

Design of Fog Sprays for Produce Storage

By Rene Guillou*

I. Estimating quantity of spray

Humidity in a produce storage depends on a balance of gains and losses of moisture. The air gains moisture by evaporation from the produce, from humidifying sprays if these are used, and rarely by evaporation from melting ice or from refrigerated spray systems. The air loses moisture by absorption in containers and packing materials, and usually by condensation on cooling surfaces.

Relatively high humidity is usually desired as a means of reducing evaporation from produce. Drying of stems is a major form of deterioration in grapes held for several months in cold storage. Nectarines may be visibly shriveled after a week or two under refrigeration. Shrinkage from moisture loss allows nectarines and peaches to become loose in their containers, exposing them to damage from transit vibrations and impacts. Shriveling is a serious problem in the storage of some varieties of apples.

In many products there is more decay at very high humidities. The ideal condition is that where the combined losses from shrivel and decay are at a minimum. Unfortunately, this ideal condition is not well defined. A special problem in grape storage is the increased absorption of sulfur dioxide fumigant by containers as humidity increases. This may presumably be offset by adding more sulfur dioxide but no formula is available for determining how much more is needed. Apparently an engineer's responsibility is to design a system capable of holding humidity up to the level and within the limits that his client wishes, with provision for adjustment downward as desired.

Provisions for controlling humidity must take into account the moisture exchange between the air, the produce, and the containers and packing materials. Figure 1 shows the gross weight gain or loss by grapes packed in wooden lugs, as measured at Davis in controlled humidity chambers. Apparently dry packed boxes will lose moisture from the start at relative humidities below about 85 per cent. At higher humidities the packed boxes gain moisture until the wood is partly saturated and then begin to lose.

Representative heat and moisture diagrams in a room cooled with unsalted ice are shown in figure 2. This is a portion of an ordinary psychrometric chart, enlarged for convenience. Each diagram represents the history of a pound of dry air as it circulates through a room and an ice bunker. To illustrate the use of diagrams of this kind for arriving at engineering designs, consider the one marked "Typical Dry System". This is based on these assumed conditions:

1. Air circulation 3,000 cfm per ton of refrigeration. Since 2,500 cfm of air per ton of refrigeration is equal to 1 lb. of air per Btu, 3,000 cfm/ton equals 1.2 lb/Btu or .8 Btu/lb. Therefore the enthalpy difference between return and blast on the chart must be .8 Btu.
2. By-pass factor .50, meaning that half of the return air passes through the ice bunker unchanged while the balance comes to equilibrium with the melting ice surface, at 32°F and 100 RH. This is an over-

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simplification of what happens but it is in reasonable agreement with final results and is a convenient basis for design. On the chart the conditions of the air blast will therefore be at the midpoint of a straight line connecting conditions at the return with those at the ice surface.

3. Moisture gained by the air stream from packed boxes must be equal to that lost by condensation on the ice, otherwise the humidity will rise or fall until the system is in balance. The slope of the line from blast to return on the chart must therefore correspond with estimated moisture gain from the packed boxes at the average relative humidity, using a conversion of 1 grain/Btu equals 42 lb/ton-day of refrigeration.
4. A line to satisfy these three conditions may be found by cut-and-try. In this case, assuming grapes on the tenth day in storage, moisture gained by the air stream from the boxes is estimated from figure 1 as about 8 lb/ton-day of refrigeration or .2 grain/Btu, at an average relative humidity of 84.

Return air relative humidity in this system is 79, which is commonly considered to be too dry. The diagram marked "Ideal Dry System" shows how the return relative humidity could be raised to 90 by increasing the air flow and so decreasing the enthalpy difference between blast and return. About four times the air flow is required and, to maintain a by-pass factor of .50, the ice-grate area must also be four times as large. In this case the line from blast to return slopes downward, indicating the estimated moisture lost by the air stream to the packed boxes at an average relative humidity of 92.5.

The upper diagram shows how the typical system may be improved by installation of fog sprays. The dotted lines indicate that enthalpies of blast and return air are not changed. A pound of dry air picks up the same amount of heat in the room and loses it again in the ice bunker. As the air passes through the room it alternately gives up moisture to the packed boxes and is rehumidified by fog spray. Evaporation of the spray raises the humidity and cools the air appreciably but does not change the enthalpy. Under the assumed conditions, spray must be introduced at two points in the room if the relative humidity to be held between 90 and 100.

Table 1 compares the engineering features of the three systems. The spray system apparently offers the best value in return for its cost. Diagrams like these can be drawn to represent the history of a pound of dry air in any proposed system, showing just what can be accomplished and the design that will be required to do it.

Heat and moisture diagrams in a room cooled by dry coils are shown in figure 3. A typical liberally designed system might have a 30F air blast and a 33F return when operating under moderate load, with the cooling surfaces at 27F. As with an iced system, the air gains moisture and heat in the room and loses them to the cooling surfaces. Relative humidity in the return air in this system is 79 per cent.

Dry cooling surfaces can not add moisture to the air and are, therefore, not able to maintain any humidity that results in moisture gain by the packed boxes. The best that an ideal dry-coil system can do is to maintain a humidity at which moisture lost by the fruit is equal to that gained by the boxes. From figure 1 this situation exists for grapes in wooden lugs on the tenth day of storage at an average relative humidity of about 87 per cent. The ideal system shown in figure 3 operates with cooling surface temperatures that entirely eliminate frosting or condensation, yet the relative humidity of the return air is about 83 per cent.

Improvement of the typical system by addition of fog spray is shown by the upper diagram in figure 3. The 30F air blast is retained and each pound of dry air that is circulated still gains .8 Btu in the room and loses it to the cooling surfaces. However, as the air loses moisture to the packed boxes it is rehumidified and also somewhat cooled by fog sprays. Table 2 compares the three "dry-coil" systems.

Similar diagrams may be drawn for a brine spray system. Relative humidity at a brine surface is somewhat less than 100 per cent, depending on the brine concentration. For given return air conditions there is somewhat more condensation in brine than on dry coils at the same temperature. This is easily offset in a fog spray system by use of more spray.

Referring again to figure 3, the condensation on the cooling surfaces per ton of refrigeration load depends on the slope of the line from air return to air blast conditions. In a fog spray system the slope of this line is not much affected by practicable cooling surface temperatures. This means that when fog spray is used the cooling surface area and temperature may be chosen for economy in the refrigeration system, without regard for effects on humidity. The advantages of performance and economy in produce refrigeration can hardly be over-emphasized.

The effect of humidification on the refrigeration load should be mentioned. Heat is absorbed by net evaporation of liquid water from the packed boxes in a dry system. If this water condenses as a liquid, on melting ice or in brine spray, it has no effect on the refrigeration load. If it freezes on the cooling surfaces it adds slightly to the refrigeration load - less than one per cent in the example in table 2. When fog spray is absorbed or condensed as water, some refrigeration is used to cool the water - a little over one per cent of the initial load in the examples above. Fog spray that freezes on the cooling surfaces is cooled and also releases its heat of fusion, adding about 4 per cent to the refrigeration load in the example in table 2.

Steam is sometimes suggested for humidifying produce storages. Any steam injected into a produce storage is removed eventually as either water or ice, after releasing its heat of vaporization and frequently its heat of fusion also. Substitution of steam for fog spray in the example in table 2 would increase the refrigeration load by over 40 per cent.

The importance of humidification while produce is cooling is sometimes questioned. Figure 5 shows vapor pressures for the cooling and initial storage periods in a typical operation. The single-hatched area represents the vapor

pressure difference between produce and humidified air; the double-hatched area represents the additional difference if humidification is not used. Conditions in any specific operation will be different from these, but the air is usually warmer when the produce is warm, and without humidification the relative humidity is lower also. Based on vapor pressure differences, the benefit from humidification is greatest at the start of the cooling operation and declines steadily thereafter.

II. Nozzle types and controls

Humidifying sprays should ordinarily evaporate before they settle to the floor or impinge on surfaces where water may cause damage or become a nuisance. Finely atomized spray settles slowly if at all and evaporates within a few seconds unless the air is close to saturation. To avoid producing a local region of saturation the discharge from any one nozzle must usually be limited to around a gallon per hour.

Satisfactory atomization from fixed nozzles supplied with water alone is possible, but ordinary water pressure is usually not sufficient and nozzles must be so fine as to threaten constant trouble from clogging. Self-contained, electric-motor-operated atomizers are available. These are designed primarily for humidification of living or working spaces. Their costs are high and their capacities are low for use in produce storages. Nozzles which use compressed air to atomize the water are commonly used for produce storage humidification and are probably the best type for this purpose.

Air atomizing nozzles may mix the air and water either inside the nozzles, or after discharge. Fineness of atomization depends on both the air pressure and on the ratio of air flow to water flow. Air pressure of 40 psi and 1 cfm of air for each gallon per hour of water should give satisfactory atomization with either type of nozzle. Discharges of air and water under different pressures are given in nozzle catalogs.

A study of catalog figures for internal-mixing nozzles shows that the air and the water are working against each other. Any change in the pressure or flow of either component results in an opposite change in the other. Close regulation of both air and water is therefore necessary to maintain the desired water flow and sufficient air to atomize it. Also water pressure usually must be at least 30 psi at the nozzles in order to maintain a flow when air pressure is 40 psi, and some water sources can not be depended on to supply this pressure at all times.

With external mixing, on the other hand, water may be syphoned from a tank or supplied through a needle valve at low pressure. Flows of water and air are practically independent and air pressure needs only to be kept sufficiently high for good atomization. Air may be supplied to these nozzles directly from a compressor. Elimination of pressure reducers is a substantial saving in expense and complication in small installations. Loss of water-head from friction in lines may be important in relation to the low heads at external mixing nozzles. If there are long water pipes between nozzles they must be large enough to limit the loss of head to a few inches.

Apparently the following conditions are favorable to the use of internal mixing nozzles:

Available water pressure never less than 30 psi at nozzles.
Air taken from central supply through a pressure regulator.
Long runs of water pipe between nozzles.
Automatic on-off control.

Conditions more favorable to external mixing nozzles are:

Available water pressure sometimes less than 30 psi at nozzles.
Air compressor used only to operate the spray system.
Short runs of water pipe between nozzles.
Manual control of water by needle valve.

Nozzles of the two types have about the same size water orifices for equal water flow and are therefore about equally subject to stoppage by solids in the water. The air orifice is considerably larger in the internal mixing type but stoppage of an air orifice in either type of nozzle is rare.

Nozzles are available with built-in fine-mesh strainers for both air and water, providing maximum protection against stoppage. A single strainer in the supply line is cheaper if there are many nozzles, and has been quite satisfactory. Copper tubing with flare fittings is convenient for air and water lines and insures against future trouble with nozzle stoppage by rust flakes. Electric soil-heating cable may be used when necessary to keep the water lines from freezing.¹/₂

Experience to date does not justify any positive conclusions about controls. Advantages and disadvantages of automatic control may be listed as follows:

Operators not required to watch conditions and make adjustments; no troubles from neglect or misunderstanding so long as system is in order.

Control usually senses conditions in only one place, takes no account of possible excess moisture somewhere else.

Sensing elements require protection from sulfur in grape storages.

Expert attention required for maintenance and repair.

First cost is considerable, especially in small installations.

Advantages and disadvantages of manual control are:

Operators must watch conditions and make adjustments; neglect or misunderstanding can cause trouble.

Operator can correct for excess moisture appearing anywhere.

No components to protect from sulfur.

Maintenance and repair are simple.

First cost is low.

Most existing installations are automatic, though the few that are manually controlled are reported by the operators to be satisfactory.

Convenient constants

1 grain moisture per Btu equals:

- 42 pounds per day per ton of refrigeration or
per ton of ice melted per day
- 5 gallons per day per ton of refrigeration or
per ton of ice melted per day
- .21 gallons per hour per ton of refrigeration or
per ton of ice melted per day

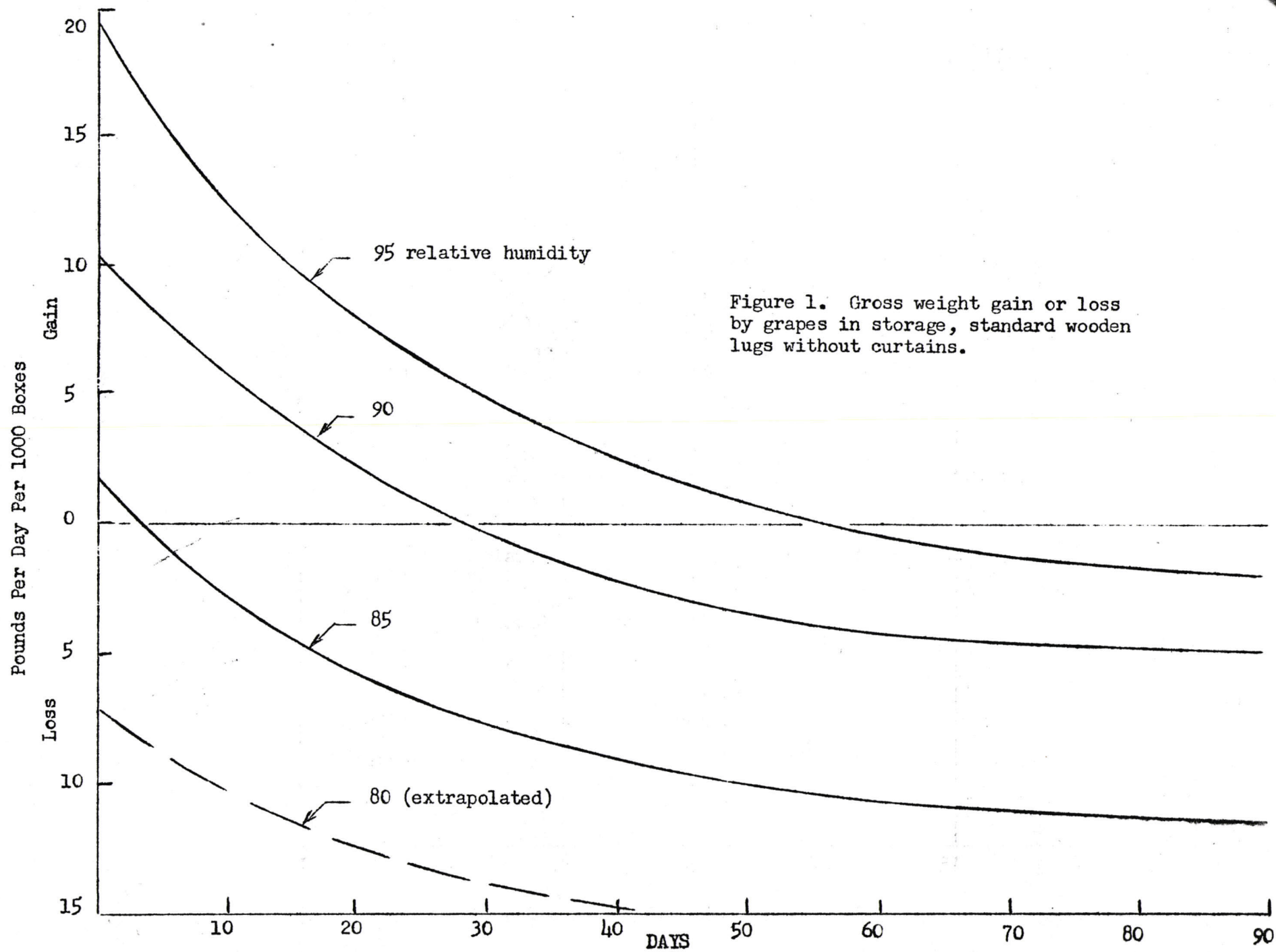
1 gallon water per hour equals 200 pounds per day.

Table 1. Comparison of ice-cooled systems

<u>System</u>	<u>Ideal Dry</u>	<u>Typical Dry</u>	<u>Fog Spray</u>
Blast temperature, F	33	35	34.6
Return temperature, F	34	38	36.6
Blast relative humidity, pct	95	89	95
Return relative humidity, pct	90	79	90
Air circulated, cfm/ton refriger.	12,000	3,000	3,000
Ice grate area, sq. ft/ton refriger.	120	30	30
Water balance, gal/ton ice melted			
Packed boxes	Gain 5	Lose 1	Gain 5
Melting ice	Evap 5	Cond 1	Cond 3
Fog spray	-	-	8

Table 2. Comparison of dry-coil systems

<u>System</u>	<u>Ideal Dry</u>	<u>Common Dry</u>	<u>Fog Spray</u>
Blast temperature, F	30	30	30
Return temperature, F	32	33	32.4
Coil temperature, F	28	27	27.6
Blast relative humidity, pct	91	88	95
Return relative humidity, pct	84	79	90
Air circulated, cfm/ton refriger.	4,500	3,000	3,000
By-pass factor	.50	.50	.50
Water balance, gal/ton-day refriger.			
Packed boxes	0	Lose 2	Gain 4
Cooling surfaces	0	Cond 2	Cond 9
Spray	-	-	13



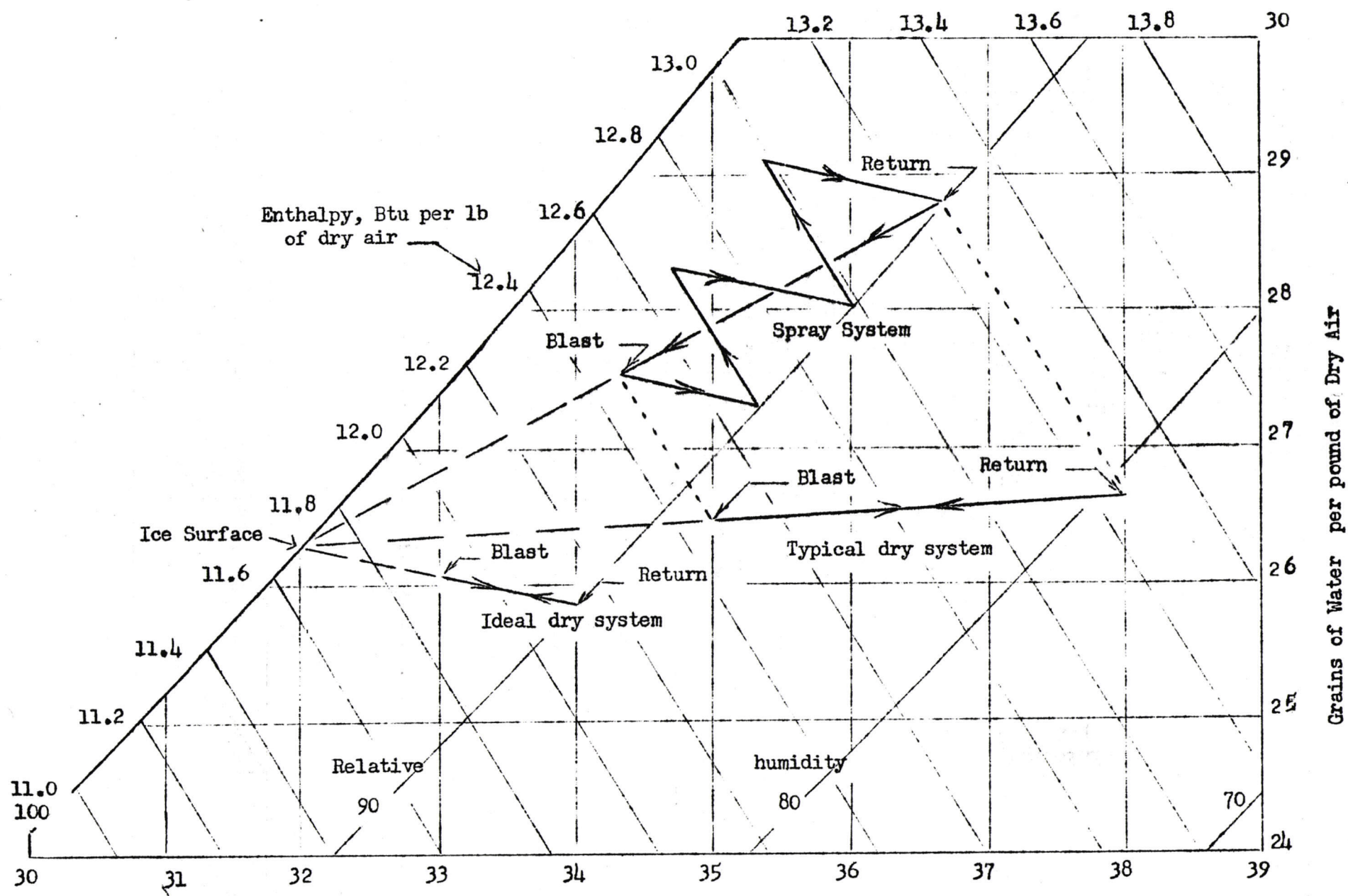


Figure 2. Air Circuits in Ice-Cooled Fruit Storage

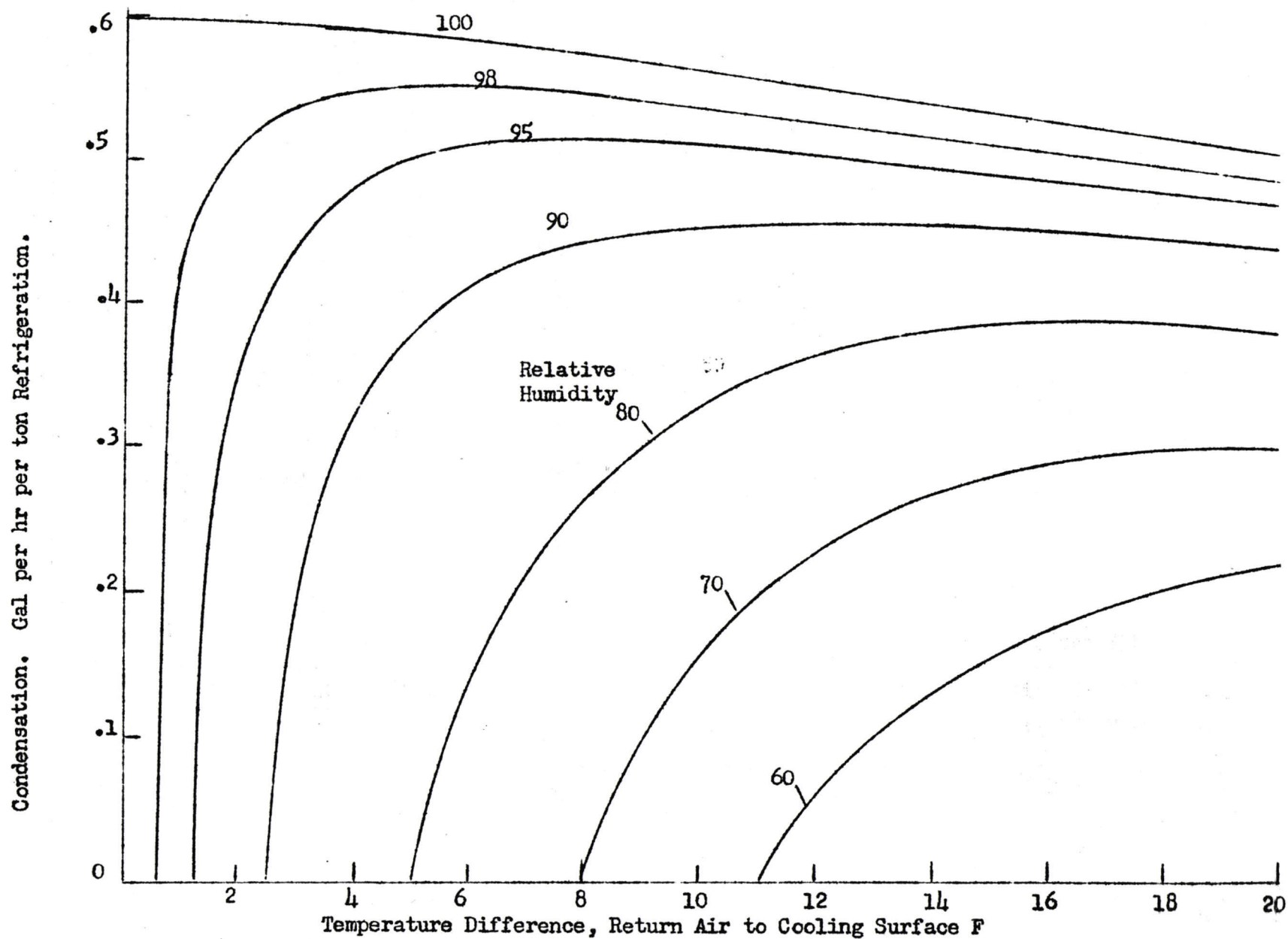


Figure 4. Condensation on Cooling Surface from Air at 35F

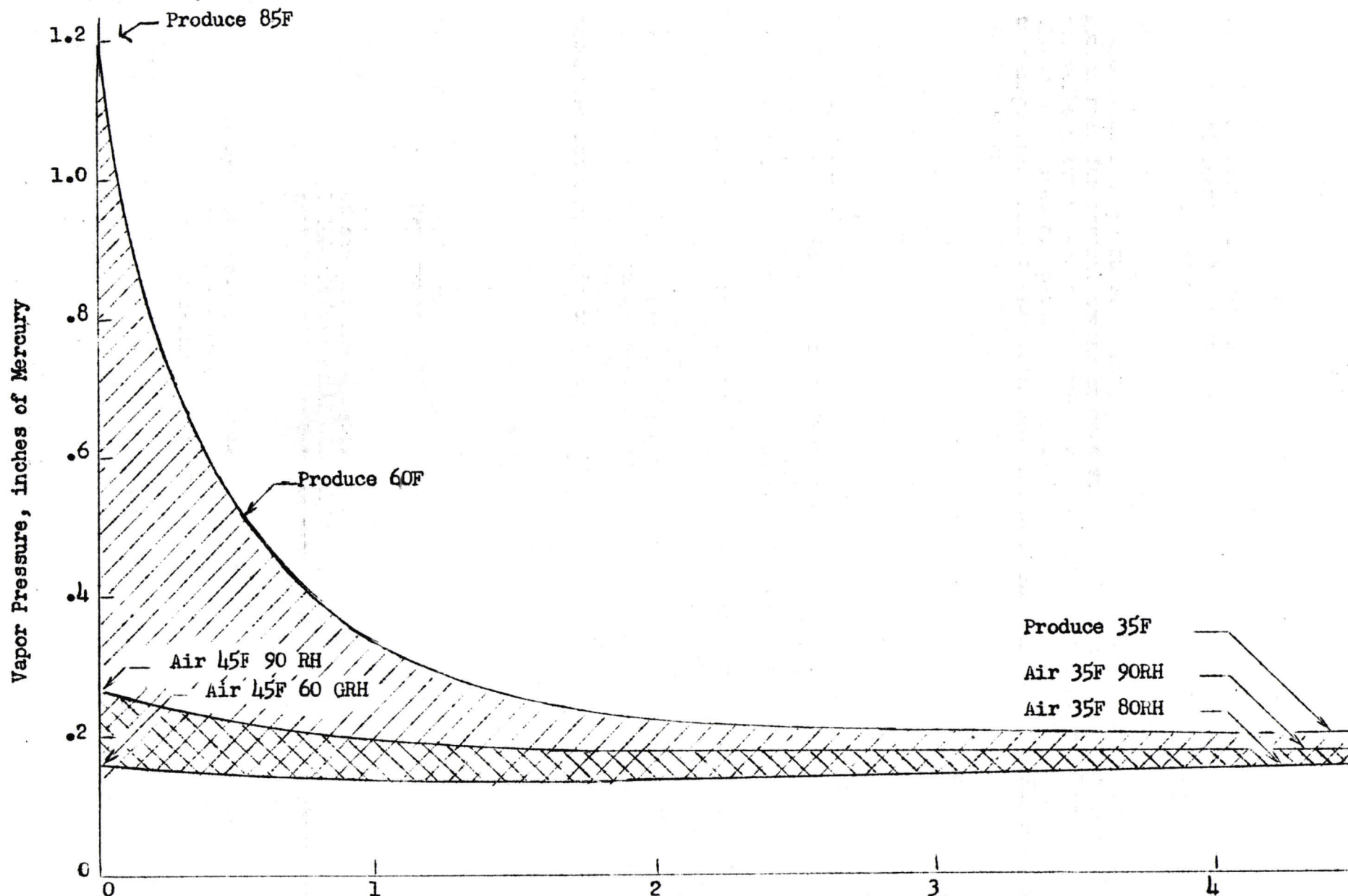


Figure 5 Water Vapor Pressures in Produce Cooling and Storage