Due to the evaporative cooling effect from the moist surface of the washed lemons plus the moisture loss from the fruit itself (81,200 Btuh in the previous Latent Heat Load calculations), the air range may be considered as only 5 F deg and the supply air temperature raised to 53 F:

$$\text{Temp range} = \frac{330{,}635 - 81{,}200}{(60)(0.24)(3440)} = \frac{249{,}435}{49{,}500} = 5 \text{ F deg}$$

58 F room temp − 5 F deg range
= 53 F supply or dew-point temp

The total moisture gain in the storage room from the latent load of 95,660 Btu per hr will be:

$$\frac{(95{,}660)(7000 \text{ gr per lb})}{(60 \text{ min})(1060 \text{ Btu per lb})(3440 \text{ lb per min})}$$

= 3.06 gr per lb of supply air
Moisture content at 53 F supply air dew point = 60 gr per lb.
Moisture gain in room = 3 gr per lb

Moisture content of storage room air = 63 gr per lb

By reference to a psychrometric chart, it is found that 63 grains per lb corresponds to 54.5 F dew-point temperature. The final storage room condition is 58 F dry-bulb and 54.5 F dew point, corresponding to 88 percent rh.

This process may be readily visualized on a psychrometric chart as follows: From 53 F dew point on the saturation line proceed along the dew point line 7 deg to intercept the 60 F dry-bulb temperature. Follow the wet-bulb line from this point to intersect the room condition dry-bulb temperature of 58 F at the point 54.5 F dew point and 88 percent relative humidity.

Refrigeration Load	Btuh
Total heat at 65 F outdoor wet-bulb	= 30.0
Total heat at 53 F apparatus wet-bulb	= 22.0
For outdoor air, apparatus must absorb	8.0
Total heat at 56 F room wet-bulb	= 23.9
Total heat at 53 F apparatus wet-bulb	= 22.0
For return air, apparatus must absorb	1.9
	Btu per min
Outdoor air load = (380 lb per min)(8.0)	= 3040
Return air load = (3,060 lb per min)(1.9)	= 5814
Add for washer spray pump = (10 hp)(42.4)	= 424
Subtotal	= 9278
Add 5% for apparatus and piping losses	= 464
Total Refrigeration Load	= 9742 = 48.7 tons

$$\text{Refrigeration per carload} = \frac{48.7 \text{ tons}}{70 \text{ cars}} = 0.7 \text{ tons}$$

$$\text{Floor area per ton} = \frac{17{,}329 \text{ sq ft}}{48.7 \text{ tons}} = 356 \text{ sq ft}$$

$$\text{Outdoor air} = \frac{45{,}500 \text{ cfm}}{48.7 \text{ tons}} = 935 \text{ cfm per ton}$$

Refrigeration Equipment

The required cooling effect may be obtained by using direct expansion coils in the air washer, or by circulating chilled water from a central cooling system. The refrigerant temperature will generally be between 33 and 40 F. The chilled water requirements and a central system will be about 200 gpm for a 6 F deg range of water temperature, calculated as follows:

$$\frac{(48.7 \text{ tons})(200 \text{ Btu})}{(6 \text{ F deg range})(8\tfrac{1}{3} \text{ lb per gal})} = 194.8 \text{ gpm}$$

The difference between the air and water leaving the washer varies with the washer design, but is usually between 1 and 2 F deg.
53 F washer dew point − 2 F deg = 51 F water leaving
51 F water − 6 F range = 45 F entering water

Select the refrigerant temperature about 10 F deg below the water temperature:
45 F water − 10 F deg = 35 F refrigerant temperature

Air Supply Nozzles (96 nozzles used; required air travel = 35 ft)

$$\text{Outdoor air per nozzle} = \frac{45{,}500 \text{ cfm}}{96 \text{ nozzles}} = 477 \text{ cfm}$$

For an air travel of 35 ft, use about 0.10 in. nozzle pressure, or 1250 fpm nozzle outlet velocity (see Table 6).

$$\text{Area per nozzle} = \frac{477 \text{ cfm}}{(0.94)(1250)} = 0.406 \text{ sq ft}$$

Therefore, make nozzle outlet 8¾ in. diameter = 0.418 sq ft area. Make nozzle about 24 in. long. For a nozzle converging at about a 14 degree included angle, the inlet diameter will be 14¾ in. diameter (Fig. 11).

Example 4: Illustrate calculations for the storage basement of *Example 2* for a combination system using low air quantities and auxiliary humidification.

A combination system uses about half as much air as a conventional system and therefore must operate with a lower apparatus dew point or supply air temperature. This tends to give too low a humidity in the room, so auxiliary humidifying sprays are provided. The auxiliary sprays in the room will produce evaporative cooling which lowers the sensible load while increasing the relative humidity.

Solution:

Air Temperatures

For a 58 F dry-bulb and 88% rh storage condition, the apparatus dew point may be set at 50 F. The supply air quantity may be calculated as though the air range is 14 F instead of 58 F − 50 F or 8 F as indicated.

This may be explained by referring to a psychrometric chart. From 50 F dew point on the saturation line proceed 14 dry-bulb degrees to intersect the 64 F dry-bulb temperature and 56 F wet-bulb line. Follow the 56 F wet-bulb line to intersect the final room condition of 58 F dry-bulb where the dew point is 54.5 F and the relative humidity 88 percent.

Air Quantities

For a combination system using about half the air quantity of the conventional system of *Example 3*, assume a 10 hp fan motor.

Total room sensible load = 279,735 Btu per hr (from *Example 2*)
Add for 10 hp fan motor: (10) (2,545) = 25,450 Btu per hr
Gross sensible heat load = 305,185 Btu per hr

$$\text{Supply air} = \frac{305{,}185}{(60 \text{ min})(14 \text{ F range})(0.24 \text{ sp ht})}$$

= 1510 lb per min
= (1510)(13.23) = 20,000 cfm

Outdoor air = 380 lb per min = 5,040 cfm (from *Example 3*)
Return air = 1510 − 380 = 1,130 lb per min
= 20,000 − 5,040 = 14,960 cfm

$$\text{Supply air per car} = \frac{20{,}000}{70} = 286 \text{ cfm}$$

Apparatus Selection

Supply fan: Required for 20,000 cfm. Assuming the resistance of the apparatus and duct system is 1¾ in. water, the fan motor needed will be 10 hp.

Air washer: Required for 20,000 cfm at 500 fpm face velocity:

$$\text{Face area} = \frac{20{,}000 \text{ cfm}}{500 \text{ fpm}}$$

$$= \frac{20{,}000}{500} = 40 \text{ sq ft face area}$$

Make washer 7 ft wide × 6 ft high above the pan. Assuming an 18 in. deep pan, the air washer will be 7½ ft high × 7 ft wide × 8 ft long. With two spray banks, a circulating pump

about 200 gpm capacity at 65 ft head will be required. A 5 hp pump motor is necessary.

Refrigeration Load

Total heat at 65 F outdoor wet-bulb temp	= 30.0 Btu per lb
Total heat at 50 F apparatus wet-bulb temp	= 20.3 Btu per lb
For outdoor air, apparatus must absorb	9.7 Btu per lb
Total heat at 56 F room wet-bulb temp	= 23.9 Btu per lb
Total heat at 50 F apparatus wet-bulb temp	= 20.3 Btu per lb
For return air, apparatus must absorb	3.6 Btu per lb
	Btu per min
Outdoor air load = (380 lb per min)(9.7)	= 3,686
Return air load = (1,130 lb per min)(3.6)	= 4,068
Add for washer spray pump = (5 hp)(42.4)	= 212
Subtotal	= 7,966
Add 5 percent for apparatus and piping losses	= 398
Total Refrigeration Load	= 8,364 = 41.8 tons

$$\text{Refrigeration per carload} = \frac{41.8 \text{ tons}}{70 \text{ cars}} = 0.6 \text{ tons}$$

$$\text{Floor area per ton refrigeration} = \frac{17{,}328 \text{ sq ft}}{41.8 \text{ tons}} = 415 \text{ sq ft}$$

$$\text{Supply air} = \frac{20{,}000 \text{ cfm}}{41.8 \text{ tons}} = 480 \text{ cfm per ton refrigeration}$$

Refrigeration Equipment

The required cooling effect may be obtained by using direct expansion coils in the air washer, or by circulating chilled water from a central cooling system. In either case, the refrigerant temperature will generally be between 33 and 40 F. The chilled water requirements from a central system will be about 170 gpm for a 6 F deg range of water temperature, calculated as follows:

$$\frac{(41.8 \text{ tons})(200 \text{ Btu})}{(6 \text{ F range})(8\tfrac{1}{3} \text{ lb per gal})} = 167.2 \text{ gpm}$$

Assuming a 2 degree temperature difference between the air and water leaving the air washer:
50 F washer dew point − 2 F deg = 48 F water leaving
48 F water − 6 F deg range = 42 F entering water

Select the refrigerant temperature about 9 F deg below the water temperature:
42 F water − 9 F temperature difference = 33 F refrigerant temperature

Air Supply Nozzles (48 nozzles used; required air travel = 35 ft)

$$\text{Air supply per nozzle} = \frac{20{,}000 \text{ cfm}}{48 \text{ nozzles}} = 417 \text{ cfm}$$

Design nozzles for 0.10 in. nozzle pressure (1250 fpm outlet velocity) for 35 ft throw (see Table 6).

$$\text{cfm} = (0.94)(\text{velocity})(\text{area})$$

$$\text{Nozzle area} = \frac{417}{0.94} \qquad \text{Area} = \frac{417 \text{ cfm}}{(0.94)(1250)} = 0.356 \text{ sq ft}$$

Therefore make nozzle outlet 8 in. diameter = 0.349 sq ft area. Make nozzle about 20 in. long. For a nozzle converging at about a 14 degree included angle, the inlet diameter will be 12⅛ in. diameter.

Auxiliary Humidifying Nozzles

Moisture content at 54.5 F room dew point = 63 gr per lb
Moisture content at 50 F apparatus dew point = 54 gr per lb
Moisture pick-up required in room = 9 gr per lb

$$\text{Total moisture required} = \frac{(1{,}510)(9 \text{ gr})}{7{,}000} = 1.94 \text{ lb per min}$$

= 116.4 lb per hr

$$\text{Moisture pick-up from room latent load} = \frac{95{,}660}{1{,}060}$$

= 90.2 lb per hr

Therefore, moisture required from auxiliary sprays = 116.4 − 90.2 = 26.2 lb per hr

Auxiliary nozzles should be provided for at about 75 percent of the total moisture requirements, as a factor of safety:
(116.4 lb per hr) (0.75) = 87.4 lb per hr
Provide one spray nozzle for each two air nozzles:

48/2 = 24 spray nozzles

$$\frac{87.4 \text{ lb per hr}}{24 \text{ nozzles}} = 3.64 \text{ lb moisture per hr per nozzle}$$

The sizes of nozzles can be selected from manufacturers' catalogs.

TRANSPORTATION

Since the bulk of the California citrus crop is sold in distant markets, the transportation problem and factors affecting quality are of great importance to the citrus growers. All other factors being equal, the maintenance of grade and quality is dependent upon time in transit, the refrigerator car and its cooling equipment and the class of service requested of the carriers.

Least loss in quality occurs when fruit is first precooled to 39 to 40 F and then loaded into pre-iced, fan equipped cars. Further information on the railway transport of refrigerated and frozen foods can be found in Chapter 56.

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