. H. Vaughan

August 1977

Blacksburg, Virginia

ACKNOWLEDGMENTS

The author wishes to express his sincere gratitude and appreciation to the members of his advisory committee, whose help and guidance were invaluable. Thanks are also due to his fellow graduate students and , for their help and friendship.

is acknowledged for her excellent typing of this thesis.

The author, last of all, wishes to express appreciation to his wife, , for her infinite patience, continued encouragement, and moral support, without which this thesis would not have been completed.

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1. INTRODUCTION

In recent years much attention has been given to the topic of energy supply. Dwindling fossil fuel reserves coupled with growing concern over the safety of nuclear power generation have resulted in a need for development of alternative energy sources. One of the most promising of these sources is wind power. Von Arx [1]* estimates the available wind energy in the earth's atmosphere as 10^6 megawatts, or approximately twice the total electric generating capacity of the United States power industry [2]. He further estimates the available wind power to be 10 times greater than the available hydroelectric power. Despite these impressive statistics, wind power has not been harnessed to a significant degree.

Several attempts at large scale utilization of wind energy have been made with varying degrees of success. The largest wind generator ever built was constructed near Rutland, Vermont during World War II. The unit was rated at 1.25 megawatts at a wind speed of 13.4 m/sec (30 mph). Its two-bladed horizontal-axis rotor was 53 m (175 ft) in diameter and was mounted on a 33-m (110-ft) tower. It was operated from 1941 until 1945 when a broken blade and wartime economics forced abandonment of the project [3]. Other units built and operated included a 0.8 MW unit in France [4], 0.1 MW units in Russia and Great Britain [5], and a 0.2 MW unit in Denmark. The Danish machine is the largest system still standing.

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* Numbers in brackets indicate references at the end of the thesis.

Smaller wind machines have been used and proven successful in a number of applications requiring a mechanical work source. The familiar farm-type windmill has been found very effective as a water pump drive, as have the famous Dutch wind machines. For larger machines, however, it is presently impractical to attempt utilization of the windmill mechanical output other than by an electric generator because of the power transmission difficulties.

The applications described above involve horizontal-axis wind machines. Studies are also being made to develop the capabilities of vertical-axis wind generators, particularly the "eggbeater" style rotor devised by Darrieus [6]. These machines have not as yet proved as effective as conventional horizontal-axis rotors, but further study of their potential has been recommended by Vance [7].

Much of the research being done in the United States at this time is concerned with the 0.1 MW unit in operation at the NASA-Lewis Plum Brook test area at Sandusky, Ohio. This unit has a rotor 38 m (125 ft) in diameter mounted on a 30-m (100-ft) tower. Hewson [8] describes this unit in some detail and also outlines the federal government's wind energy program.

The purpose of the present investigation is to study the utilization of a small wind generator for the production of electric power. The generator output is to be applied toward the electrical needs of a refrigerated fruit storage facility. The three major parts of the investigation include wind generator performance study and analysis, storage facility design and construction, and cooling system design and installation. This particular study is primarily concerned with the third part of the investigation, but some consideration of the other parts is necessary and are discussed. Initial portions of the study involve considerations of the fruit storage requirements, wind generator theory and application, and refrigeration system requirements. A more detailed study of the refrigeration system is then undertaken with the objective of designing a complete, self-sustained system. Special considerations were necessary due to the unique nature of the power source, and the completed system thus represents an innovative design compared to conventional systems.

2. PRELIMINARY CONSIDERATIONS

The fruit selected as the subject of the study was apples. The capacity of the storage facility was chosen to be approximately 1000 bushels. Although during the initial phases of the design the windmill had not been selected, the intention was to purchase the largest commercially available unit, subject to price and delivery considerations. The rated maximum capacity of the wind generator was anticipated to be approximately 6 to 10 kilowatts.

2.1 APPLE STORAGE CHARACTERISTICS

An excellent source of information on the storage of apples is provided by Patchen [9]. Of primary interest were the recommended storage conditions, particularly the temperature and humidity standards and approximate storage life. This information for several different apple varieties is summarized in Table 1.

The standard storage container for apples is the bushel box, the capacity of which is approximately 18.1 Kg (40 lb). The outside dimensions of the box are 457 x 356 x 305 mm (18 x 14 x 12 in.). In a typical fruit storage the boxes are stacked onto pallets of 36 each (6 layers of 6 boxes each). The stacking arrangement exposes at least one surface of each box to the room air, thus providing a free cooling surface.

Variety	Storage Temperature C (F)	Relative Humidity %	Freezing Temperature C (F)	Specific Heat $\frac{KJ}{Kg \cdot K}$ $(\frac{btu}{lb^\circ R})$	Approx. Length of Storage Period Months
Delicious	-1 (30)	85-90	-1.5 (29.3)	3.64 (0.87)	4-8
Golden Delicious	-1 (30)	85-90	-1.5 (29.3)	3.64 (0.87)	4-8
Jonathan	2 (35)	85-90	-1.5 (29.3)	3.64 (0.87)	3-6
Winesap	-1 (30)	85-90	-1.7 (29.0)	3.64 (0.87)	5-8
York	-1 (30)	85-90	-1.5 (29.3)	3.64 (0.87)	4-8
McIntosh	2 (36)	85-90	-1.5 (29.3)	3.64 (0.87)	4-8
Stayman	-1 (30)	85-90	-1.5 (29.3)	3.64 (0.87)	4-8
Yellow Newtown	3 (38)	85-90	-1.5 (29.3)	3.64 (0.87)	5-8

Table 1. Recommended storage conditions for different varieties of apples (from reference 9).

2.2 CLIMATE CONSIDERATIONS

Apples are harvested in the late summer or early fall in most parts of the country [9]. In Virginia, for example, most varieties of apples are picked from the middle of September through the latter part of October [10]. The length of the storage period (see Table 1) may vary from 3 to 8 months depending on the variety of apples stored. During this period a wide variation in ambient conditions is likely to occur. For example, the monthly mean average temperature in Blacksburg, Va., varies 18 C (32 F) and the extreme temperature difference (highest high to lowest low) can range as high as 56 C (100 F) between the months of September and January. The mean average temperatures for the projected storage period months are shown in Table 2.

Month	Mean Average Temperature C (F)
September	17.5 (63.5)
October	11.8 (53.3)
November	6.2 (43.2)
December	0.8 (33.4)
January	-0.3 (31.4)
February	1.4 (34.6)
March	4.7 (40.4)
April	10.9 (51.7)

Table 2. Mean average monthly temperature for Blacksburg, Va., based on 1955-64 (data compiled by VPI&SU Dept. of Agricultural Eng.).

2.3 WIND GENERATOR CHARACTERISTICS

The power output that can be obtained from a wind generator can be calculated using the equation:

$$P = 1/2\ (0.593)\ \eta\rho AV^3 \qquad (1)$$

where η is the efficiency, ρ is the air density, A is the area swept by the windmill blades, and V is the wind velocity [8]. The numerical constant 0.593 is the maximum theoretical fraction of the wind's kinetic energy that can be extracted to produce useful work, as shown by Betz [11]. The phenomena limiting the energy extraction is spillage. Values of the efficiency η for a typical wind generator are approximately 0.65 to 0.80 [8].

From equation 1, it can be seen that the power output of a wind generator is proportional to the swept area (and thus to the square of the rotor diameter) and to the cube of the wind velocity. Doubling the rotor diameter and wind speed would thus increase the power output by a factor of 32.

A typical power versus wind speed characteristic for a wind generator is shown in Fig. 1. Note that the power output increases with increasing wind speed until point "a" is reached, when the machine is operating at its rated capacity. At greater wind speeds the power output is generally held constant at this value by adjusting the blade pitch or some other appropriate method. At point "b", referred to as the "furling point" [8], the unit is shut down to avoid

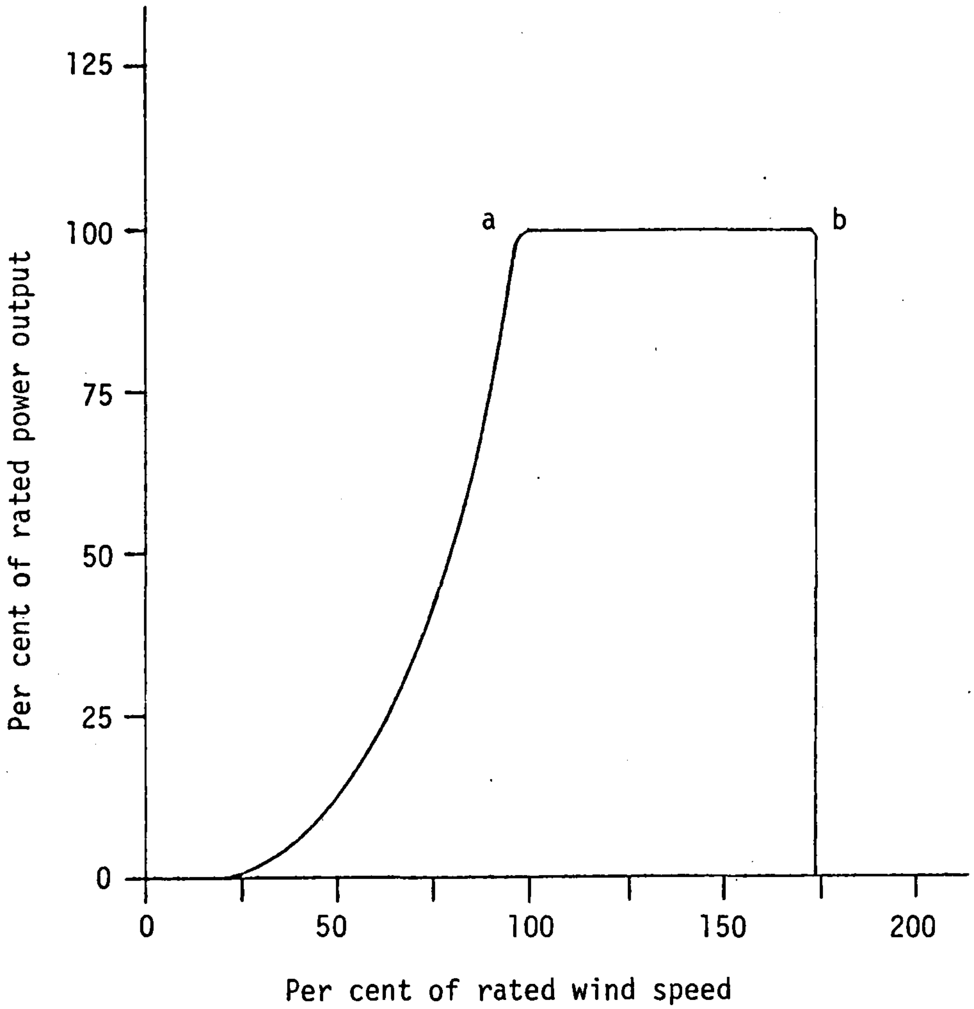


Figure 1. Typical wind generator characteristic
(from reference 8)

damage. This is done by either feathering the blades or turning the machine out of the wind.

2.4 SITE SELECTION

The selection of a wind generator site is a crucial task, requiring careful collection and evaluation of wind data. In order to make accurate predictions of the wind energy available at a given site, it is necessary that the wind speeds be measured continuously and a time distribution of the wind speed obtained. Because the wind power is proportional to the cube of the velocity, using the average wind speed will result in underestimating the average power. Also, two sites with the same average winds may differ markedly in their average wind energy, with the more steady site being less preferable.

A useful parameter for calculating the average power available from wind data is the root-mean-cubed (rmc) wind speed, defined by:

$$V_{rmc} = \left[\frac{1}{n} \sum_{i=1}^{n} V_i^{3} \right]^{1/3} \tag{2}$$

where n is the number of wind samples over the time period being considered and the V_i's are individual wind speed samples. Knowing the rmc wind speed over a time period enables one to make a more accurate prediction of the average power available over a period using equation 1. The larger the number of samples, n, taken over a given time period, the more accurate the prediction becomes.

To make a complete evaluation of the wind data, it is necessary to have several years of continuous wind measurements available [8]. In many cases this is, of course, simply not practical, and a site must be selected on the basis of a short duration wind study. The result may be the selection of a less than optimal site, but more than likely the errors incurred in using the short term wind data will not prove disastrous.

Other factors will, of course, enter into the site selection, in particular considerations of land availability and convenience. The optimum site, as determined by wind surveys, may be at a location that is inconvenient to the personnel involved with the construction and operation of the facility. The site may also be located on land that is not available for use. The best overall site must therefore be selected, based on wind surveys, from the locations that are acceptable based on all other considerations.

2.5 STORAGE ROOM ARRANGEMENT

Proper arrangement of storage pallets, access aisleways, and refrigeration equipment in the storage building is important for three reasons: its effect on the total refrigeration requirements, the proper performance of the air distribution system, and the accessibility for loading, unloading, and maintenance of the apples.

The building cooling load is directly proportional to the surface area of the storage building. It is thus obvious that the most efficient building design would be the one with the smallest surface area per unit of storage capacity. Proper internal arrangement of the building can effectively reduce the external dimensions.

Certain standard pallet stacking arrangements are used to insure proper circulation of the cooling air [9]. As mentioned previously, the arrangement of the stacked boxes into a pallet insures that at least one surface of every box is exposed to the room air. To maintain this exposure the pallets are arranged in rows, leaving spacings of approximately 76 mm (3 in.) between pallets and 152 mm (6 in.) between the pallets and the room walls. The pallets may be stacked vertically if sufficient ceiling height is available.

Accessibility is extremely important in an apple storage facility. Means must be provided for convenient inspection and sampling of the entire contents and for simple loading and removal of apples from any part of the facility. This is especially important for applications involving the storage of several varieties of apples in the same facility. Situations are likely to occur requiring fruit stored earlier to be removed before the later varieties. Improper planning may result in unloading and reloading large quantities of apples simply to remove one pallet.

2.6 LOAD MATCHING

A fundamental consideration involved in wind generator applications is the most efficient method of loading the generator. The problem is different from that found in, for example, a conventional steam power plant. When the electrical load increases a control system will automatically supply more steam to the turbogenerator to compensate. With wind generators, of course, there is no such control capability.

For efficient operation the electrical load must be variable so that it can be matched to the generator output. Rechargeable electric storage batteries are ideally suited to this application as they are inherently capable of matching their charging rate to the power output of the device used to charge them. The primary disadvantages associated with storage batteries is their inefficiency and high cost. Typical watthour efficiency for a lead-acid storage battery at rated operation is 75 to 80 per cent. Nickel-Cadmium Alkali (Nicad) cells are slightly less efficient [12].

The preferable method of utilizing the electrical energy produced by the wind generator would be to load it directly with the energy consuming system components, thus avoiding the battery losses. Some provision, however, would have to be made to control the power consumption of the system components to match the power output of the wind generator. Conceptually, this could be done through the design and development of an automatic control system. Practically, however, such an undertaking is beyond the scope of this investigation. In order to simplify the system design and operation, it was decided to use electrical storage batteries to load the wind generator and to subsequently serve as a system power supply.

2.7 ENERGY STORAGE

The peak power demands for the storage facility and the peak power outputs of the wind generator will in general occur at different times. For this reason some type of energy storage is required to insure proper system operation. There are several means by which this may be done. The simplest method, considering that storage batteries have been chosen to load the wind generator, would be to provide sufficient battery capacity to satisfy the storage needs. This solution would be simple and convenient, but because batteries are inefficient and expensive, as mentioned before, an additional storage method was sought.

Mentioned frequently in connection with wind energy systems are such energy storage devices as pump storage, compressed air storage, and flywheels [8]. These methods, however, are in general proposed as storage devices for larger wind energy systems. Fisher and Nephew [13] proposed an ice storage system as part of their peak-shaving home heating and cooling. A specially designed ice maker manufactured ice in the winter produced by energy extraction by a heat pump. The ice was melted in the summer to help reduce cooling costs. This type of "thermal" energy storage takes advantage of the latent heat of fusion of water to create a very high energy capacity-per-unit-volume. There are also no inefficiencies associated with storing and recovering energy. This type of system would adapt well to the wind energy-powered refrigeration system.

There is an underlying advantage, however, of electrical storage in batteries as opposed to thermal energy storage. The refrigeration unit used to make the ice will operate by converting units of electrical energy into units of thermal energy. A typical refrigeration machine might operate with a coefficient of performance of approximately 2, which means that 1 unit of electrical energy is converted into 2 units of thermal energy. Thus an energy storage for thermal energy would need to store twice the energy units as an electrical energy storage to provide the same effective capacity.

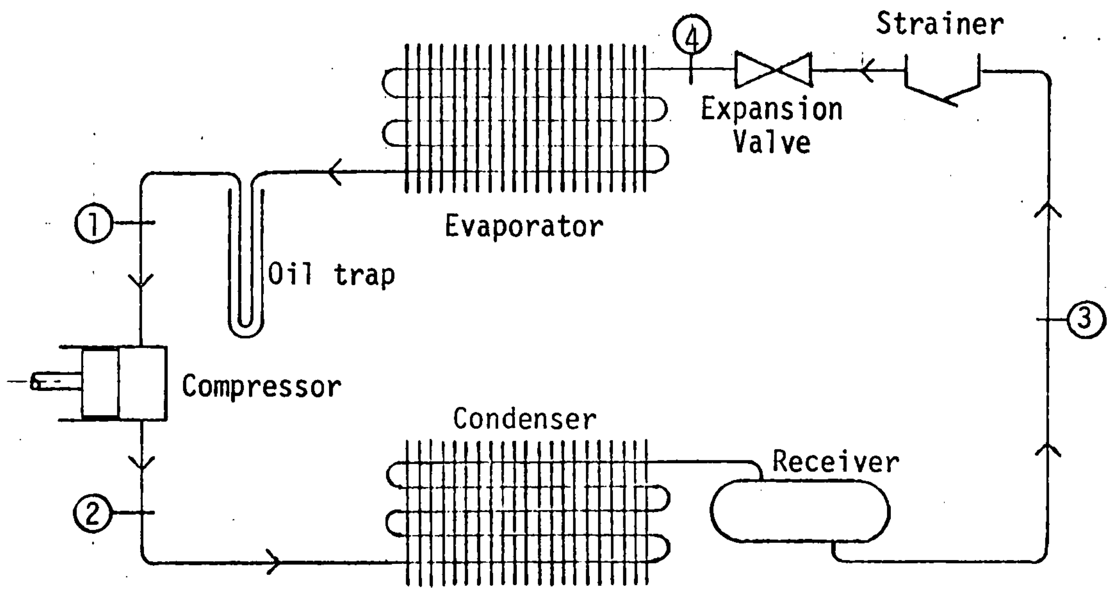
The storage system selected was a combination of electric and thermal energy devices. Some electrical storage is necessary to take care of situations where the wind generator output is greater than the power consumption of the cooling system. Some electrical storage may also be needed to provide for operation of devices not directly associated with the cooling system. It was planned, however, to use as small an electrical storage as possible in deference to a larger thermal energy storage, primarily in the interests of efficiency and cost.

2.8 REFRIGERATION UNIT

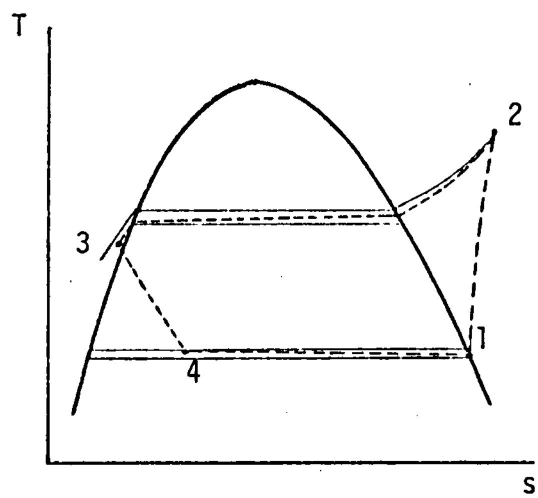
For simplicity and convenience a vapor compression refrigeration machine with Freon-12 refrigerant was selected. A schematic and a temperature-entropy diagram of the vapor compression cycle are shown in Fig. 2. For smaller commercially available systems the compressor, condenser, and receiver are assembled in a single unit referred to as a "condensing unit".

2.9 COOLING AIR DISTRIBUTION

To insure achievement of a uniform temperature distribution throughout the storage facility it is imperative that proper cooling air distribution be maintained. The temperature difference between the return and delivery air to the cooling system is referred to as



a) Cycle schematic.



b) Temperature-entropy diagram.

Figure 2. The vapor-compression refrigeration cycle.

the "split". Patchen [9] points out that the split is directly related to the volume of air circulated and the quantity of heat picked up in the room. For each 3520 W (12,000 Btu/hr or 1 ton) of refrigeration delivered, an air circulation rate of 1700 m^3/hr (1000 cfm) will result in a split of approximately 6 C (11 F). Air circulation systems for fruit storage facilities are normally designed to achieve this figure.

2.10 FINAL DESIGN CRITERIA

The final design criteria for the apple storage facility are summarized in Table 3.

Location	-	Blacksburg, Va.
Storage Capacity	-	1000 Bushels
Apple Varieties	-	Delicious, Golden Delicious, Staymen, Winesap, and York
Storage Temperature	-	-1°C (30°F)
Storage Humidity	-	85-90%
Power Source	-	Wind generator
Special Considerations	-	Thermal energy storage to be used to utilize power output. Wind generator to be loaded using batteries.
Refrigeration Unit	-	Vapor compression, Freon-12

Table 3. Summary of design criteria.

3. COOLING SYSTEM DESIGN

As was previously mentioned, this study is primarily concerned with the design and installation of the cooling system for the storage facility. The details of the site selection and building design will not be discussed, except for factors that affect the cooling system design.

Using the data for the building design, calculations were performed to determine the expected cooling loads for the facility. Based on these calculations the refrigeration unit was selected.

The design of the thermal energy storage is the most important step in the cooling system design. The storage capacity was estimated from the results of the cooling load calculations. The type of storage mechanism was then devised, with simplicity of operation emphasized.

The design of the air distribution system was based on the refrigeration unit capacity and pressure drop calculations for the various components through which the air is circulated. The power required to operate the circulating fan was determined from these calculations.

The criteria for the selection of the electrical storage battery capacity was somewhat arbitrary and is discussed in some detail.

Component interfacing, controls, and instrumentation are also discussed.

3.1 STORAGE BUILDING

The approximate dimensions and interior arrangement plan for the storage building are presented in Fig. 3. The capacity of the building is 1008 bushels. Note that there is ample aisle space allowed for accessibility. An area approximately 7.9 by 1.8 meters was allowed for installation of the thermal energy storage.

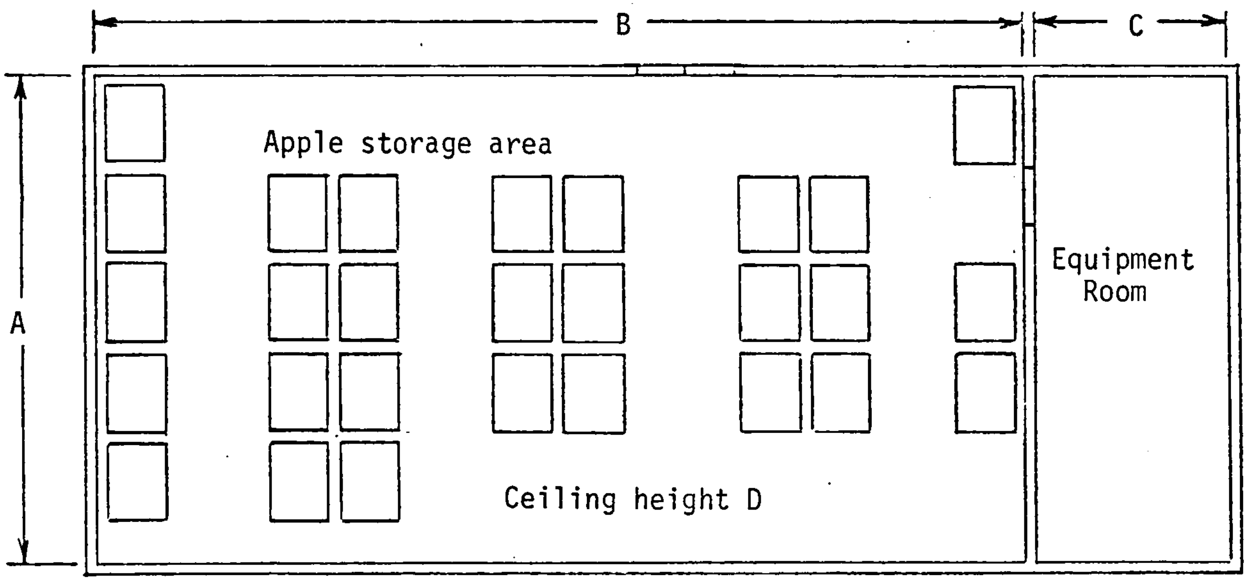
The design insulation heat transfer resistance values for the building are given below in Table 4.

Surface	Insulation Resistance	
	m^2-C/W	(ft^2-hr-F/Btu)
wall	3.94	30.0
ceiling	5.25	40.0
floor	3.55	27.0

Table 4. Approximate design insulation resistance for apple storage facility (from reference 14).

3.2 CALCULATION OF COOLING LOAD

To determine the proper sizes and capacities of the various cooling system components, it is necessary to calculate the expected cooling load. It is especially important in this application that this be done as accurately as possible, as proper equipment sizing will result in the most efficient operation. Conservative design



	m	ft
A	7.50	24.9
B	14.44	47.4
C	3.05	10.0
D	2.74	9.0

Figure 3. The apple storage facility

should be held to a minimum to insure operation of equipment at or near rated capacity as much as possible.

In determining the cooling load for the storage facility, there are two periods of interest to be considered: the initial cooling period, just after the apples are stored, and the steady state period, after the fruit temperature has been lowered to its storage value. The initial cooling period will in general have the higher average cooling load due to the additional load imposed by the removal of "field heat" from the apples. A detailed discussion of the various contributions to the total cooling load follows.

The most important contribution to the steady-state cooling load is the building heat gain, that is, the heat transferred due to the temperature difference between the outdoor and indoor air. In general this term is calculated using the equation:

$$Q_1 = \frac{A}{R} \Delta T \qquad (3)$$

where Q_1 is the building heat gain rate, A is the building surface area, R is the heat transfer resistance, and ΔT is the temperature difference between the outdoor and indoor air. Following general practice [9], the heat loss was calculated separately for the ceiling, walls, and floor. The temperature difference, ΔT, for the ceiling was taken to be 6 C (10 F) greater than for the walls due to high attic temperatures resulting from solar radiation. Because the ground temperature is generally somewhat cooler than the air, the temperature difference for the floor was taken to be 6 C (10 F) lower than for the walls. The overall heat transfer resistance, R, takes into consideration the combined effects of convection, conduction, and radiation and is thus a function of wall thickness and thermal conductivity, wall surface color and temperature, air temperatures and velocities, and other factors. Standard practice is to sum tabulated resistance values for the construction materials used and to include inner and outer film resistances. The temperature difference adjustment mentioned earlier is made to account for solar effects. Because building material resistances for this facility will be comparatively high, the film resistances can be neglected. Resistance values of some typical building materials are given in Table 5.

Building Material	Resistance per Unit Thickness $m^2 \cdot C/W/m$	$(ft^2 \cdot hr \cdot F/Btu/in)$
Plywood	6.46	1.25
Concrete	2.07	0.40
Insulation (Roll or loose)	19.4	3.75
Styrofoam	20.7	4.00
Fiberglass	15.5	3.00

Table 5. Heat transfer resistance values of some common building materials (from reference 14).

These values were taken into account in the calculation of the resistance values given in Table 4.

A second contribution to the overall cooling load is the heat given off by the fruit itself, called the heat of respiration. This heat given off is a strong function of the fruit temperature, as shown in Fig. 4. From a cooling load standpoint it is thus obvious that it is advantageous to store the fruit at as low a temperature as possible. It is also worth mentioning that the decay rate of the apples is similarly affected by temperature. The heat of respiration is calculated using the equation:

$$Q_2 = \dot{m}_{app}\, q_{resp} \tag{4}$$

where Q_2 is the heat of respiration, $\dot{m}_{app}$ is the mass of the apples, and q_{resp} is the specific heat of respiration, determined for a given fruit temperature from Fig. 4.

A third contribution to the total cooling load is the heat produced by incidental sources. These sources include motors, lights, and other electrical equipment, air infiltration through windows and doorways, workmen, forklifts, and other extraneous sources. A comparison of the magnitude of these sources is given in Table 6.

Source	Heat Produced
Workman	300 W (1000 Btu/hr)
Motors	0.9 W/W (3000 Btu/hr/HP)
Lights	1 W/W (3.4 Btu/hr/W)
Forklift	10^4 W (35000 Btu/hr)

Table 6. Incidental heat load contributions of several common sources (from reference 9).

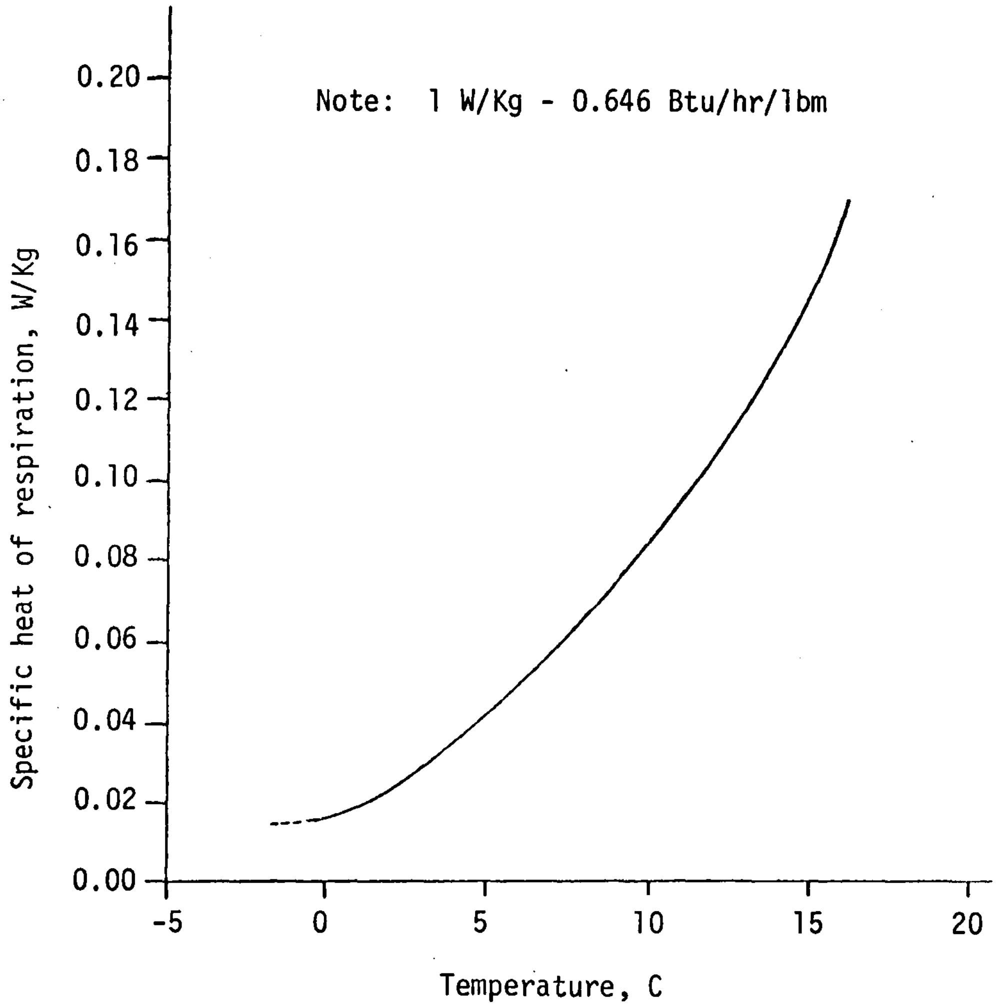


Figure 4. Heat of respiration for apples
(from reference 9)

By estimating the exposure time of these and other sources to the storage room, a calculation of the incidental heat load could be made. For this study, however, the incidental heat load will be simply estimated as a fixed percentage of the steady state load. It is, of course, advantageous to reduce the incidental load as much as possible by keeping doors closed, staying out of the room except when absolutely necessary, and avoiding using machinery in the room.

These three contributions, i.e., the building heat loss, the heat of respiration, and the incidental heat, make up the steady state cooling load. To estimate the maximum expected steady state cooling load for the 1008 bushel facility shown in Fig. 3, an outdoor temperature of 25 C (77 F) was assumed. From Fig. 4, the specific heat of respiration at the storage temperature of -1 C (30 F) is approximately 0.013 W/Kg (0.021 Btu/hr/lb). The calculations to determine the maximum steady state cooling load are summarized in Table 7.

During the initial cooling period, as previously mentioned, there is an additional contribution to the cooling load imposed by removal of "field heat" from the fruit. This can be expressed by the equation:

$$Q_4 = \frac{m_{app}\ C\ \Delta T_{field}}{t} \qquad (5)$$

where Q_4 is the field heat removal rate, C is the specific heat of apples, ΔT_{field} is the temperature difference between the apples as-loaded and as-stored, and t is the time duration of the field heat

1. Building heat gain (ΔT for walls = 26 C) using Eq. 3:

Walls: $\frac{120}{3.94}(26)$ =	792 W
Floor: $\frac{108}{3.55}(26-6)$ =	608 W
Ceiling: $\frac{108}{5.25}(26+6)$ =	660 W
Total building heat gain =	2060 W
2. Heat of respiration: (0.013)(1008)(18.2) =	240 W
Subtotal =	2300 W
3. Incidental Heat @ 10% of subtotal =	230 W
Total maximum steady state cooling load =	2530 W (8630 Btu/hr)

Table 7. Summary of calculations for maximum steady state cooling load.

* * *

1. Building heat gain (from Table 7) =	2060 W
2. Heat of respiration: (0.048)(1008)(18.2) =	880 W
Subtotal =	2940 W
3. Incidental heat @ 20% of subtotal =	590 W
4. Field heat: $\frac{(252)(18.2)(3.64)(1000)(14)}{(7)(24)(3600)}$ =	380 W
Total maximum initial period cooling load =	3910 W (13340 Btu/hr)

Table 8. Summary of calculations for maximum initial period cooling load.

removal. Note that there is a fixed quantity of heat to be removed and the longer the time period allowed for its removal, the lower the cooling rate. There are,however, two reasons for attempting to cool the apples as quickly as possible. If the fruit is allowed to remain at a higher temperature for a longer period of time, its storage life is decreased, and its heat of respiration is increased. For these reasons one week is cited as the longest acceptable cooling period [9].

To estimate the maximum cooling load during the initial cooling period an ambient temperature of 25 C (77 F) was again chosen. Because the loading schedule for the apples was not available, a four-week loading period was assumed where 252 bushels are loaded and cooled each week. The loading temperature was assumed as 13 C (55 F). The average specific heat of respiration over the initial cooling period is about 0.048 W/Kg (0.074 Btu/hr/lb) [9]. It was also assumed that due to increased activity the incidental heat would be doubled. The calculations for the maximum initial period cooling load are summarized in Table 8.

By varying the outdoor temperature the expected steady state cooling load for various months can be determined using the temperature data from Table 2. This variation is shown in Fig. 5. The effect of the additional load during the initial period is also shown, again using outdoor temperature data from Table 2.

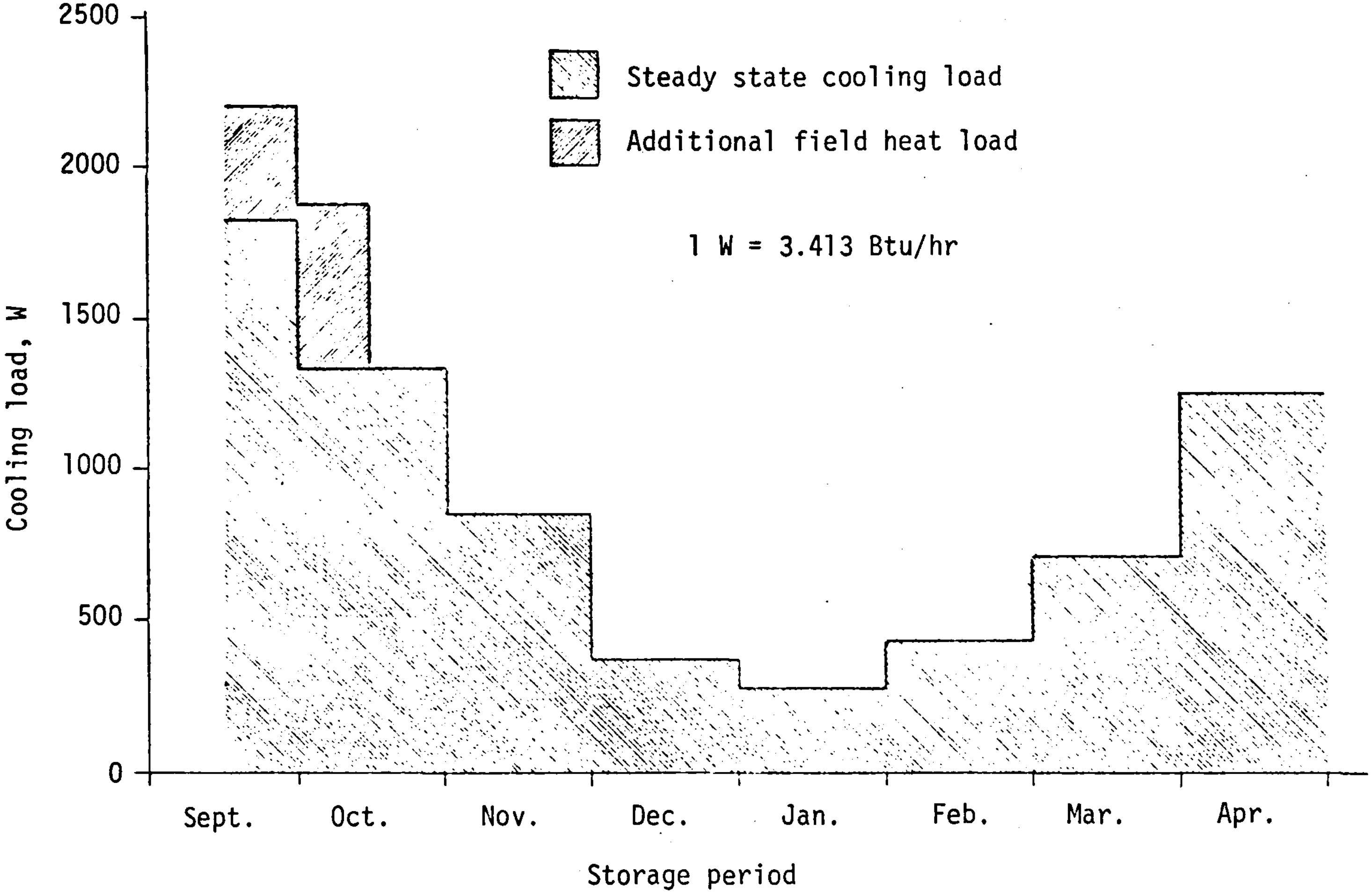


Figure 5. Cooling load calculation results.

3.3 THERMAL ENERGY STORAGE DESIGN

The capacity of the thermal energy storage was based on two considerations: (1) providing additional capacity for periods of high cooling loads, in particular the initial cooling period, and (2) providing a "peak-shaving" device, to store cooling capacity during periods of high energy availability and low loads and to provide cooling during periods of low energy availability.

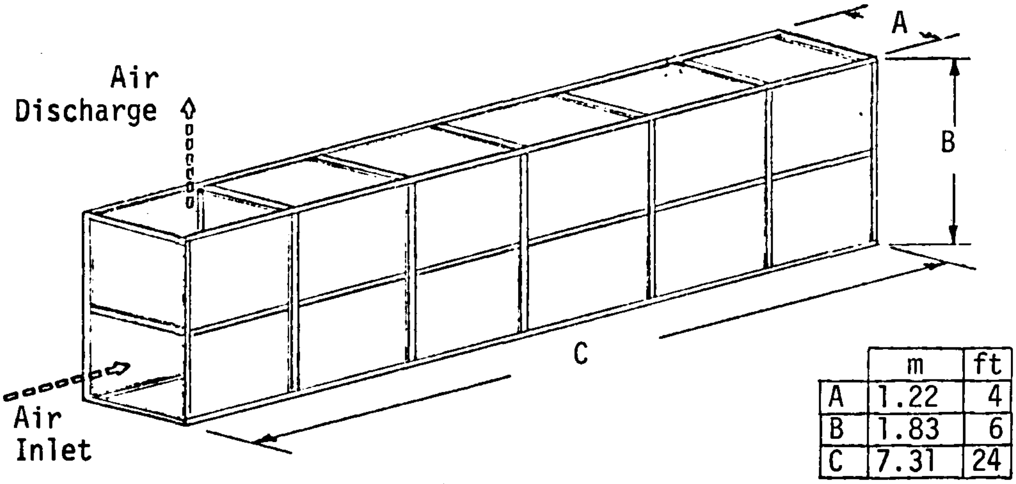
During the initial cooling period the cooling load is very high due to the additional load imposed by the removal of field heat, as previously noted. If a 17.5 C (63.5 F) average ambient temperature is assumed for the loading period (typical for September) the steady state and initial period cooling loads for the facility are approximately 1830 and 2950 W, respectively. Assuming that the windmill-produced energy is sufficient to carry the steady state portion of the load, the energy storage capacity required would be 1.9×10^5 W-hr (6.42×10^5 Btu).

The "peak-shaving" capacity provided must be sufficient to carry the cooling load for the longest anticipated period of wind energy shortage. The limited wind data available indicate that this period would be approximately 50 hours in length. For the average September cooling load of 1830 watts, the required "peak-shaving" capacity would be 9.2×10^4 W-hr (3.12×10^5 Btu), so that the total energy storage required is 2.8×10^5 W-hr (9.6×10^5 Btu). This capacity would provide complete capability for field heat removal and "peak-shaving" during the initial cooling period and for conservative storage during the steady state period.

As mentioned previously, the type of storage selected was a latent heat storage, the advantages of which have been discussed. Because the design storage temperature is -1 C (30 F), a dilute glycol solution in water can be used as the freezing medium. The operation of the storage unit is simple. Circulating air will be chilled by the refrigeration unit to a temperature lower than the design storage temperature and introduced to the storage unit. There the air will be warmed by heat transfer from the ice solution in the storage and introduced to the room. During periods in which the refrigeration unit is not operating, return air from the room is introduced to the energy storage and is subsequently cooled by heat transfer to the ice solution. The air delivered to the room will be in either case, very close to the temperature of the ice solution.

Physically, the design selected was a simple two-pass cross-flow heat exchanger. The ice solution is contained in vertically-mounted tubes arranged in a close-packed staggered arrangement. A sketch of the storage unit showing its outside dimensions and the air inlet and discharge is shown in Fig. 6. Since the unit was designed to be built in 1.22 m (4 ft) square segments, the tube size selection was based on fitting a tube configuration to a segment. The analysis of the tube selection follows:

Figure 7 defines the variables used in expressing the tube spacing for a tube bank. The spacing of the tubes will obviously



	m	ft
A	1.22	4
B	1.83	6
C	7.31	24

Figure 6. Thermal energy storage unit

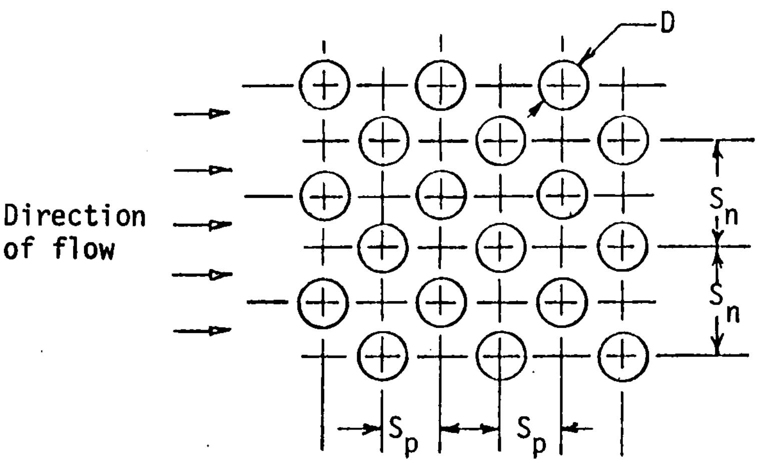


Figure 7. Staggered tube bank
(from reference 15)

affect the number of tubes that can be fit into a given area, but there are limitations to how close the tubes can be packed. Due to pressure drop and heat transfer considerations, the tube spacings S_n/D and S_p/D were chosen as approximately 2.25 and 1.25, respectively [15]. Using these values as approximate standards, several arrangements using standard pipe sizes were chosen and the relative costs and capacities were compared. The capacities are based on six 1.22-m (4-ft) square sections of pipes filled to a height of 1.83 m (6 ft). The results are given in Table 9. On the basis of thermal capacity alone, the 51 mm and 152 mm (2 in. and 6 in.) sizes are superior. Comparing the relative costs, however, reveals that the 152 mm (6 in.) size is the optimum choice. Also, the 6-section arrangement provides the proper capacity as specified previously.

Nominal Pipe dia. mm (in.)		$\frac{S_n}{D}$	$\frac{S_p}{D}$	# of pipes	Capacity W-hrx10^{-5}	(Btux10^{-5})	Relative Cost
51	(2)	2.33	1.17	768	2.8	9.6	1.00
102	(4)	2.44	1.33	168	2.3	8.0	0.65
152	(6)	2.11	1.28	90	2.8	9.7	0.67
203	(8)	2.32	1.16	48	2.6	9.0	0.63

Table 9. Comparison of various pipe sizes for use in the thermal energy storage unit

The total surface area available for heat transfer using a 6-section arrangement of 152 mm (6 in.) pipe is approximately 87 m^2 (940 ft^2).

A thermodynamic heat balance was performed on the storage unit to determine the discharge air temperature, given the inlet conditions. The analysis follows:

Setting the heat transfer from the air equal to the heat transfer to the pipes yields:

$$[\dot{m}\ C_p\ \Delta T]_{air} = UA\ \Delta T_m \tag{6}$$

where $\dot{m}$ is the air mass flow rate, C_p is the specific heat, and ΔT is the temperature rise. U is the overall heat transfer coefficient for the pipe and A is the surface area. ΔT_m is the log-mean temperature difference and is expressed by:

$$\Delta T_m = \frac{T_2 - T_1}{\log\left[(T_{ice} - T_1)/(T_{ice} - T_2)\right]} \tag{7}$$

where T_1 and T_2 are the entering and leaving air temperatures and T_{ice} is the ice solution temperature. Because the pipe has a high thermal conductivity and an ice-water solution has a high convection coefficient (compared to air), the outside convection coefficient, h, can be substituted for U. The convective coefficient can be calculated using (from reference 15):

$$h = \frac{k_f}{D}\ C\ Re_{max}^{n}\ Pr^{1/3} \tag{8}$$

where C and n are empirical constants tabulated for values of the spacing ratios S_n/D and S_p/D. The air properties are evaluated at the film temperature (assume approximately 0°C) as follows:

$$k_f = \text{thermal conductivity} = 0.024\ \frac{W}{m\cdot C}\ (0.0139\ \frac{Btu}{hr\cdot ft\cdot F}\ )$$

$$v_f = \text{kinematic viscosity} = 13.2\text{x}10^{-6}\ \frac{m^2}{sec}\ (142\text{x}10^{-6}\ \frac{ft^2}{sec}\ )$$

$$Pr = \text{Prandtl number} = 0.714$$

The Reynolds number is calculated using the maximum flow velocity between the tubes. The air flow rate for the system can be estimated assuming that the refrigeration capacity specified will be approximately equal to the maximum expected cooling load (see Table 8) of 3910 W. Using the "rule of thumb" introduced in Section 2.9 an air flow of approximately 1900 m^3/hr (1110 cfm) would be necessary. Since the frontal area of the air intake to the storage unit is approximately 1.0 m^2 (10.9 ft^2) the average flow velocity is:

$$u_\infty = \frac{1900}{1.0}\ (\ \frac{1}{3600}\ ) = 0.53\ \text{m/sec (104 fpm)}$$

The maximum flow velocity and the Reynolds number are thus:

$$u_{max} = u_\infty\ (\frac{S_n}{S_n - D}) = 0.53\ (\frac{2.11}{2.11-1}) = 1.01\ \text{m/sec (200 fpm)}$$

$$Re_{max} = \frac{u_{max} D}{u_f} = \frac{(1.01)(0.152)}{13.2 \times 10^{-6}} = 1.16 \times 10^4$$

The constants C and n are evaluated as 0.576 and 0.556, respectively. The convective heat transfer coefficient is, thus, from Eq. 8:

$$h = \left(\frac{0.024}{0.152}\right)(0.576)(1.16 \times 10^4)^{0.556}(0.714)^{1/3}$$

$$= 14.8 \frac{W}{m^2 \cdot C} \quad \left(2.6 \frac{Btu}{ft^2 \cdot hr \cdot F}\right)$$

The unknown quantity in the heat balance equation (Eq. 6) is the discharge air temperature T_2. Assuming the ice solution to be at a temperature of -1 C (30 F) and the entering air to be at -5 C (23 F), results in a solution T_2 as follows:

$$\dot{m}\ Cp\ (T_2 - T_1) = hA \frac{T_2 - T_1}{\log\left[(T_{ice} - T_1)/(T_{ice} - T_2)\right]} \qquad (6\ \&\ 7)$$

$$T_2 = T_{ice} - \frac{T_{ice} - T_1}{10^{\frac{hA}{\dot{m}\ Cp}}} = -1 - \frac{-1 + 5}{10^{\frac{(14.8)(87)}{(0.68)(1005)}}}$$

$$= -1.05\ C\ (29.9°F)$$

Note that the air discharge temperature is only 0.05 C (0.1 F) different than the ice temperature. This result verifies the temperature control capabilities of the storage unit. The heat exchanger effectiveness of the unit is:

$$\varepsilon = \frac{T_2 - T_1}{T_{ice} - T_1} = \frac{-1.05 + 5}{-1 + 5} = 0.986 \qquad (9)$$

which is very high but not unreasonable considering the large surface area involved.

To check the ability of an open-ended vertical pipe to withstand repeated freezing and thawing cycles a full-size sample was filled with a 4% aqueous solution of ethylene glycol and repeatedly frozen and thawed in a walk-in freezer. No damage to the pipe was noted after 5 cycles. The expansion of the ice appeared to be upward through the pipe rather than outward against it.

The construction of the thermal energy storage unit will be performed on site at the facility.

3.4 COOLING SYSTEM COMPONENTS

A schematic diagram of the final refrigeration system design is shown in Fig. 8 and a listing of the major components may be found in Table 10. The selection of the various components is discussed below.

3.4.1 Wind Generating Unit

The wind generator selected is a Swiss-made modified Electro WVG 50 unit, rated at 10 kilowatts at a wind speed of 10.7 m/sec (24 mph). The electrical output is 3-phase alternating current rectified to direct current. An approximate output power versus wind speed characteristic for the wind generator is shown in Fig. 9.

The wind generator will be mounted on a modified Rohn 55 G tower at a height of 27.4 m (90 ft).

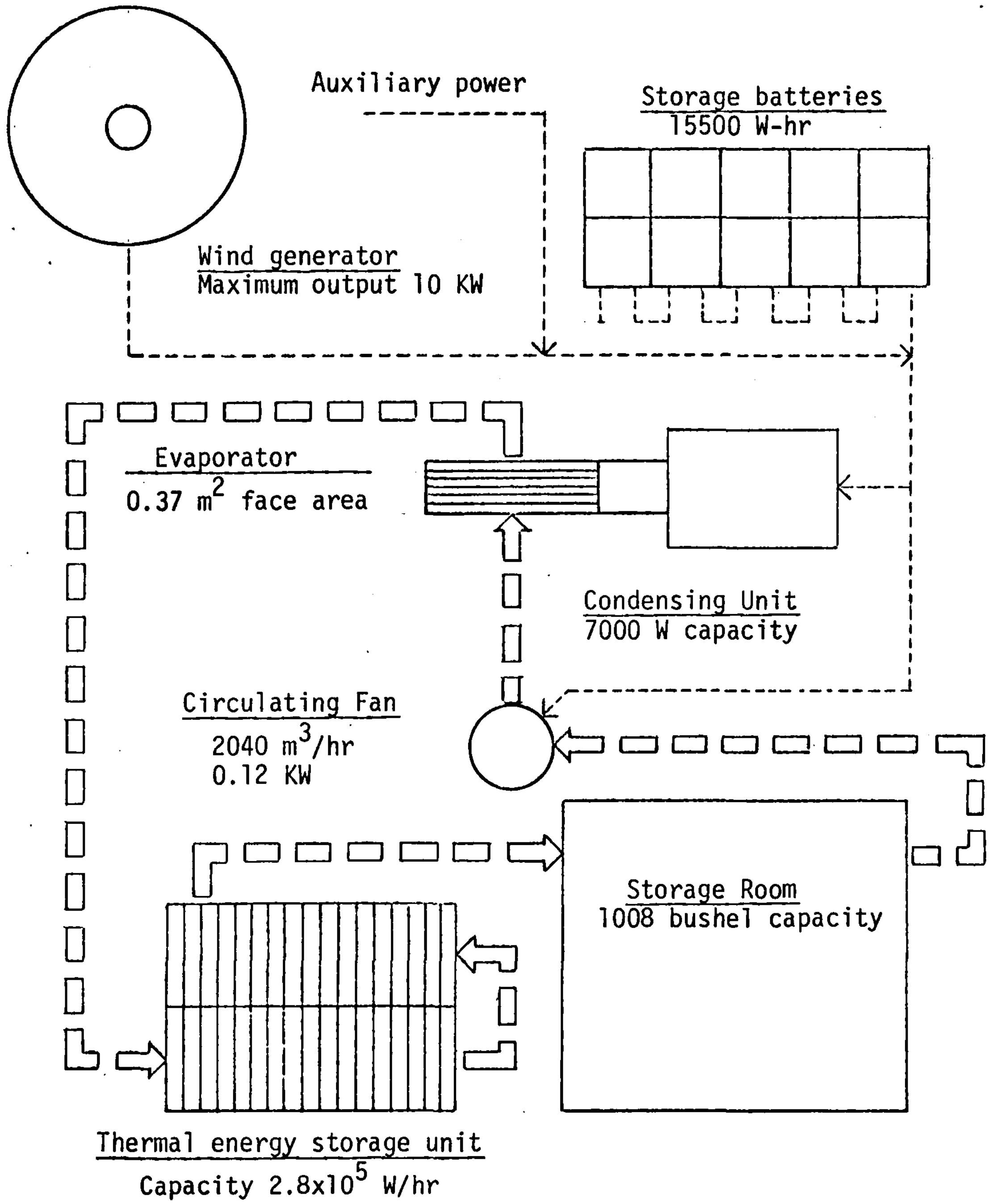


Figure 8. Cooling system schematic.

Component	Manufacturer and Model
Wind Generator	Electro WVG50
	10 kW @ 10.7 m/sec (24 mph)
Tower	Rohn 55G (modified)
	27.4 m (90 ft)
Condensing Unit	Lehigh 11302 (d.c. motor)
	2.24 kW (3 HP)
Evaporator	Aerofin DP DX 144 3NTL
Circulating Fan	Aerovent 315 BIA DWDI
	0.12 kW @ 124 rad/sec
	(0.16 HP @ 1180 RPM)

Table 10. List of major cooling system components

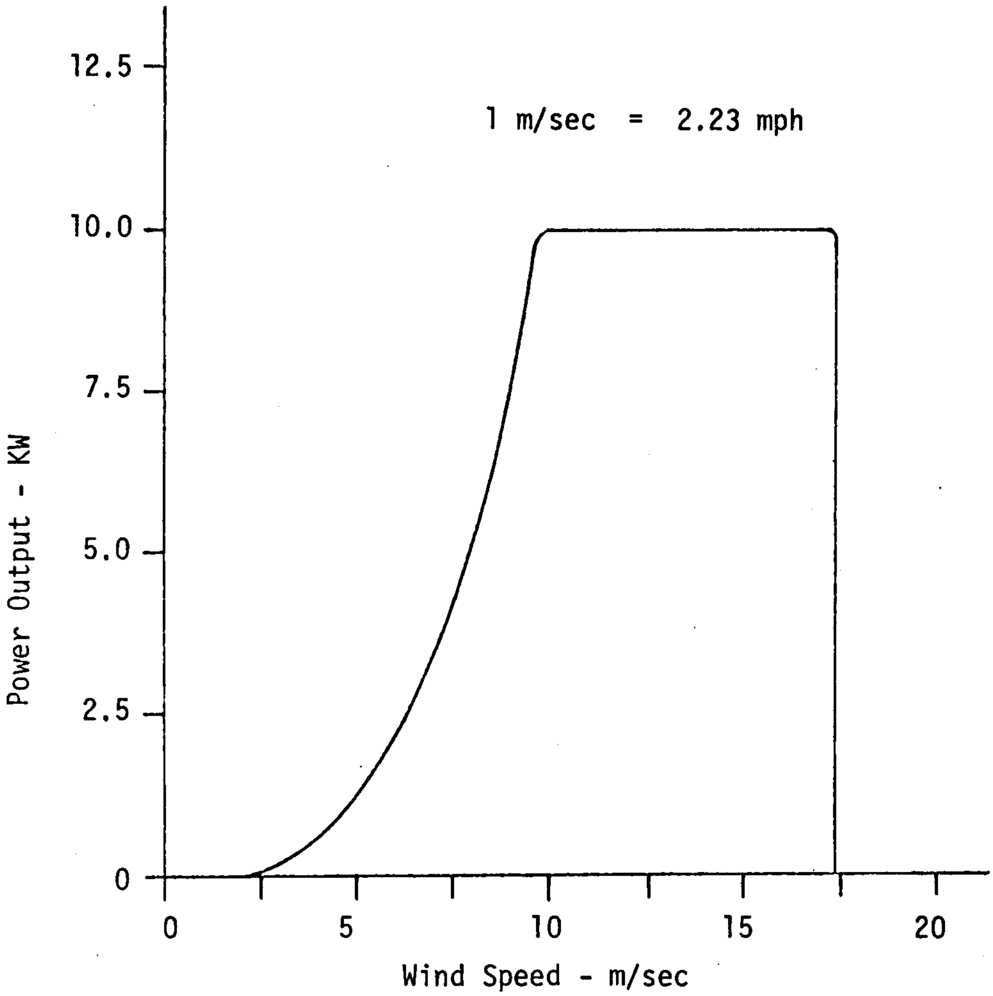


Figure 9. Wind generator characteristic for modified Electro WVG50.

3.4.2 Condensing Unit

The condensing unit (consists of compressor and motor, condenser and condenser fan, receiver, pressure controls, oil trap, and frame) selected is a Lehigh 11302 2.24 kW (3 HP) heavy-duty, air-cooled, open-type unit. The capacity was based on the maximum expected cooling load (initial cooling period) of 3910 W (13340 Btu/hr) at an outdoor temperature of 25 C (77 F). An availability of 13 hours per day or 54 per cent was assumed to allow for coil defrosting (based on manufacturer's recommendation). The resulting required capacity is thus 7240 W (24,700 Btu/hr). The unit chosen is capable of delivering the required capacity.

An open-type unit was specified to facilitate the conversion of the unit to a direct current drive.

3.4.3 Evaporator

A direct-expansion evaporator was selected (Aerofin DP DX 144) with a coil face area of approximately 0.37 m^2 (4 ft^2). A minimum fin spacing of 2.5 mm (0.1 in.) was specified to prevent frost blockage. The manufacturer's specification for air pressure drop at the design velocity of 90 m/sec (300 fps) was approximately 25 Pa (0.1 in. H_2O). Defrosting is accomplished using room air.

3.4.4 Circulating Fan

The general requirements for the air distribution system were discussed earlier, and it was pointed out that typical design procedure specifies a minimum air circulation rate of 1700 m^3/hr

(1000 cfm) per 3520 W (12000 Btu/hr) of refrigeration capacity. The refrigeration unit chosen has a rated capacity of approximately 7320 W (25000 Btu/hr) but because of the operational characteristics of the thermal energy storage only about one-half of this is available for room cooling. The circulating rate chosen was approximately 2000 m^3/hr (1200 cfm) which during the maximum expected cooling period would result in a "split" of approximately 5 C (9 F), which is acceptable.

As was discussed earlier with regard to the thermal energy storage unit, the circulating fan will run continuously, charging and discharging the energy storage unit.

For complete specification of the fan, the pressure required must be determined. This is obtained through a calculation of the pressure drop for the air distribution system. The analysis follows:

Because the distribution system is relatively short the friction drop in the straight ducting will be negligible compared to the losses incurred in the various duct fittings. The pressure losses that are encountered in some common duct fittings are summarized in Table 11.

The design velocity for delivery and return ducting was 2.5 m/sec (500 fpm). Discharge and return air openings were assumed to have discharge velocities of 0.75 m/sec (148 fpm).

The pressure drop for the thermal energy storage can be estimated using the method from reference 16 for tube banks. The loss is

Component	Pressure Drop in Velocity Heads
90° elbow	1.15
45° branch	0.25
Entry from chamber	0.85
Abrupt enlargement	1.00
Sloped enlargement	0.25
Abrupt entrance	1.50
Sloped transition	0.25

Table 11. Friction losses for various duct components (from reference 14).

given by

$$\Delta P = N f \rho \frac{u_{max}^2}{2} \tag{9}$$

where N is the number of tube rows, ρ is the density and f is an empirical friction factor dependent on the Reynolds number and the tube configuration. For the given arrangement, f was determined to be approximately 0.33. Substitution yields

$$\Delta P = (72)(0.33)(1.3)\frac{1.01^2}{2} = 15.8 \text{ Pa } (0.06 \text{ in. } H_2O)$$

Since the air is also turned 180 degrees as it flows through the tank, this estimate was increased to 25 Pa (0.1 in. H_2O).

Using the specifications mentioned for the duct design the total pressure drop for the system can be estimated. The calculations are summarized in Table 12.

Component	Pressure Loss Pa	In. H_2O
Thermal energy storage	25	0.10
Evaporator	25	0.10
Delivery duct elbows (3)	15	0.06
Delivery discharge	-	
Return duct elbows (4)	20	0.08
Return inlet	-	
Fan inlet	10	0.04
Fan discharge	10	0.04
Miscellaneous	20	0.08
TOTAL	125	0.5

Table 12. Pressure drop calculation for air distribution system

The fan selected is an Aerovent 315 backward-inclined airfoil-blade double-width double-inlet centrifugal blower. It will deliver 2000 m^3/hr at 125 Pa (1200 cfm, 0.5 in. H_2O) at 124 rad/sec (1180 RPM) with a power consumption of approximately 0.12 kW (0.16 HP). Should the system pressure require operation at a higher or lower speed to achieve the desired flow, the drive can be altered to accommodate this.

As with the condensing unit, the fan will be driven by a d.c. motor.

3.4.5 Electric Storage Batteries

The capacity of the storage batteries was based on the energy required to operate the entire system (condensing unit and fan) for a period of 12 hours. As was noted before, the maximum expected steady state cooling load (see Table 7) is approximately 2530 W (8630 Btu/hr). Assuming a coefficient of performance of 2.0, the resulting electrical requirement would be 1270 W. Assuming a fan requirement (including drive efficiency) of 130 W, the total system power requirements sums to 1400 W. The storage required for a 12-hour reserve is 16800 W-hr.

The voltage and charge rate characteristics of the batteries are also important. The voltage desired to be compatable with the wind generator is approximately 120 to 130 volts and a charge rating of up to 100 amps (or provision to "dump" the over-current) must be available.

The batteries selected are a Nife C3600 bank of 92 type KB12 Nicad cells. The cell bank capacity is 120 amp-hr at 130 volts (15600 W-hr) which approximately meets the above specification. The watt-hour efficiency of the cells is approximately 0.70.

The total energy storage rating of the entire cooling system (thermal plus electrical is thus approximately 3.16×10^5 W-hr (1.08×10^6 Btu) with 90 per cent being thermal and 10 per cent being electrical.

3.4.6 System Instrumentation

The cooling system instrumentation will be used to gather data of two types: (1) performance data useful in monitoring system operation, and (2) research data useful in evaluating the system design. For this part of the investigation the measurement of the performance data received first priority.

The two most important performance parameters are room temperature and humidity. It is imperative that these variables be carefully monitored at several spots within the facility to insure proper system operation. It is, of course, desirable to maintain an even distribution of temperature and humidity throughout the room.

To monitor the performance of the condensing unit the suction and discharge pressures of the compressor will be measured along with the accompanying temperatures. The power consumption of the compressor motor will also be monitored.

The circulating air fan motor power consumption will be measured along with the air pressure rise to monitor the fan performance.

Measurement of the above parameters will provide the information necessary for an evaluation of the system performance.

3.4.7 Controls

For a typical refrigerated fruit storage the input parameters to the control system are temperature and humidity. Sensors measure these parameters, compare the values to predetermined quantities, and the refrigeration system is operated accordingly.

The operation of the control system for the wind generator-powered cooling system, however, will be different. Because the thermal energy storage unit provides automatic temperature control, the operation of the condensing unit will be controlled by the battery charge state. Higher voltages indicating a full battery charge allow the condensing unit to operate at full capacity. A lower voltage will prevent the condensing unit from starting and allow the batteries to charge. An extremely low battery voltage or a rising room temperature would serve as signals to use auxiliary power.

3.5 SYSTEM OPERATION

To more clearly explain the operational features of the cooling system, it is convenient to postulate the situations that will occur during the storage period and explain the system operation for each.

During the period preceding the loading of the apples into the building the primary objective is to fully charge the energy storages, i.e., freeze the entire ice solution and fully charge the batteries. If, for reasons of low energy availability, this proves too much for the system to accomplish, then auxiliary power will be used.

When the apples are first loaded into the facility, the cooling load is very high. The ice solution will be depleted but the air discharge temperature will remain essentially constant. The cooling load will be reduced as the apple temperature is lowered.

During steady state operation the key component is the thermal storage. The condensing unit will operate intermittently, depending on the electrical energy availability. The cooling load will vary with the ambient conditions. The cooling system as a whole is designed to continuously deliver 2000 m^3/hr (1200 cfm) of -1 C (30 F) air to the storage room. The thermal storage thus in effect acts as a thermal capacitor, charging and discharging depending on the operational situation.

By comparing the available wind data readings with the cooling load calculations it is obvious that the first few weeks of the storage period are the most critical. The highest energy requirements are encountered along with, unfortunately, low expected wind energy because of the season.

4. CONCLUSIONS AND RECOMMENDATIONS

The basic system design for a wind generator-powered cooling system has been presented. The most significant component is the thermal energy storage unit, which functions as a cold storage capacitive unit, storing excess cooling capacity when available and delivering it when required. Simplicity of design is the outstanding overall feature.

Installation and operation of the cooling system require the performance of several additional tasks:

1. Design detailing such as locating and sizing ductwork, locating component placements, and detailing ductwork connections.
2. Design an automatic humidification system to maintain 85 to 90 per cent relative humidity with low power consumption. Simplicity is again a desired feature.
3. Design a control system to implement the functions specified previously and in addition insure that sufficient evaporator defrost time is provided and that the thermal energy storage unit is not completely depleted or subcooled.
4. Devise a maintenance and inspection program designed to monitor the system operation, evaluate the performance, and prevent mishaps.

Completion of these tasks will, it is anticipated, result in an efficient, low-maintenance system which will be completely or almost completely powered by wind energy.

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DESIGN OF A WIND-POWERED COOLING
SYSTEM FOR AN APPLE STORAGE FACILITY

by

John Clisby Blanton

(ABSTRACT)

A study was undertaken to determine the problems involved in applying wind-produced electric energy toward the needs of an apple storage facility, and to design an appropriate energy system. Fruit requirements, wind generator characteristics, and energy storage are discussed.

Detail design of a cooling system was performed, including design calculations and equipment specification. The expecting cooling loads for the building were calculated and a vapor-compression refrigeration condensing unit was selected. An energy storage device utilizing the latent heat of fusion of ice was designed and calculations were performed to determine its energy capacity and heat transfer characteristics. The requirements of the air distribution system are discussed, and the pressure drop calculations are shown in connection with the circulating fan selection. The overall system operation is discussed, including control system requirements.