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Title: Energy Conservation Through Evaporator Fan Cycling in a Refrigerated Controlled Atmosphere Apple Storage Facility

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A two year experiment demonstrated electrical energy savings through evaporator fan cycling in part of a full-sized refrigerated controlled atmosphere apple storage facility in Yakima, Washington. Room cooling demand and evaporator fan operation was controlled by the bulk fruit temperature. Refrigerant temperature was regulated by a computer controlled ramping sequence and a weighted average of three thermistors rather than a room thermostat. Evaporator fans remained off 60-65% of the time during periods of fan cycling operation. Seasonal average fan energy savings approached 50-55% when the product remained in storage for a typical length of time. Room environment conditions, product mass loss, and fruit quality were not compromised. Bulk fruit temperature was exceptionally stable as compared to fruit stored in non-fan cycled rooms controlled by traditional means.

Less fan motor heat input in the fruit storage space also results in compressor energy savings. Total electrical energy

savings were projected to an 18 room fruit storage facility implementing computer controlled evaporator fan cycling. Projected electrical energy savings achievable through evaporator fan cycling for the state of Washington were made based on reported apple storage capacity.

Energy Conservation Through Evaporator Fan Cycling in a Refrigerated Controlled Atmosphere Apple Storage Facility

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INTRODUCTION

The apple industry is of significant importance to the Pacific Northwest states of Oregon and Washington. Apple sales added approximately 20 million dollars to Oregon's economy in 1991 (Miles, 1992) and approximately one billion dollars to Washington's economy in 1990 (Daniels, 1992). Washington produced 2.09 million metric tons of both fresh and processed apples in 1991, which accounted for 45% of the United States apple crop, or about 20% of the world apple crop (Stover, 1992). Sixty six percent of Washington's apple production in 1991 was marketed as fresh produce (Stover, 1992). Consumer demand and the ability to store fresh apples for long periods has created the need for specialized storage facilities. Forty nine percent of the total United States apple crop (2.26 million metric tons) was in storage as of November First, 1991 (Stover, 1992). Total storage capacity in Washington state was 2.77 million metric tons as of October 1991 (WA State Department of Agriculture). Sixty five percent of the storage capacity in Washington state is controlled atmosphere (CA), the remainder being common or regular atmosphere (air) storage. Industry wide, apple production and the amount of storage capacity continues to increase.

Refrigerated CA storage facilities are utilized to extend market availability of fresh apples for periods up to 10 months. Modern CA storages carefully maintain temperature just above the freezing point of the product (-1 to 0°C), high relative humidity (90-96%), and an atmosphere low in oxygen (1.5-2.0%), high in carbon dioxide (approximately 1.0%), and the balance nitrogen. Early attempts to prolong the storage period of fresh apples relied on fruit respiration to lower oxygen concentration to desired levels. Oxygen concentration was then controlled by permitting outside air to enter the room (Dalrymple, 1989). Modern CA storage rooms have atmosphere generating devices that remove oxygen from the storage room air much more rapidly than can be done by fruit respiration. Excess carbon dioxide generated from fruit respiration can be scrubbed from the air by means of dry hydrated lime, water, caustic soda, activated carbon, or molecular sieves (Hardenburg et al., 1986).

Storage of apples for long periods requires systems that minimize the unavoidable reduction in quality experienced by all fruit following harvest. High humidity CA environments are used to minimize the two mass loss processes of respiration and transpiration. Apples stored in refrigerated CA environments typically experience approximately 3% mass loss over the storage period; however, considerable variation exists between warehouses. Well designed and operated warehouses may realize 1-2% mass loss while others experience as much as 4-5% mass loss for similar storage periods (Tukey, 1992). Fruit variety, orchard location, seasonal weather conditions, harvest time, post-harvest handling, storage environment conditions, and storage time all contribute to and affect product mass loss during storage. Product mass loss affects fruit quality and marketability, and profits, since the product is sold by weight. Rapid establishment and long term consistency of desired environmental conditions minimize product mass and quality loss once the fruit is sealed in the warehouse.

Operation of the equipment necessary to maintain fruit under optimized conditions for long periods consumes large amounts of electrical energy. Bartsch (1982) estimated that the New York state apple industry consumed 201 kWh of electrical energy per metric ton of apples stored. Application of Bartsch's data to the Washington state apple storage industry suggests that approximately 280 million kWh of electrical energy or 32 MW of power are consumed annually. Electrical energy data metered from a single Pacific Northwest warehouse during the 1991-92 storage season projected to the entire Washington state apple storage industry yields an annual electrical energy consumption of 310 million kWh or 35 MW. Data provided by Green (1992) indicate the Washington apple storage industry consumes 400 million kWh of electrical energy or 45 MW of power annually. At current regional prices ($0.045/kWh), the cost of the energy consumed by the Washington state apple storage industry ranges from $12.6 to $18.0 million annually.

LITERATURE REVIEW

REFRIGERATION CYCLE

Refrigeration may be accomplished by liquid vaporization, thermoelectric means, steam injection systems, or by use of air, absorption, or vapor compression cycles. A simple mechanical refrigeration system utilizing a vapor compression cycle can be built using four essential components: compressor, condenser, evaporator, and expansion device. Additional components and system controls are included in virtually all practical systems for economy, safety, and maintenance assistance. Basic vapor compression refrigeration cycles incorporate boiling and condensing of a working fluid (refrigerant) at different temperatures and pressures. Heat added to the refrigerant at reduced temperature and pressure provides the latent heat necessary to convert the liquid to vapor. This vapor is then mechanically compressed to a higher pressure and corresponding saturation temperature. The absorbed latent heat can then be rejected, allowing the vapor to be condensed back to a liquid. The total cooling effect will be the enthalpy change of the refrigerant in the evaporator. A working system requires a connection between the condenser and evaporator to complete the circuit. A pressure reducing and metering valve compensates for pressure differences across the connection. Pressure reduction through the expansion valve causes some of the liquid refrigerant to flash to vapor. The latent heat necessary to vaporize the refrigerant comes from the liquid, thereby lowering the temperature. The refrigerant leaves the expansion valve as a saturated liquid-vapor mixture at a temperature corresponding to the lower pressure.

Most refrigeration systems use ammonia (R-717) or one of the halocarbons (R-11, R-12, R-22, R-502, etc.) as refrigerants for vapor compression cycles. Ammonia is the most commonly utilized refrigerant for vapor compression cycles in large fruit storage applications in the United States.

Fruit storage refrigeration systems are sized to rapidly remove field heat from the product. Following fruit cool-down, the amount of cooling required to maintain the desired storage temperature is much less. Fruit warehouse refrigeration systems are typically connected to many storage rooms which are not equally loaded throughout the year. Ideal refrigeration system design accommodates operation under variable loads.

<u>Compressors</u>

The compressor accepts low pressure dry gas from the evaporator and raises the pressure to that of the condenser. A compressor is generally chosen to have a displacement capacity adequate to handle the maximum refrigeration load of a system. Several problems can arise during operation under part load with a refrigeration system lacking capacity control. The evaporator flow control device usually throttles back the refrigerant mass flow rate when refrigeration load decreases. Constant displacement compressors, however, pump a constant volume of gas. A smaller mass quantity of refrigerant gas entering the compressor causes pressure and temperature to decrease. A drop in suction line pressure will result in lower evaporator temperature which may cause increased frost accumulation. Accumulated ice causes reduced heat transfer, resulting in a further drop in evaporating temperature and suction line pressure. A lower refrigerant mass flow rate may also carry an inadequate amount of oil back to the compressor, resulting in loss of lubrication.

All methods of compressor capacity control function by reducing the amount of compressed refrigerant delivered to the condenser. The simplest method of controlling capacity, on-off control, involves starting and stopping the compressor in response to refrigeration need. Fruit storage warehouses commonly utilize multiple reciprocating compressors. Operation of less than the full number of compressors, in sequence, can be considered a variation of on-off control. A multi-cylinder machine can have any number of its cylinders unloaded for capacity reduction. Smaller machines may have a valved bypass across the inlet and outlet ports in the cylinder head, or a variable clearance pocket in the cylinder head. Capacity may also be reduced by external bypass piping. Compressor speed may be controlled by two-speed electric motors or by electronic variation of motor speed.

Condensers

The condenser accepts the high temperature, high pressure gas from the compressor and rejects the superheat and latent heat absorbed in the evaporator. Refrigerant exits the condenser as a moderate temperature, high pressure liquid. The liquid is usually slightly subcooled by the time it reaches the expansion valve.

Expansion Valves

The expansion valve controls refrigerant flow from the high pressure condensing side of the system into the low pressure evaporator. Pressure reduction is achieved through either a modulating or two-position variable flow orifice.

Evaporators

The evaporator receives low pressure, low temperature fluid from the expansion valve. Multiple finned distribution tubes within the evaporator provide close thermal contact with the room heat load. Absorbed heat causes the liquid refrigerant to vaporize within the evaporator. The refrigerant generally leaves the evaporator either as a saturated or slightly superheated vapor.

Air cooling evaporators have fans to move air past finned pipe coils. Evaporators designed for use with ammonia systems have aluminium fins on either stainless steel or aluminium tubes. In most designs, tube diameters vary from 9 mm to 32 mm, and fin spacings range from 2 mm to 12 mm, depending on coil size and application.

Evaporator coils designed for use in apple and pear storages must be capable of maintaining high relative humidities and temperatures near 0°C. Room relative humidity directly affects product mass loss and fruit quality, especially in long term storages. Evaporator coils with large amounts of fin surface area, operated at high airflow rates, permit low temperature differences between the cooled air and evaporating refrigerant. Evaporator coils sized too small must operate at a lower refrigerant temperature to produce the same refrigeration effect as one with more surface area. Evaporator coil design is important because the coil's operating temperature directly affects the dew point temperature of the room. High ambient dew point temperature, and therefore relative humidity, are achieved when evaporator coils are operated at temperatures closer to the desired room temperature.

Fin spacing is also important to evaporator coil design. Water vapor in the surrounding air will freeze or condense on fin surfaces that are at a temperature below the room dew point temperature. Accumulated frost hinders heat transfer and must be removed periodically. A coil with close fin spacing will require more frequent defrosting than a coil with greater fin spacing. Coil fin spacings of 6.4 to 8.5 mm (3 to 4 fins per inch) are recommended for apple storages. A noticeable increase in defrost cycle frequency is observed as coil spacing decreases to 5.1 mm (5/inch) (Staples, 1992).

Two commonly used types of refrigeration coil systems include dry or direct expansion evaporators and flooded evaporators. Large ammonia refrigeration systems are most popular in Pacific Northwest fresh produce storages. Flooded coil evaporators are kept full of boiling liquid refrigerant supplied from the low pressure receiver. Pressure inside the evaporator and low pressure receiver is controlled by a thermostatically controlled pressure modulating regulator (back pressure regulator). State of the art flooded systems constantly vary the rate of cooling as prescribed by a single thermostat located in the storage room. The thermostat responds to changes in room temperature by modulating the back pressure regulator in direct proportion to the difference between the set point and sensed temperature.

Flooded coil evaporator systems have several advantages over direct expansion systems when applied to fruit storage warehouses. A flooded coil, in which most or all of the evaporator tube surface is wetted by liquid refrigerant, is more effective than a partially wetted direct expansion coil. A smaller temperature differential between the cooled air stream and evaporating refrigerant is possible, permitting higher room relative humidity. A flooded coil system can also be more energy efficient since a higher evaporating temperature requires less compressor power. These advantages outweigh the fact that flooded coil evaporator systems are generally more expensive than direct expansion coils. A large quantity of refrigerant is necessary to ensure flooding, resulting in the need for an accumulator and related piping not needed with direct expansion systems.

ENERGY BUDGET

Energy consuming components of the refrigeration system include motors that operate evaporator fans, compressors, compressor cooling pumps, condenser fans and pumps, and evaporator coil defrost equipment. Evaporator fan energy use is the single largest component of the annual storage facility electrical budget. Bartsch (1986) monitored energy consumption of various components of the refrigeration system for a 2290 metric ton apple storage facility in New York state. After loading and cool down, the evaporator fans consumed 750 kWh per day while compressor use averaged 500 kWh per day. The condenser fans and pump consumed 30 kWh per day. Waelti (1989) estimated daily power consumption in a 386 metric ton CA room to be 275 kWh upon cool down. Evaporator fans used 170 kWh/day and compressors consumed an estimated 105 kWh/day.

Heat gain from evaporator fans is the heat equivalent of the electrical power used to drive the fan motors including efficiency losses. The evaporator fans are also the largest heat source in a storage room. On an overall seasonal basis, the heat input from evaporator fans is often more than the heat gain from all other sources (Sainsbury, 1985). Bartsch (1986) indicated that the heat added by the evaporator fan motors upon cool down is 2-3 times greater than the heat of respiration generated by the fruit. Adre and Hellickson (1989) reported evaporator fan energy use ranged from 45-52% of the total refrigeration load for several CA storage rooms located near Hood River, Oregon. Heat of respiration generated by the stored product comprised 28-33% of the heat load while conduction, infiltration, and miscellaneous loads contributed the remaining 20-21% of the heat load.

ENERGY CONSERVING METHODOLOGIES

The energy crisis of the 1970's led to widespread energy conservation measures. Additional energy conservation programs have been articulated by regional utilities in 1992 (Bonneville Power Administration, 1992a and Pacificorp, 1992). When river levels are low, the Pacific Northwest currently produces just enough electricity to meet the region's needs. The Northwest Power

Planning Council's most recent power plan (1991) calls for the region to immediately begin acquiring 1500 MW of conservation and 800 MW of specific additional low-cost power resources. The goal is to have these in place by the year 2000.

Fruit storage warehouses have excellent potential for energy conservation. Traditionally, evaporator fans run continuously when the room is loaded regardless of cooling requirement. The Eastern United States and Europe initiated intermittent evaporator fan operation (fan cycling) to offset rapidly rising electric rates (Bartsch, 1986 and Waelti, 1987). Implementation, however, was not widespread. Low electric rates and concerns for storage environment stability prevented warehouse managers in the Pacific Northwest from adopting fan cycling. However, recent increased costs of electricity in the region, continued emphasis on energy conservation, and the uncertainties of salmon recovery policies on Columbia River hydropower generation have caused warehouse operators to consider equipment changes and management procedures, including evaporator fan cycling, that will reduce electrical energy consumption.

One basic method of conserving energy is to minimize the refrigeration load. Refrigeration load is comprised of many elements including heat conduction from warmer areas through the building structure, solar radiation, convection gains from infiltration, heat input from auxiliary equipment (fan motors, pumps, defrosting, etc.), and internal electric lighting. The magnitude of each of these elements needs to be determined so that the cost of modifications and associated reductions in load can be justified.

Facility operating conditions should be compared with design specifications to identify discrepancies. Items that reduce refrigeration performance include incorrect adjustment of controls, dirt on coils and fans, ice on coils, obstructions to airflow such as poorly stacked produce, shortage of refrigerant or excess oil in evaporator, incorrect adjustment of expansion valve, worn machinery, and poorly adjusted drive belts. Incorrect operation of pumps or valves, causing supply tanks to overflow with the loss of chilled or hot water, and damaged insulation and vapor barriers also contribute to unnecessary energy use.

Refrigeration systems designed for energy economy utilize efficient compressors and fan motors. High efficiency motors consume 2 to 6% less electrical energy than standard motors, depending on size and load (Bonneville Power Administration, 1992b). Two-speed or electronic adjustable speed drives for compressor, pump, and fan motors can also reduce energy consumption.

Refrigeration systems operated at lowered condensing temperatures improve energy efficiency. Excessive condenser pressure is the greatest cause of unnecessary power consumption and reduced system performance (Trott, 1989). Wilcox (1989) developed a computer model to estimate energy, demand, and cost savings resulting from reduced condensing pressures. Reductions of 9.8 to 11.7% in combined compressor and condenser fan power were observed during reduced condenser pressure operation. Systems operated at higher evaporating temperatures also improve energy efficiency as lower evaporating temperatures require increased compressor power. This can be achieved through the use of large, flooded evaporators.

Thompson and Knutson (1989) reported that electrical energy consumption can be reduced by lengthening the cool down period. Lengthening cooling times can sometimes be accomplished without any equipment changes and was shown to be very attractive financially. Product mass loss and fruit quality were not investigated by Thompson and Knutson (1989). Electrical energy savings obtained through extending the cool-down period may be offset by increased product mass loss and a reduction in fruit quality, since product respiration is highest during this period.

Evaporator fan cycling can potentially conserve large amounts of electrical energy with low financial investment. Fan cycling not only saves energy through reduced fan operation time, but also significantly decreases heat load in the storage space, which further decreases refrigeration system operating time. Bartsch (1986) stated that a fan cycling scheme in which the evaporator fans remained off 50% of the time would reduce compressor and condenser use by 25%, resulting in a total savings of 40% during fan cycling operation. Similarly, a 75% evaporator fan off time would yield an overall savings of 59% during fan cycling operation. Simulated results of a six hours on and six hours off fan cycling scheme presented by Adre and Hellickson (1989) indicated a 24% reduction in overall seasonal refrigeration system energy use could be achieved.

EVAPORATOR FAN CONTROL STRATEGIES

Early fan cycling practices initiated in the Eastern United States and Europe consisted of turning refrigeration systems off at night and on in the morning (Bartsch, 1986 and Waelti, 1987). Other warehouse operators reduced electrical energy cost by turning refrigeration systems off during periods of high electrical energy demand (Waelti, 1987). Some facilities reduced air circulation rates by disabling a portion of the evaporator fans once the product reached a stable temperature.

Other facilities employed time-clock control of all the evaporator fans. Various schemes have been used including 12 hours on, 12 hours off, or six hours on, six hours off (50% off time); six hours on, 12 hours off (67% off time); 1.5 hours on, 4.5 hours off (75% off time), etc. These control strategies have proven effective in conserving energy and maintaining relatively constant fruit temperatures; however, varying heat loads can not be fully accommodated. By utilizing a time-clock control strategy in a 2290 metric ton apple storage facility, Bartsch (1986) demonstrated a total energy savings of nearly 50% and stated that a potential 59% could be saved. This corresponded to a potential annual savings of 9 million kWh as a result of evaporator fan cycling in CA storages in New York state. Waelti (1987) and Yost (1984) both indicated that fruit temperatures can be maintained within desired tolerances of ±0.5°C with intermittent evaporator fan operation as well as with continuous fan operation. Waelti (1987) measured fruit temperature in the center of the bin and 10 cm below the surface of the bin.

Yost (1984) measured fruit temperature by inserting a probe into the flesh of an apple. Waelti (1989) maintained fruit temperature measured at the center of the bin within ±0.3°C in a fan cycled room. The temperature of the fruit on the surface of the bins and the outer surface of the fruit will fluctuate more than that measured by Waelti and Yost. Several time-clock schemes were practiced by both Waelti and Yost. Although fruit temperatures in the fan cycled room did not vary significantly more than in non-fan cycled rooms, mass loss as measured by Waelti (1989) was substantial at 3.5% in a non-fan cycled room and 4.3% in a fan cycled room. Yost did not monitor fruit mass loss.

Another control method is to cycle fans and refrigeration with a solid-state thermostat. A remote temperature sensor located in the room controls the fans and refrigeration together. Thermostat placement is crucial because air is much more responsive to temperature changes than the fruit. Air has a lower specific heat and there is much less air mass; therefore, too frequent and unnecessary fan cycling may take place. Bartsch (1986) reported that in many CA rooms the temperature controller, not the fan cycling practice, was the cause of major temperature variations.

It is important that fan cycling does not adversely affect fruit quality or product mass loss. Fluctuations in the storage environment can harm product quality and cause increased mass loss, which offset savings due to fan cycling. For example, at $0.045/kWh a 2290 metric ton storage facility that fan cycles 50% off may achieve an overall annual savings of $4600 while a 1% mass loss at a market price of $400 per metric ton represents $9200 (Bartsch, 1986).

OBJECTIVES

Regional emphasis on energy conservation and the growing volume of refrigerated storage combine to increase the need for energy conservation measures in fruit storage facilities. Therefore, an experiment was developed to cycle evaporator fans as demanded by the fruit mass temperature. Controlling the refrigeration system with the fruit mass temperature would allow fruit temperature to remain as stable as the prescribed set-point temperature rise, and storage environment conditions to remain relatively stable. This control strategy would also enable maximum energy savings, since fans would remain off if no cooling was needed. The objectives specifically identified for this research are:

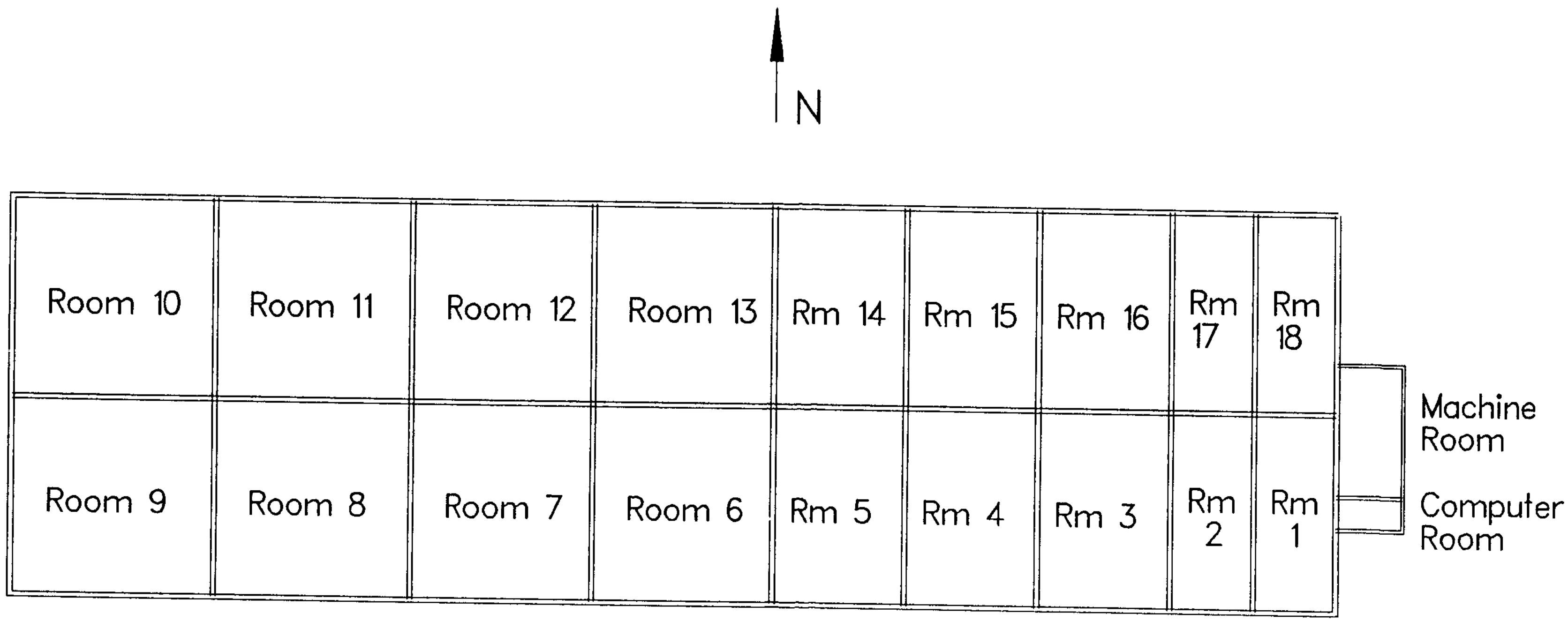
1. Cycle evaporator fans as demanded by the bulk fruit temperature in two CA apple storage rooms using a computer to control the storage room temperature.
2. Document and compare energy consumption, room and fruit temperature, and fruit quality and mass loss in the two fan cycled rooms to two similar rooms in which evaporator fans operated continuously.
3. Project total potential energy savings for a warehouse implementing this scheme in all of their storage rooms.
4. Project total potential energy savings to the Washington state apple storage industry.

PROCEDURE

A two year project conducted during the 1990-91 and 1991-92 storage seasons was initiated to document energy savings and fruit quality preservation in full-sized CA apple storage rooms as a result of evaporator fan cycling. Four rooms in the Snokist Growers Mead Avenue Complex B fruit storage warehouse in Yakima, WA were selected for this research project. Complex B has 18 rooms with a total storage capacity of 9410 metric tons (24,390 bins, each sized 1.2 X 1.2 X 0.61 m). Rooms 4 and 5 on the south side of the structure and Rooms 14 and 15 on the north side were selected as test rooms (Figure 1). Each test room had a storage capacity of approximately 460 metric tons (1200 bins) of fruit and measured 12.5 m wide by 19.5 m long by 8.5 m high. Individual bins within a room were identified by row, stack, and bin numbers as shown in Figure 2. For example, the bin on the floor at the left rear of a room (under the evaporator coils) was designated as Row 1, Stack 14, Bin 1, or R1S14B1. After stable storage environmental conditions were achieved, Rooms 4 and 15 were fan cycled and Rooms 5 and 14 served as controls by functioning under traditional means of continuous fan operation.

FACILITY REFRIGERATION EQUIPMENT

Two ammonia flooded coil evaporator units equipped with five 373 W, 460 Volt fan motors were resident in each test room. Evaporator coil fin spacing was 8.5 mm (3/inch). Each fan was capable of delivering air at a rate of 9300 m^3/h. The facility



Fan Cycled Rooms: 4 and 15

Control Rooms: 5 and 14

Figure 1. Facility Room Layout

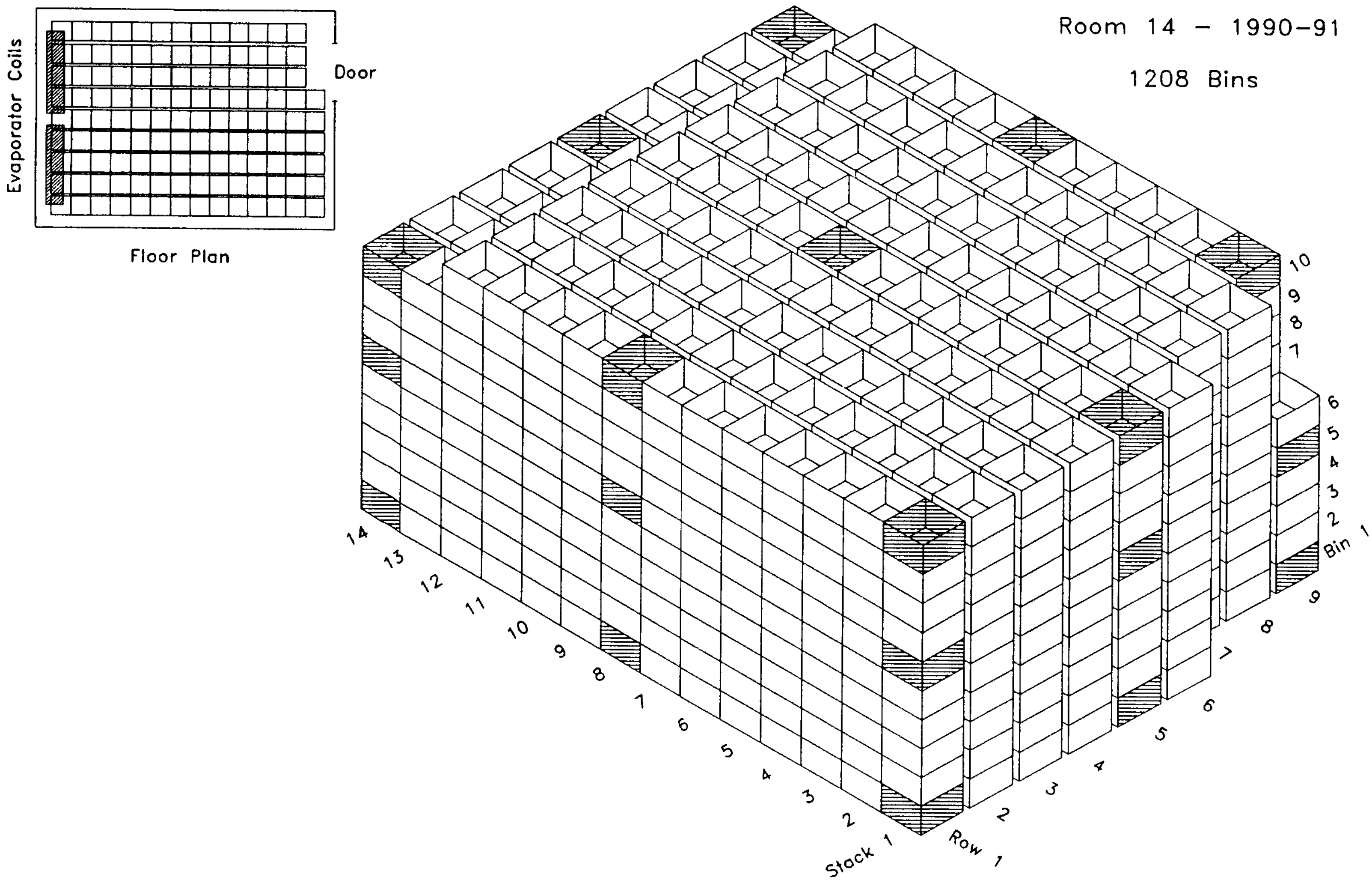


Figure 2. Typical Bin Stacking Pattern

refrigeration system contained two reciprocating compressors (4 cylinder, 45 kW and 8 cylinder, 75 kW) and one rotary screw compressor (260 kW). The refrigeration system has a total heat absorbing capacity of 1500 kW at -7°C suction temperature and 35°C condensing temperature. A schematic of the refrigeration supply and return lines to the evaporator coil for a typical room is shown in Figure 3. The facility contained an evaporative condenser equipped with a 15 kW axial fan motor and a 7.5 kW pump. A 3.7 kW pump was utilized to distribute water used to defrost evaporator coils.

INSTRUMENTATION

All four test rooms were instrumented to provide dry-bulb temperatures at 39 locations, one dew point temperature, weight loss from approximately 100 apples, atmospheric gas sampling at three locations, and refrigerant flow into the evaporators. Fan electrical energy was measured either directly or indirectly in each experimental room. Refrigerant temperature entering the condenser was measured at one location.

Temperature was measured using thermistors with 0.017°C resolution. All thermistors were recalibrated using an ice bath between the two test seasons. Dew point temperature was measured with an optical-condensation hygrometer. Relative humidity was calculated using psychrometric relations of dry-bulb and dew point temperatures (ASHRAE, 1989).

Strain gage load cells with 45 kg capacity were used to continuously monitor fruit mass. The load cells were wired to an amplifier which provided measurement of mass in increments of

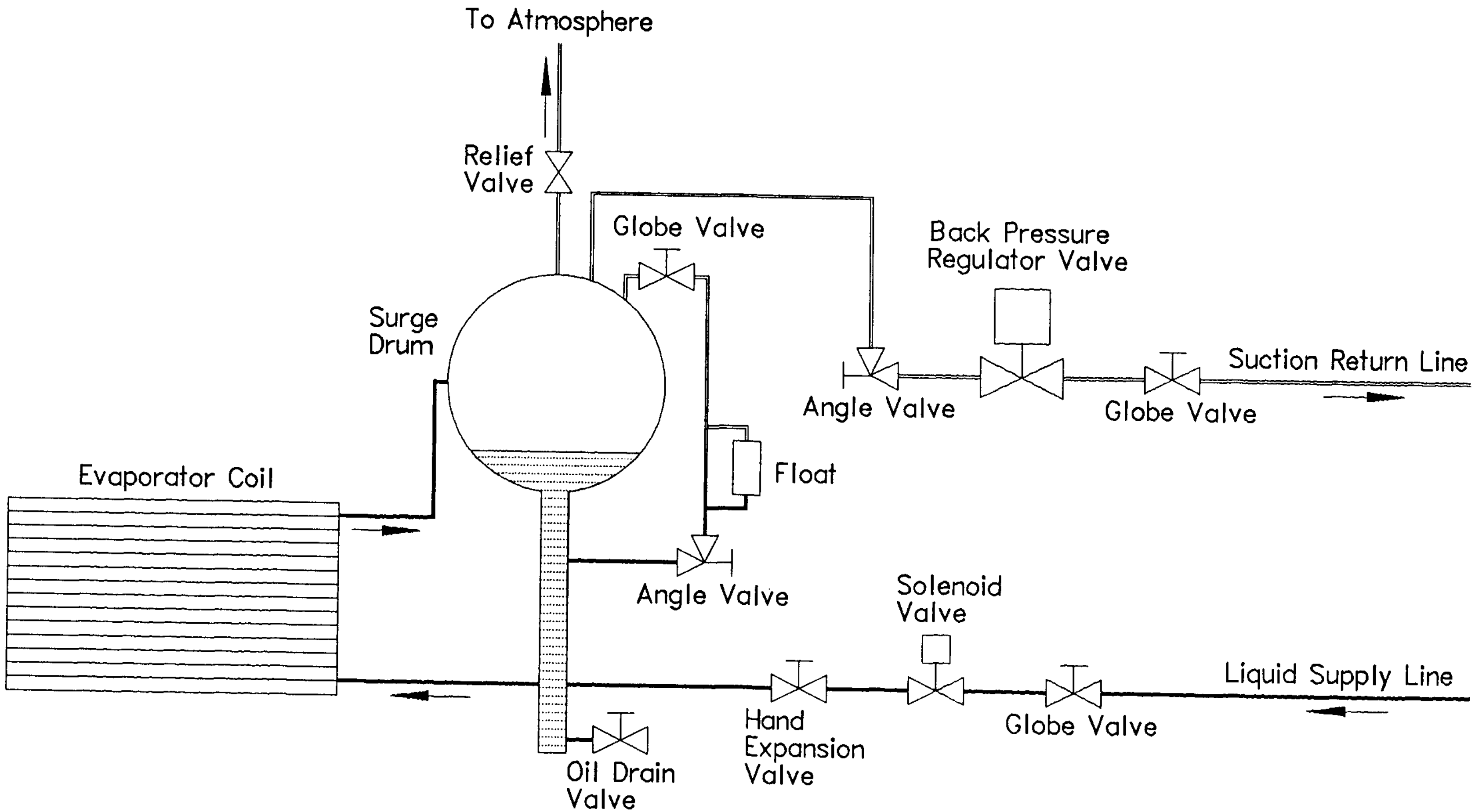


Figure 3. Typical Room Refrigeration Equipment Arrangement

4.54 g. Atmospheric gas samples were drawn twice daily from each room by Snokist personnel to measure carbon dioxide and oxygen content. Dates and times air was added to each room were also documented. Refrigerant flow was measured with paddle-wheel flow meters inserted into each room's evaporator supply line.

Electrical power use by half of the evaporator fans in Room 4 was measured with a watt-transducer during both seasons. Each test room was equipped with two identical evaporator units. Therefore, power used by the five fans of one evaporator unit in Room 4 was doubled to obtain room total fan electrical energy consumption. Fan on-off times were monitored in the two experimental rooms (4 and 15) during both storage seasons. A second watt-transducer was installed in Room 15 to measure energy use of all ten evaporator fans during the 1991-92 storage season.

The resident computer that monitored and controlled various environmental parameters was programmed to record data at either one or five minute intervals in the four test rooms depending upon the functions being monitored. Data were periodically transferred from the computer's hard disk memory to floppy disks and mailed from Yakima to the Bioresource Engineering Department at Oregon State University for analysis. Approximately three million data records were accumulated each month under this operational scheme.

ROOM PREPARATION

Each test room was filled with 'Red Delicious' apples using standard warehouse procedures. As bins were stacked in a room, 27 thermistors were placed in the bins designated for monitoring (cross-hatched bins in Figure 2). Thermistors used to monitor fruit temperature were placed within the fruit mass, but not inserted into the flesh of an apple. Thermistors were placed beneath at least two, but not greater than three layers of fruit. Nine thermistors were placed to monitor room air temperatures at predetermined locations. Air temperatures were measured at the evaporator inlet and outlet, and at several air spaces located at 0.6 m and 7.6 m below the ceiling. Thermistor probes were placed within 0.5 m of the evaporator fan inlet and within 1.5 m downstream of the evaporator fan outlet. Refrigerant temperatures were measured into and out of the evaporator coils, and immediately following the hand expansion valve. Refrigerant temperatures were obtained by placing the thermistor probe in contact with the pipe beneath an outer layer of insulation.

The load cell apparatus, dew point sensor, and one thermistor were placed into an empty bin located at R5S8B10. A perforated plastic sheet was stapled to the the top of the bin to prevent excess air movement around the load cell created by the evaporator fans. Required airflow past the dew point sensors was provided by two vacuum pumps located in the equipment hallway between opposite rooms.

Closure dates for the 1990-91 season for Rooms 4, 5, 14, and 15 were October 3, 10, 6, and 9, and opening dates were July 18, 31, June 13, and June 21, respectively. Thus, actual storage periods for the 1990-91 season were 288, 294, 250, and 255 days, respectively. Room closure dates for the 1991-92 storage season were October 4, 12, September 30, and September 28, for Rooms 4, 5, 14, and 15, respectively, and opening dates were July 30, July 16, January 15, and April 16, respectively. Actual storage periods for the 1991-92 season were 300, 278, 107, and 201 days, respectively.

FRUIT QUALITY MONITORING

Snokist personnel recorded standard fruit quality parameters from samples taken from each numbered bulk lot placed in each room. Recorded data as the fruit was placed into storage included high, low, and average pressure (flesh firmness) from three apples, and starch content and soluble solids content (sugar content) measured from one apple. High, low, and average pressure values were recorded from three apples from the same bulk lot when the fruit was removed from storage. Very little starch remains in apples after storage due to metabolic activity. Thus, starch content was not measured at the end of storage. Soluble solids content was not measured at the end of storage since flesh firmness is a better indicator of fruit quality at that time.

Flesh firmness is a force measurement that is used as an indicator of how ripe fruit is. Lower force measurements indicate riper fruit. Flesh firmness decreases with time in storage. Starch was measured by means of an iodine test and scaled one to six. A one indicates an immature fruit (100% starch) and a six indicates on over-mature fruit (0% starch). Soluble solids content, measured with a refractometer, is used as an indicator of sweetness because sugars are a major component of soluble solids. Soluble solids content typically increases over time in storage. Recommended values of these fruit quality indices at harvest depend on several factors including intended storage duration, atmosphere and temperature establishment rates, and fruit strain. Flesh firmness for 'Red Delicious' apples should be greater than 71.2 N and starch content should be 2.0 to 2.5 at harvest for a storage duration of 6-7 months. Flesh firmness should be greater than 73.4 N and starch content 1.5-2.0 at harvest for a storage duration of 10 months (Tukey, 1992).

FAN CYCLING

All four rooms were operated as conventionally managed storages for approximately one month after closure. During this period, the evaporator fans operated continuously except for 25 minute defrost cycles. Defrost cycles were executed twice each day during the first two to three weeks of storage and once per day during the remainder of the storage periods. On November 14, 1990, and November 1, 1991, fan cycling in both Rooms 4 and 15 was initiated as a function of the average temperature of five thermistors placed among the fruit. The five thermistors that controlled fan cycling during the 1990-91 season were located at R1S13B6, R5S1B6, R5S8B6, R9S1B1, and R9S14B1 in Room 4, and at R1S1B10, R1S14B1, R5S8B6, R9S2B1, and R9S14B6 in Room 15. The five that controlled fan cycling during the 1991-92 storage season were R1S13B8, R5S1B10, R5S8B6, R9S1B1, and R9S13B6 in Room 4, and at R1S1B10, R1S14B9, R5S8B6, R9S2B1, and R9S14B6 in Room 15. The computer software used to control the system was modified to allow the averaged temperature monitored by the five thermistors to increase 0.11°C before restarting the evaporator fans and refrigeration. The evaporator fans were turned off and refrigerant flow ceased when the average temperature returned to the set point value. The evaporator fans in Rooms 5 and 14 operated continuously except for the daily defrost period.

FRUIT MASS LOSS

Two separate procedures were incorporated to determine fruit mass loss during each storage season. Approximately 16 kg of apples were placed in a plexiglass box 0.51 m long by 0.30 m wide by 0.25 m high to simulate bulk storage conditions during the 1990-91 storage season. The plexiglass box of fruit was suspended from a strain gage load cell, which enabled continuous measurement of sample mass without disrupting the fruit or the environment. An additional 20 apples were individually numbered, weighed on a scale with 0.1 g resolution, and placed on top of fruit in a wooden bin. Immediately after each room was opened, exit weights of the 20 numbered fruit were measured and recorded.

Approximately 16 kg of apples (70-80 fruit) were individually numbered, weighed, placed in a plexiglass box, and suspended from a strain gage load cell in each room during the 1991-92 season. An additional 20 apples were numbered, weighed, and placed in contact with the surface of a wooden bin (Figure 4). Immediately after each room was opened, exit weights of all fruit were measured and recorded.

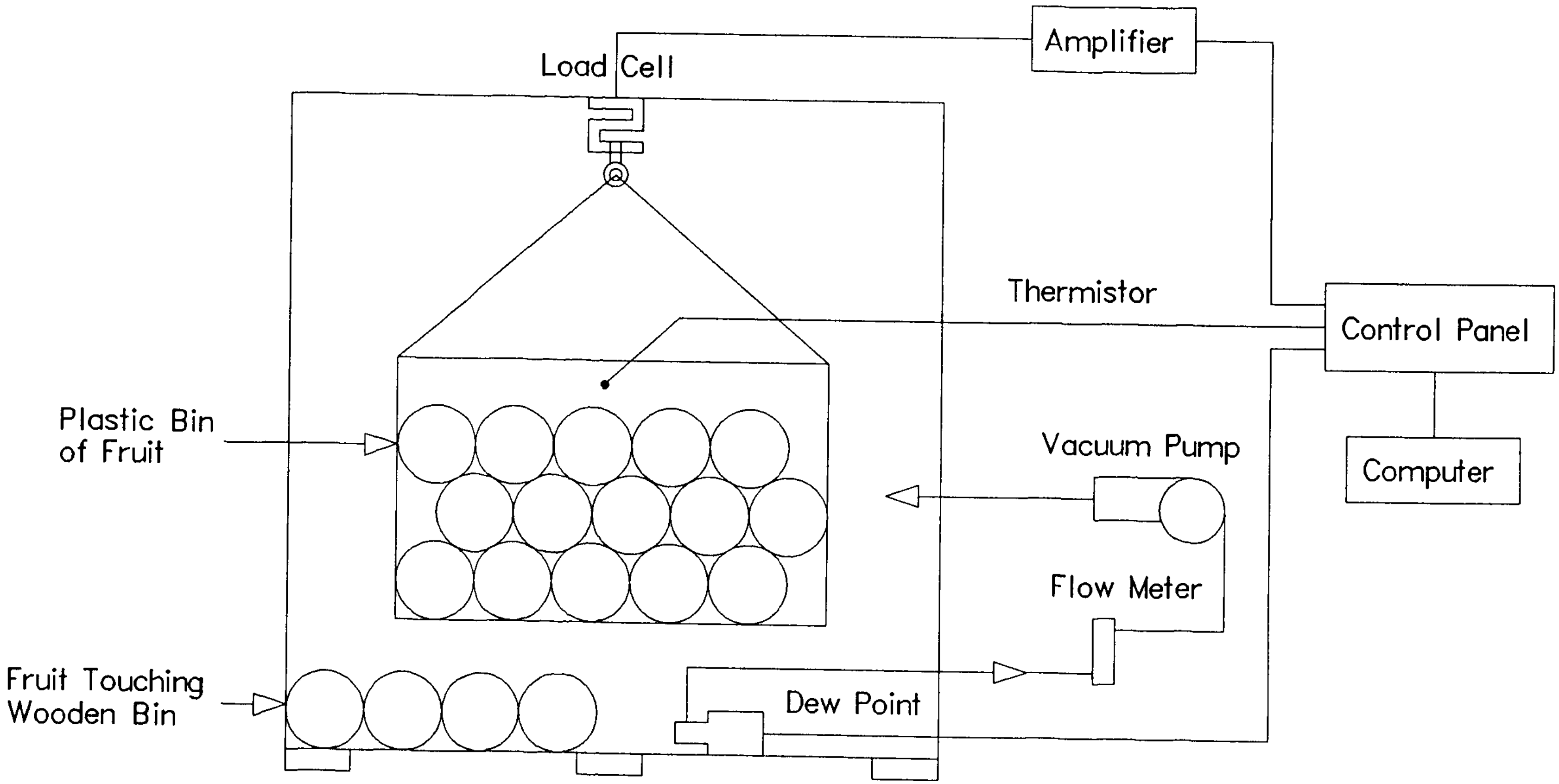


Figure 4. Test Bin and Instrumentation

RESULTS AND DISCUSSION

ELECTRICAL ENERGY CONSUMPTION

Fan energy use for continuous operation was obtained by averaging data recorded in parts of October and November prior to initiation of fan cycling during the 1990-91 season in Room 4, and during the 1991-92 season in Rooms 4 and 15. Average power use was 4368 W and served as the base for continuous fan operation energy use. Figure 5 illustrates the recorded power use in Room 4 on October 26, 1990. The typical minimum variation of the data trace, except during the 25 minute defrost cycle when the fans were deactivated, indicates that power use by the fans was nearly constant under full operation. Average energy use by the evaporator fans during continuous operation (23.58 hours per day) was 103.0 kWh/day. All evaporator units were of identical manufacturer and size, thus energy savings due to fan cycling were determined from this base value.

Tables 1 and 2 summarize evaporator fan electrical energy use in Rooms 4 and 15 for the 1990-91 and 1991-92 storage seasons, respectively. The average energy use per day in Room 4 during October 1990, and in Rooms 4 and 15 during October 1991, was lower than the expected value of 103.0 kWh/day because the evaporator fans were shut off for short periods during several days. Monthly energy savings ranged from 63.1 to 89.5% in Room 4 and from 41.1 to 84.4% in Room 15 during the first season. Fan energy savings during fan cycling months averaged 77.7 and 72.0% for Rooms 4 and 15, respectively, during the first storage season.

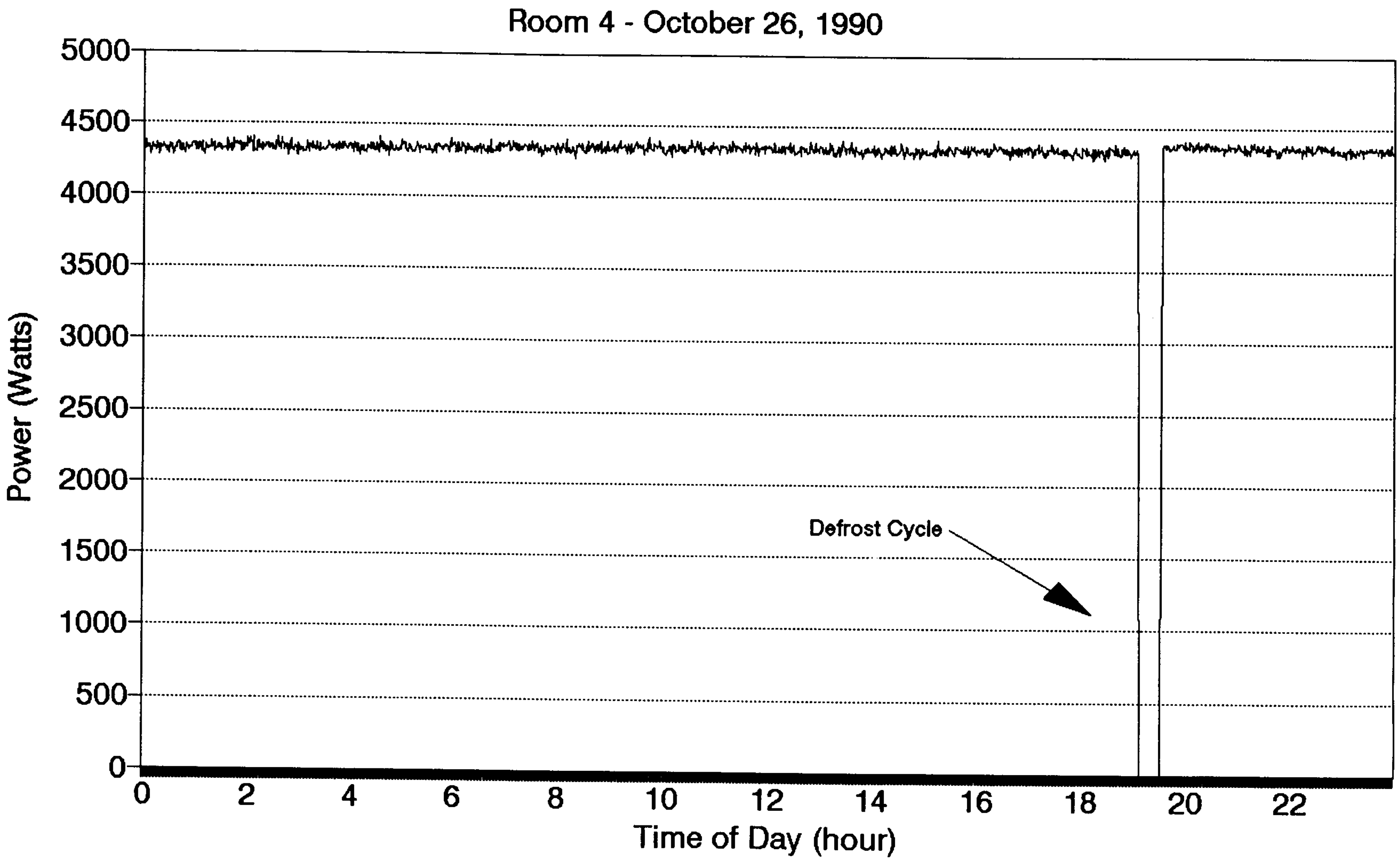


Figure 5. Typical Evaporator Fan Energy Use Prior to Fan Cycling

Table 1. Evaporator Fan Electrical Energy Use During 1990-91

	1990-91 SEASON			
	Room 4		Room 15	
Month	Avg Power (kWh/day)	Savings* (%)	Avg Fans On (%/day)	Savings* (%)
October	100.5	-	N/A	-
Nov 1-11	102.6	-	N/A	-
Nov 12-30	16.3	84.2	28.5	71.5
December	26.1	74.7	58.9	41.1
January	10.8	89.5	36.9	63.1
February	22.6	78.1	23.0	77.0
March	17.1	83.4	15.6	84.4
April	38.0	63.1	18.4	81.6
May	25.7	75.0	19.4	80.6
June	23.4	77.3	16.3	83.7
July	28.1	72.7	Room Open	Room Open
Fan Cyc. Avg	23.0	77.7	28.0	72.0
Season Avg	33.1	67.9	37.4**	62.6**

* Based on 103 kWh/day during continuous fan operation

** Estimated value

Table 2. Evaporator Fan Electrical Energy Use During 1991-92

	1991-92 SEASON			
	Room 4		Room 15	
Month	Avg Power (kWh/day)	Savings* (%)	Avg Power (kWh/day)	Savings* (%)
October	99.6	-	99.6	-
November	49.2	52.2	53.5	48.1
December	36.6	64.5	45.9	55.4
January	32.7	68.3	39.6	61.5
February	34.4	66.6	33.1	67.8
March	56.3	45.3	50.2	51.2
April	53.4	48.2	39.0	62.1
May	63.6	38.3	Room Open	Room Open
June	85.1	17.4		
July	84.1	18.4		
Fan Cyc. Avg	54.6	47.0	44.1	57.2
Season Avg	58.7	43.0	53.0	48.5

* Based on 103 kWh/day during continuous fan operation

Average fan energy savings over the course of the entire season was 67.9% in Room 4 and an estimated 62.6% in Room 15. Fan operation data from Room 15 was not obtained during the first one and a half months (during continuous fan operation) of storage in 1990.

Energy savings were considerably lower during the 1991-92 season. Energy savings during fan cycling months ranged from 17.4 to 68.3% and averaged 47.0% in Room 4. Savings ranged from 48.1 to 67.8% and averaged 57.2% in Room 15. Average fan energy savings over the entire season was 43.0% in Room 4 and 48.5% in Room 15.

Figures 6, 7, and 8 illustrate the recorded energy use values during the entire storage seasons in Room 4, 1990-91, and in Rooms 4 and 15, 1991-92, respectively. Evaporator fans operated unnecessarily for periods during the end of December 1990 and during the last week of April 1991. Software design and implementation problems caused unstable evaporator fan operation during the first half of November 1991. Refrigeration system malfunctions and evaporator fan control problems also occurred during the second half of the 1991-92 storage season in Room 4, especially during the months of June and July. Analysis of fan on-off data show that the fans were on continuously and defrost cycles were not executed (discussed later).

The data in Tables 1 and 2 reflect actual fan performance as shown in Figures 6, 7, and 8. Energy savings from evaporator fan cycling is potentially higher. If times when evaporator fans operated unnecessarily are eliminated, energy savings during months of fan cycling would approach 60 to 65% and overall seasonal average savings would approach 50 to 55% for the 1991-92 season.

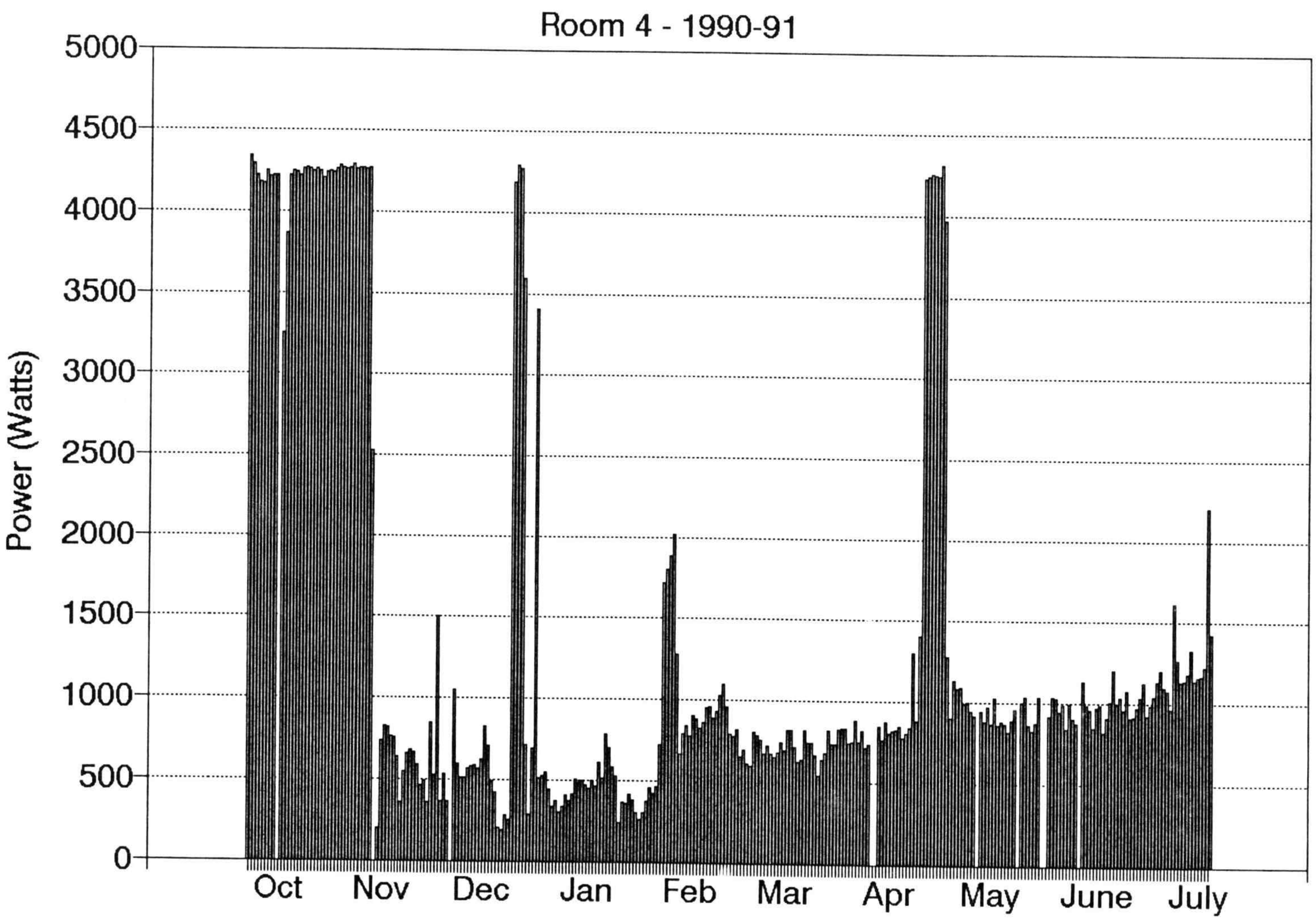


Figure 6. Evaporator Fan Electrical Energy Use in Room 4 During 1990-91

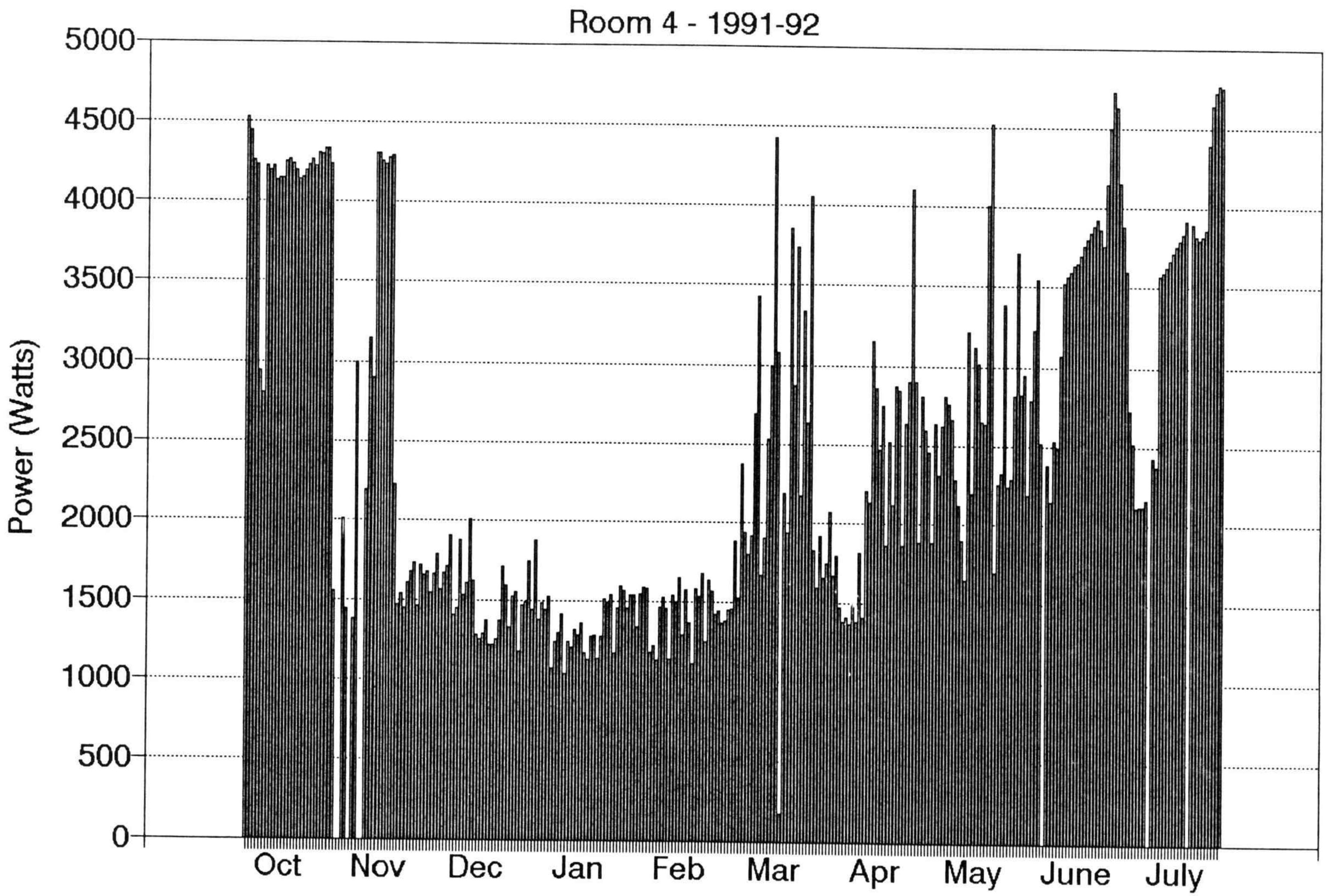


Figure 7. Evaporator Fan Electrical Energy Use in Room 4 During 1991-92

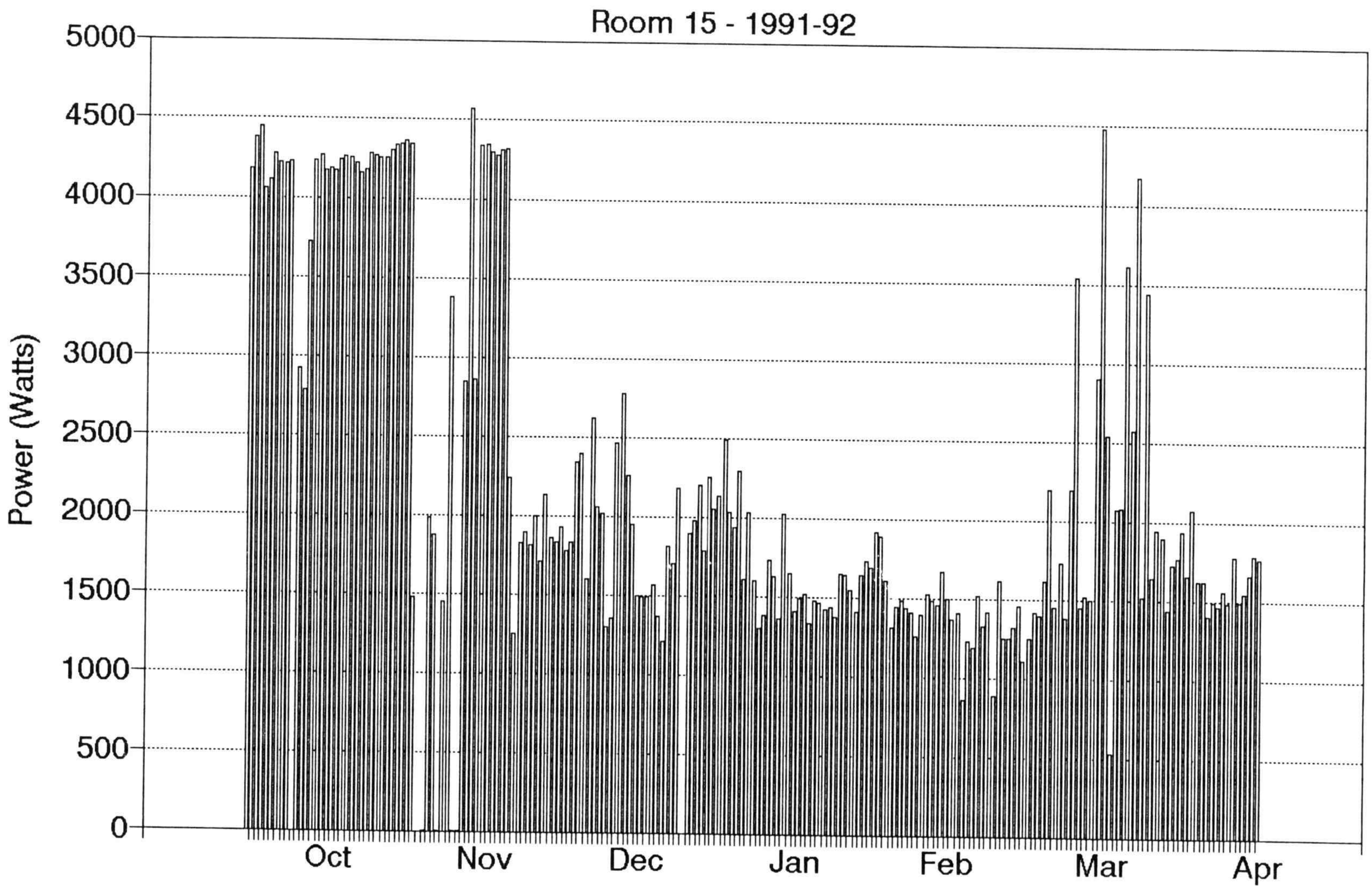


Figure 8. Evaporator Fan Electrical Energy Use in Room 15 During 1991-92

Increased electrical energy consumption near the end of the storage periods during both seasons can be partly attributed to increased refrigeration load as a result of increased conduction heat gain in the summer months. Monthly average outside air temperatures during both storage seasons recorded at the Yakima municipal airport and the mean temperature are shown in Figure 9. Monthly average temperature ranged from -5.1°C in December 1990 to 21.4°C in July 1991 during the first season, and from 1.6°C in December 1991 and January 1992 to 21.1°C in June 1992 during the second season. The annual average outside temperature was considerably warmer during the second season as compared to the first season. Average temperature for the 11 month storage season was 8.6°C during the first season and 10.5°C during the second season. The average temperature during this period is 8.9°C.

Modified operation and control of the evaporator fans and back pressure regulator (discussed later) caused somewhat lower savings in Rooms 4 and 15 during the 1991-92 storage season.

FREQUENCY OF FAN CYCLING

One of the thermistors selected to control operation of the evaporator units in Room 4 during the 1990-91 season proved to be unstable. The thermistor located at R5S1B10 reacted almost instantaneously to operation of the evaporator fans and quickly cooled below the set point temperature. Although the temperature monitored by the thermistor at R5S1B10 was averaged with the temperature measured by four other thermistors, dynamic fluctuation at R5S1B10 caused the evaporator unit to cycle on and

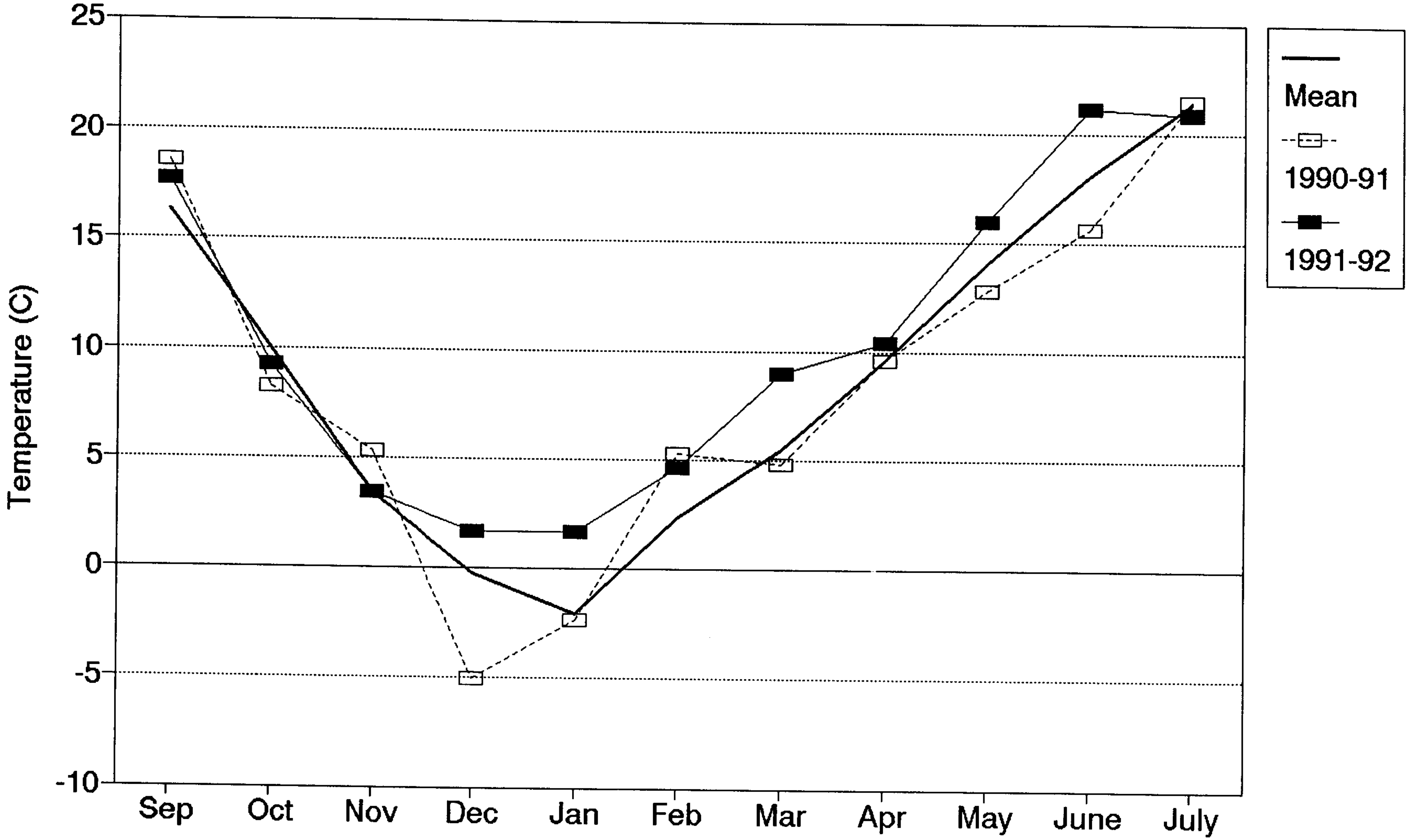


Figure 9. Yakima Average Monthly Temperature

off nearly continuously. This unstable condition was rectified by selecting the thermistor located at R5S1B6 to be one of the five averaged for fan cycling control. Observation of the thermistor at R5S1B10 during room unloading revealed that placement was on top of, rather than within, the apple mass.

Periodic analysis of fan power recordings in Room 4 and fan on-off periods in Room 15 during the 1990-91 season indicate that the evaporator fans cycled more frequently in Room 4. Comparisons on November 26, 1990 show the evaporator units cycled 18 times in Room 4 and three times in Room 15. On January 10, 1991, the fans cycled four times in Room 4 and twice in Room 15. Although the fans cycled on and off more frequently in Room 4, operating times were consistently shorter as is reflected in the monthly and overall percent energy savings listed in Table 1. The reduced frequency of fan cycling experienced in Room 15 versus Room 4 may be attributed to several factors. Placement of the five thermistors averaged to control the fan cycling was not exactly the same in each room. Room 15 was also located on the north side of the warehouse and did not experience direct solar radiation on the vertical exterior wall as did Room 4. Differences in actual operation of the back pressure regulator in each room also directly affected cooling rates and on-off times.

Thermistor placement in the apple bins was carefully verified to insure correct depth (at least two, but not greater than three layers of fruit) in the fruit mass during room loading in 1991. Fan cycling frequency was found to be three times per day in Room 4 and two times per day in Room 15 during February 7 and 8, 1992. Total fan on-time for Room 4 during these days was 396 and 367 minutes, respectively. The evaporator fans were on for 456 and 408 minutes, respectively, in Room 15 during the same days.

RELATIVE HUMIDITY

Relative humidity values were determined from the psychrometric relationship between the dew point and dry-bulb temperatures recorded at R5S8B10. Figure 10 provides a comparison of monthly average values recorded in each test room during the 1990-91 season. For the 1990-91 storage season, average relative humidity levels in the two fan cycled rooms stabilized approximately three percent lower than in the conventionally operated rooms. Relative humidity levels continued to increase for the first four to five months in the non-fan cycled rooms.

Relative humidity was improved in the fan cycled rooms during the 1991-92 season. Relative humidity in Rooms 4 and 5 stabilized at approximately 95%. Room 15 relative humidity plateaued at 93% (Figure 11). Overall average relative humidity, dry-bulb temperature, and dew point temperature at R5S8B10 for each room during each season are listed in Table 3. Average relative humidity in the fan cycled rooms during the second season increased to levels observed in non-fan cycled rooms. Average relative humidity in Room 14 during the second season was low because of the short storage time. Relative humidity in Room 4 dropped slightly during the last two months of storage as a result of ice accumulation on the evaporator coils (discussed later).

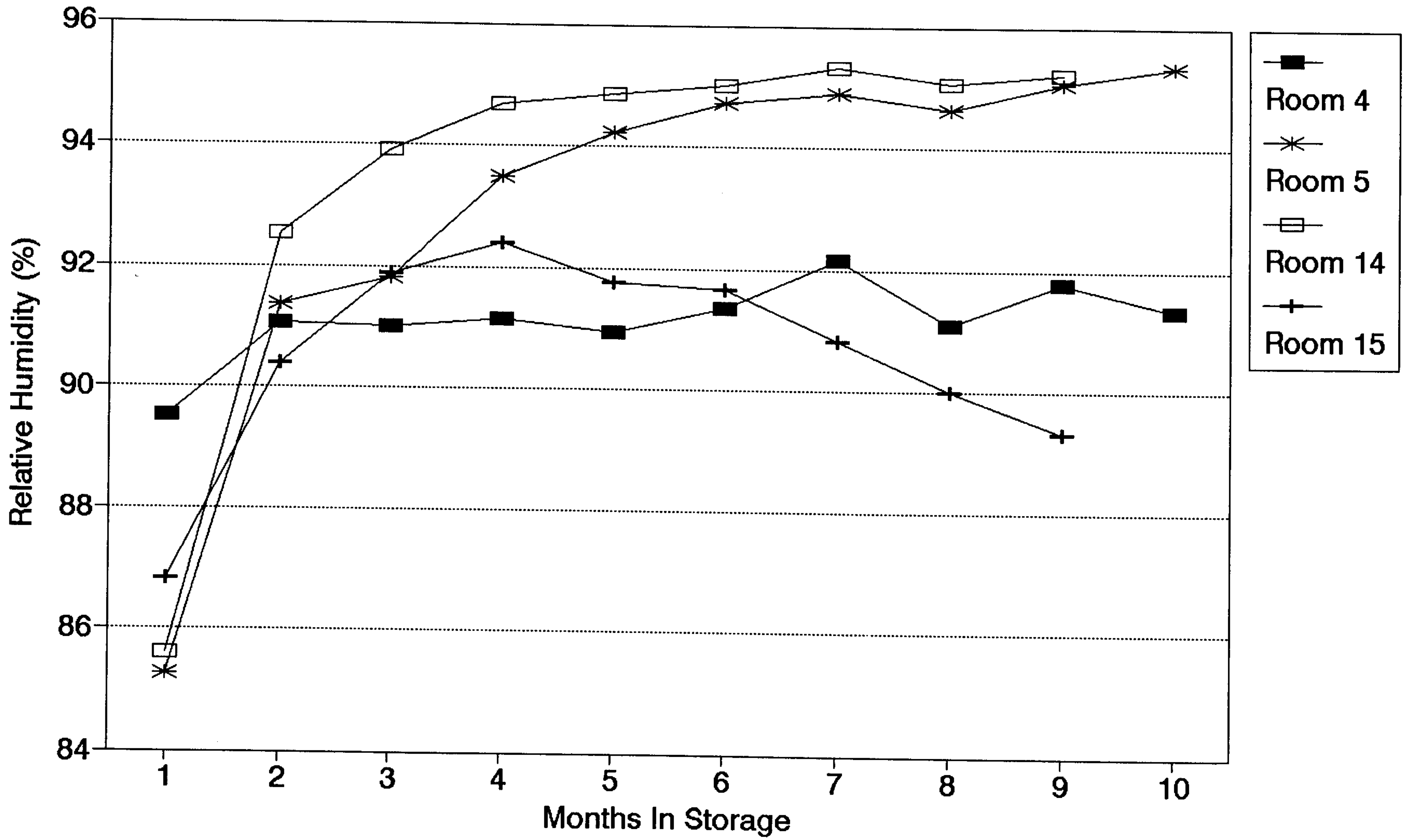


Figure 10. 1990-91 Average Relative Humidity Levels

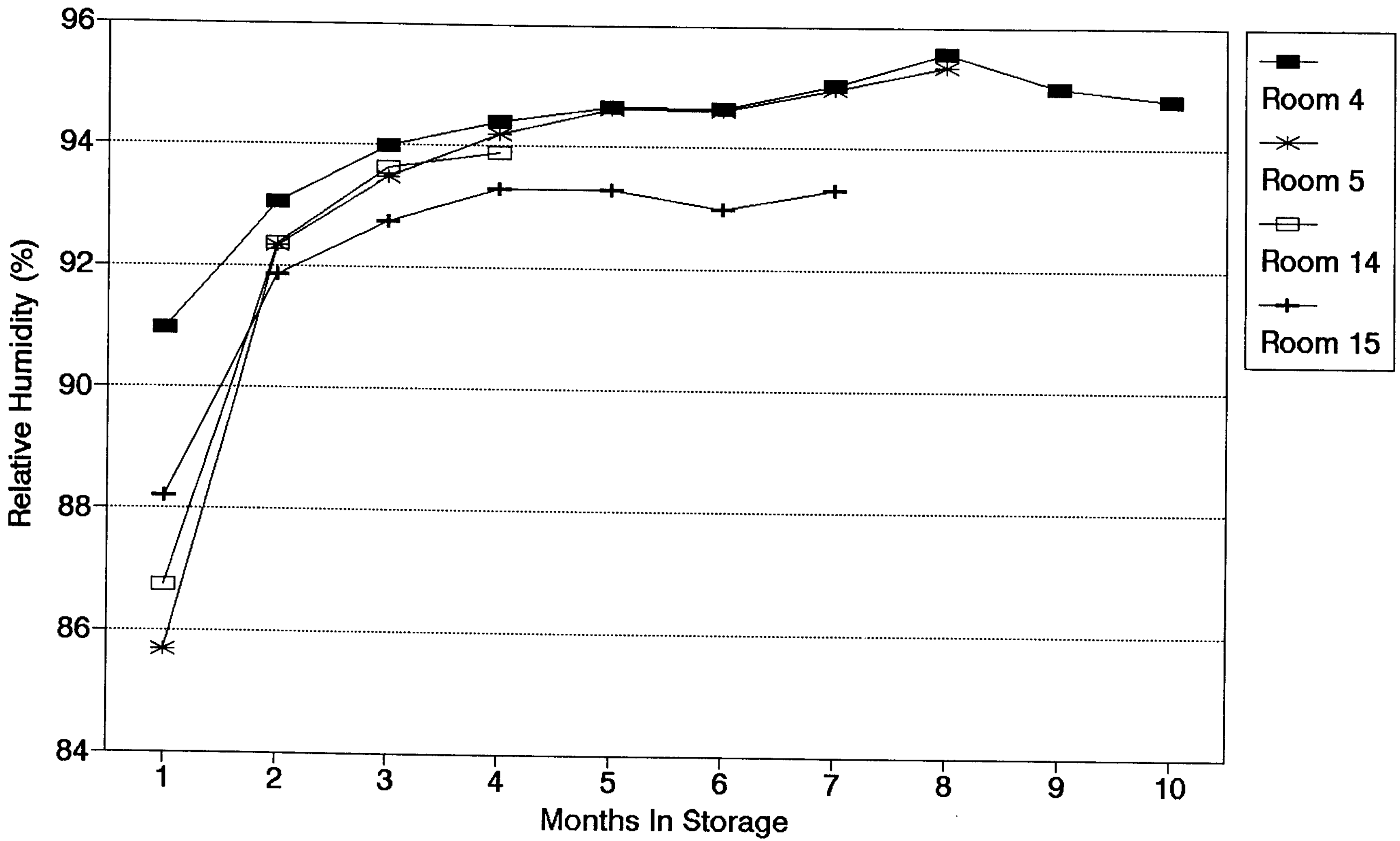


Figure 11. Average Relative Humidity Levels During 1990-91

Table 3. Average Relative Humidities and Temperatures at R5S8B10

	1990-91			1991-92		
	RH (%)	Dry-Bulb Temp (C)	Dew Point Temp (C)	RH (%)	Dry-Bulb Temp (C)	Dew Point Temp (C)
Room 4	91.1	-0.11	-1.23	94.2	0.11	-0.63
Room 5	93.1	-0.19	-1.06	93.3	-0.21	-1.04
Room 14	93.5	-0.18	-0.99	91.2	-0.20	-1.31
Room 15	90.8	0.19	-1.00	92.2	0.13	-0.87

Prior to fan cycling, fluctuations in relative humidity values during any one day were similar in all rooms at approximately 1.5 to 2%. The largest single fluctuation typically occurred during the daily defrost cycle. Relative humidity rose rapidly during the time the fans were off and warm water was circulated over the evaporator coils. When the fans were reactivated and refrigeration commenced, relative humidity rapidly dropped three to six percent. Recovery to the approximate level prior to initiation of the defrost cycle required about 30 minutes after the fans resumed operation. Figures 12 and 13 show five minute interval relative humidity values in Rooms 4 and 5, respectively, for a typical day in March 1991, after fan cycling was initiated. The large fluctuations caused by the fan cycling in Room 4 are readily apparent. Each time the fans cycled off in Room 4, relative humidity levels reacted essentially as described during a defrost cycle. Typical fluctuations caused by the defrost cycle appear at approximately 0230 h in Room 5.

BACK PRESSURE REGULATOR OPERATION AND CONTROL

1990-91 Storage Season

Each cold storage room is equipped with a back pressure regulator (BPR) that controls refrigerant temperature within the evaporator coils in that room (Figure 3). Operation and control of the BPR had significant influence on both relative humidity and energy consumption in the fan cycled rooms. Initially, the BPR was controlled by its own thermostat in the storage room and reacted independent of the fan cycling and defrost operations. The BPR

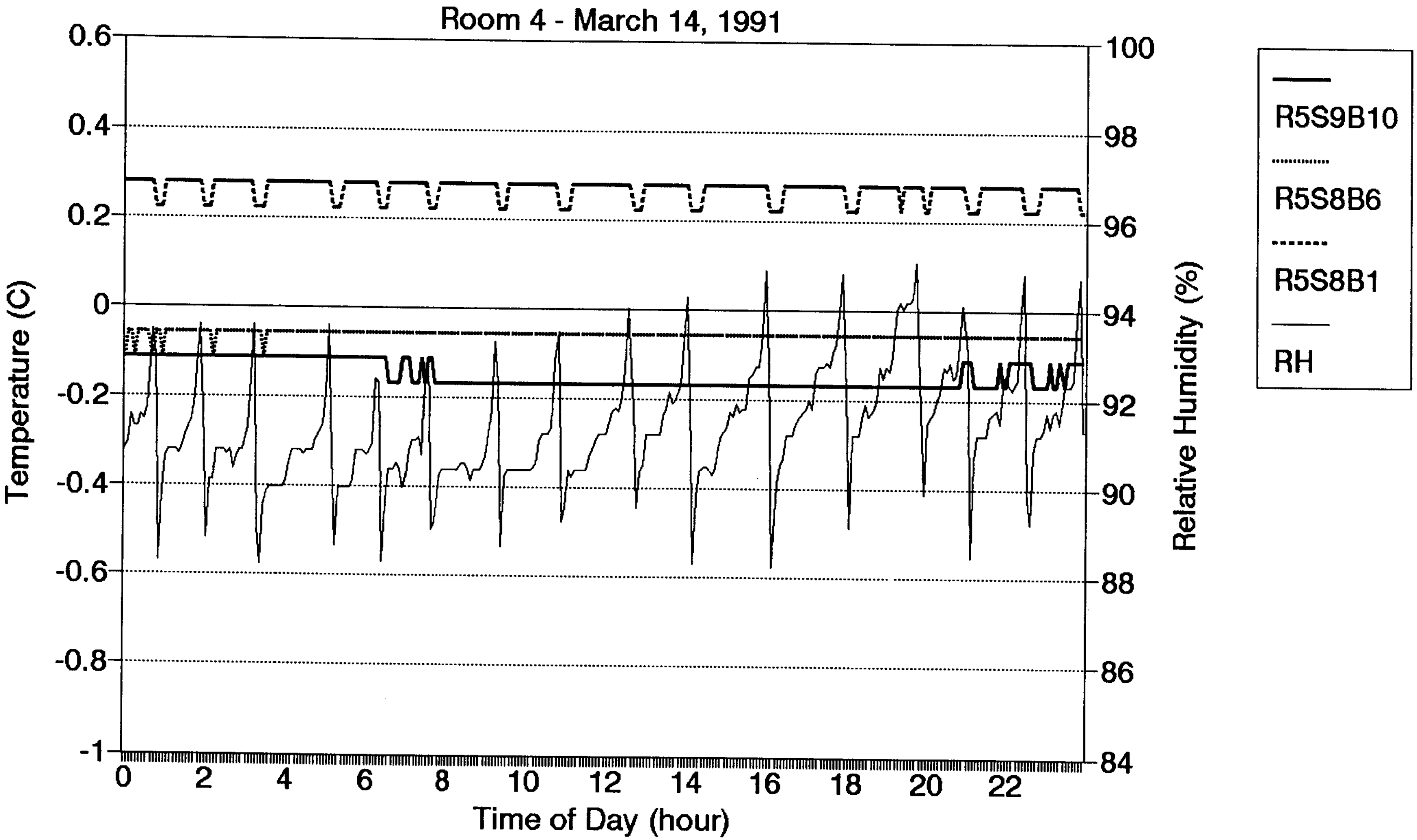


Figure 12. Fruit Bin Temperature and Relative Humidity Variations in a Fan Cycled Room

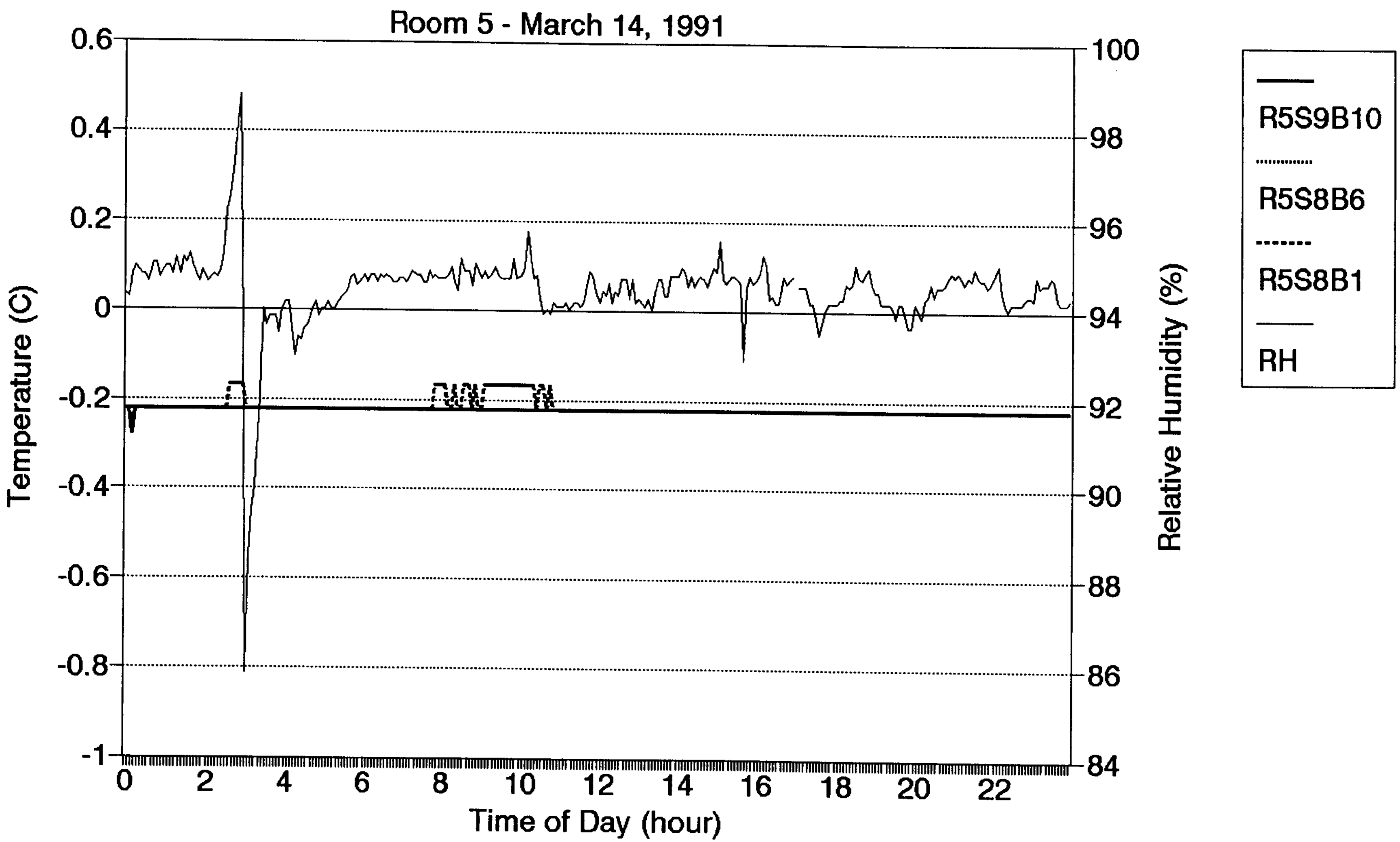


Figure 13. Fruit Bin Temperature and Relative Humidity Variations in a Non-Fan Cycled Room

thermostat was located in the return air stream near the air inlet to the evaporator unit. During each period the evaporator fans cycled off, or during a scheduled defrost cycle, the BPR thermostat sensed an elevated air temperature. This caused the BPR to open more fully, resulting in reduced suction line pressure. Decreased pressure allowed additional refrigerant in the evaporators to change phase causing the heat exchange surface to become excessively cold. Consequently, when the evaporator fans restarted following an off-cycle, room air blown past the coils contacted very cold fin surfaces which rapidly reduced the relative humidity in the storage environment.

On February 8, 1991, a time-delay relay was installed on the BPR to buffer its opening after fan-off periods in Room 4. The system was modified to allow the evaporator fans to operate for a set period of time before energizing the BPR. This mode of operation tended to eliminate the normal temperature stratification induced in the room while the fans were idle and caused the thermostat that controlled opening of the BPR to sense a temperature more representative of actual room conditions. Because the BPR sensed a lower temperature, the suction line pressure was maintained at a higher level and the evaporator coil was not cooled excessively. The time delay was initially set at 30 minutes and subsequently reduced to 10 minutes after determining that the 30 minute fan operation with no cooling caused unwanted heating within the storage room and unnecessary energy use.

In early March, wiring to the BPR was modified to cause the modulating motor to latch in the position it was operating prior to fan cycling and not return to normal operation until the time delay expired. This mode of BPR control also proved to be less than optimum as is illustrated by the frequent fan cycling and low average relative humidity in Figure 12. Although fruit temperature remained stable, low average relative humidity caused increased transpiration and product mass loss.

The rapid drop in relative humidity caused by restarting the refrigeration system following a fan off-cycle reduced room relative humidity below the rooms that did not incorporate fan cycling. This reduced humidity level caused the overall mass loss experienced during the storage season to be greater in the fan cycled rooms. Operation of the BPR directly controls the temperature of the refrigerant in the evaporator coils and thus the temperature of the air that is circulated past the finned surfaces. When suction line pressure decreased, vigorous boiling of the refrigerant in the coils caused the temperature to quickly drop. Analysis of data indicated that the initial setting of the BPR was such that the evaporator coils operated about 1.8°C warmer in Rooms 14 and 15 than in Rooms 4 and 5. However, operation of the BPR during fruit cool down in Room 4 caused the evaporator coil to operate at approximately -2.8°C during the first part of the storage operation in contrast to -9.4 to -6.7°C in the three other rooms. The colder coils had a distinct adverse effect on room relative humidity during the first few days of storage.

<u>1991-92 Storage Season</u>

Prior to the 1991 apple harvest, BPR control in Rooms 4 and 15 was modified such that operation was completely controlled by the computer. Software was added to the computer that caused the BPR to close when the evaporator fans were not in operation and to ramp open through a finite number of steps when the fans turned on. The set point temperature used to control opening of the BPR was provided by three thermistors located in the air at the evaporator inlet, at the evaporator outlet, and above the fruit midway between the evaporator coils and the front wall of the room. A weighted average of 10, 3, and 5, respectively, was used to buffer reaction of the BPR to the temperatures being sensed by each thermistor. This control change dramatically improved both humidity levels and fruit temperature stability in the fan cycled rooms.

ATMOSPHERIC GAS CONCENTRATIONS

A catalytic oxygen burner was utilized during the first two to three days of storage to reduce the oxygen concentration from 21% to approximately 4%. Fruit respiration further reduced the atmospheric oxygen to set point values. Air was manually added to each room as needed to maintain the desired oxygen level throughout the rest of the storage periods. Seasonal average oxygen content values, upon dropping below 4%, were 1.8, 1.6, 2.0, and 2.0% in Rooms 4, 5, 14, and 15, respectively, during the 1990-91 season and 1.6, 1.9, 2.0, and 2.0% during the 1991-92 season (Table 4). Room oxygen concentrations increased toward the end of the storage periods to 3.5 and 2.6% in Rooms 4 and 5, respectively, during the 1990-91 season and to 3.1% in Room 5 during the 1991-92 season. These gradual increases occurred due to air leakage into the storage rooms. Thermal expansion and contraction of the south facing walls and

Table 4. Average Gas Concentrations in Percent

	1990-91		1991-92	
	Oxygen	Carbon Dioxide	Oxygen	Carbon Dioxide
Room 4	1.8	0.5	1.6	0.6
Room 5	1.6	0.5	1.9	0.7
Room 14	2.0	0.5	2.0	0.7
Room 15	2.0	0.8	2.0	1.0

doors of these rooms leads to loss of an airtight door seal. Bottled nitrogen gas was injected into rooms when leakage affected desired oxygen levels. Target oxygen levels were 1.5% in all rooms during the 1990-91 season and in Rooms 4 and 5 during the 1991-92 season. Target oxygen concentration levels in Rooms 14 and 15 were 2.0% during the 1991-92 season.

Carbon dioxide was controlled with dry hydrated lime. Perforated bags of lime were placed on pallets and stacked on top of selected bins of fruit. Carbon dioxide concentration levels were to be kept below 1.5%. Seasonal average carbon dioxide values averaged 0.5, 0.5, 0.5, and 0.8% upon pull-down in Rooms 4, 5, 14, and 15, respectively, during the 1990-91 season and 0.6, 0.7, 0.7, and 1.0% during the 1991-92 season (Table 4). Carbon dioxide concentrations slowly increased during the storage period.

Figures 14 and 15 illustrate oxygen and carbon dioxide concentration levels at approximately 12 hour intervals for Rooms 4 and 5, respectively, during the 1990-91 season. The frequency at which air was added to the storage room is also indicated. Gas concentration levels depended on set point levels, air leakage, and frequency and duration of times when air was added to the storage room. Evaporator fan operation had no effect on gas concentration levels. Gas concentration profiles for other rooms during both seasons were similar to those shown in Figures 14 and 15.

Atmospheric gas samples were periodically drawn from three different heights at room location R5S8 (approximately Bin 1, Bin 6, and Bin 10) in all four rooms. Gas concentration levels were

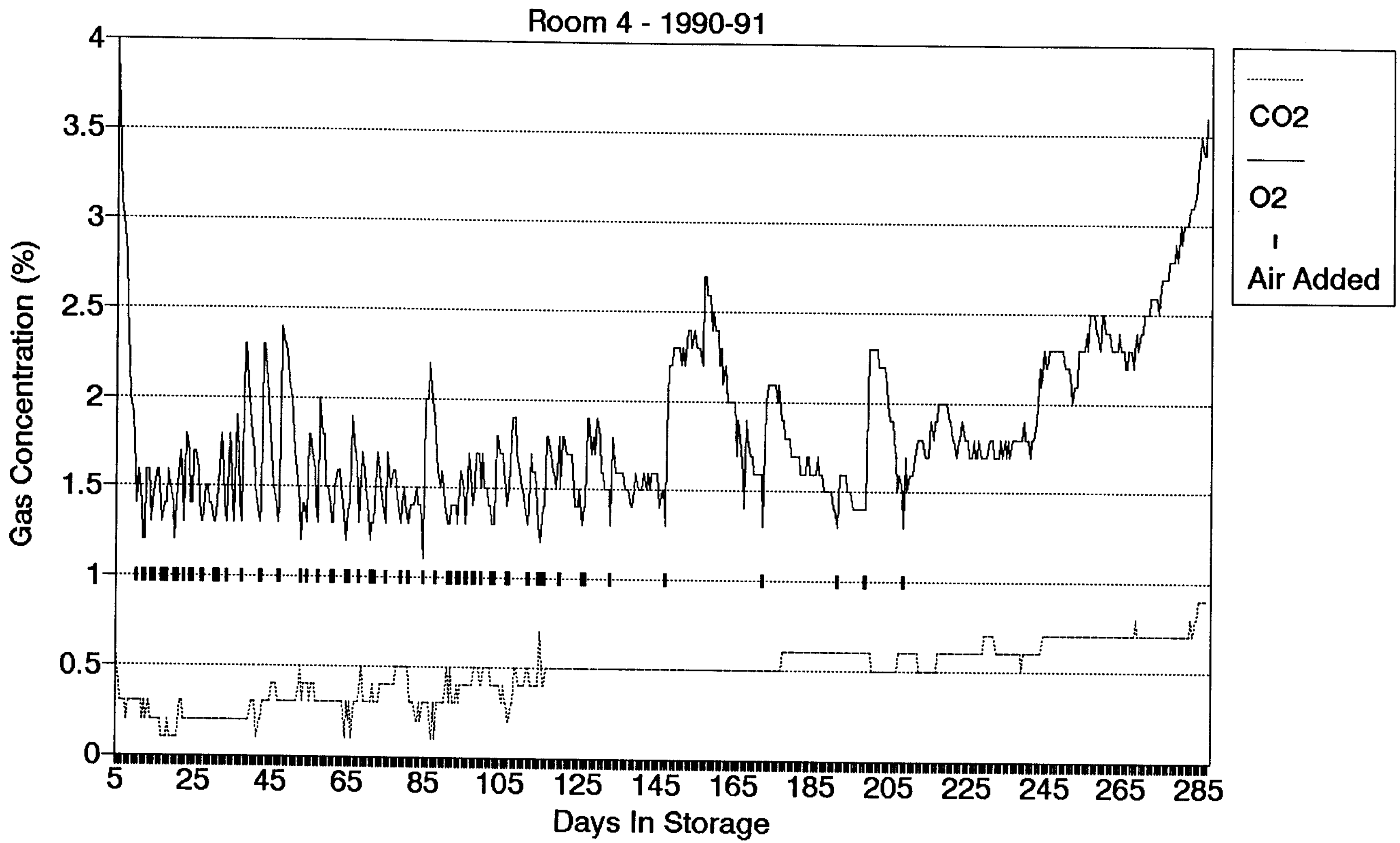


Figure 14. Gas Concentration Levels in Room 4

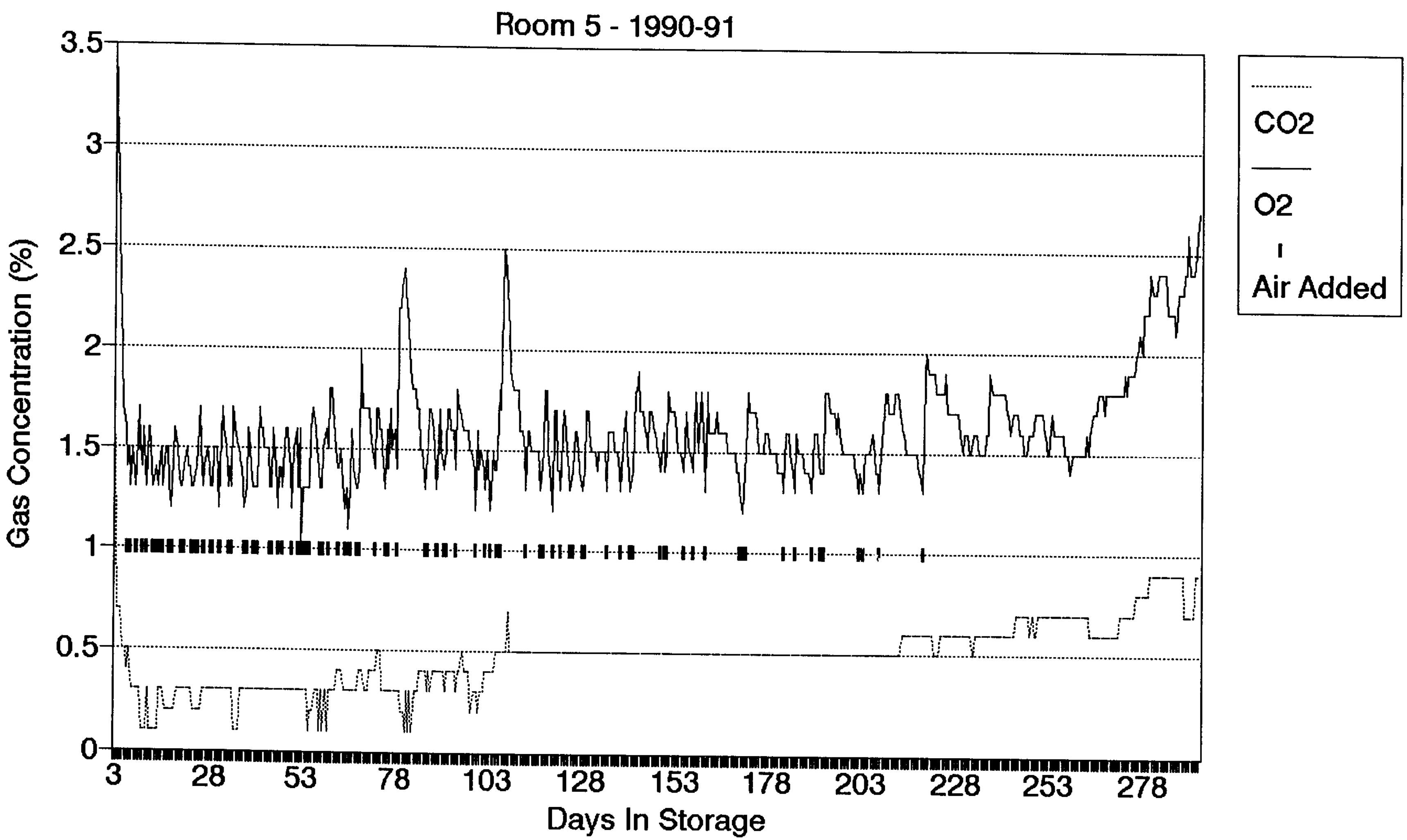


Figure 15. Gas Concentration Levels in Room 5

measured after the evaporator fans had been off for a period of six to eight hours. No evidence of gas stratification was observed.

TEMPERATURE HISTORIES WITHIN BINS

Analysis of the temperature probes throughout the storage rooms provided information as to overall system thermal performance. The desired fruit temperature in all rooms during the 1990-91 season and in Rooms 4 and 5 during the 1991-92 season was 0°C ±0.3°C. Target fruit temperature in Rooms 14 and 15 was -0.3°C ±0.3°C during the 1991-92 season.

Temperatures within the apple mass fluctuated very little within the computer controlled, fan cycled rooms. For example, Figures 12 and 13 present temperature histories at five minute intervals in one stack of bins for one day in Rooms 4 and 5, respectively, during the 1990-91 season. The slight variation in recorded temperatures indicates that fan cycling did not adversely affect fruit temperatures within a bin.

Figures 16 through 27 document four hour average temperatures recorded in Bins 1, 6, and 10, and in Stacks 1, 8, and 13 for Rooms 4 and 5 during January 1992. Temperature data recorded in Rows 1, 5, and 9 were averaged together in Figures 16-27. Figures 16, 17, and 18 show the average temperature at Stack 1, 8, and 13 for Bins 1, 6, and 10, respectively, in Room 4 (horizontal temperature profiles). Figures 19, 20, and 21 show the average temperature at Bin 1, 6, and 10 for Stacks 1, 8, and 13, respectively, in Room 4 (vertical temperature profiles). Identical parameters are graphed in Figures 22-27 for Room 5.

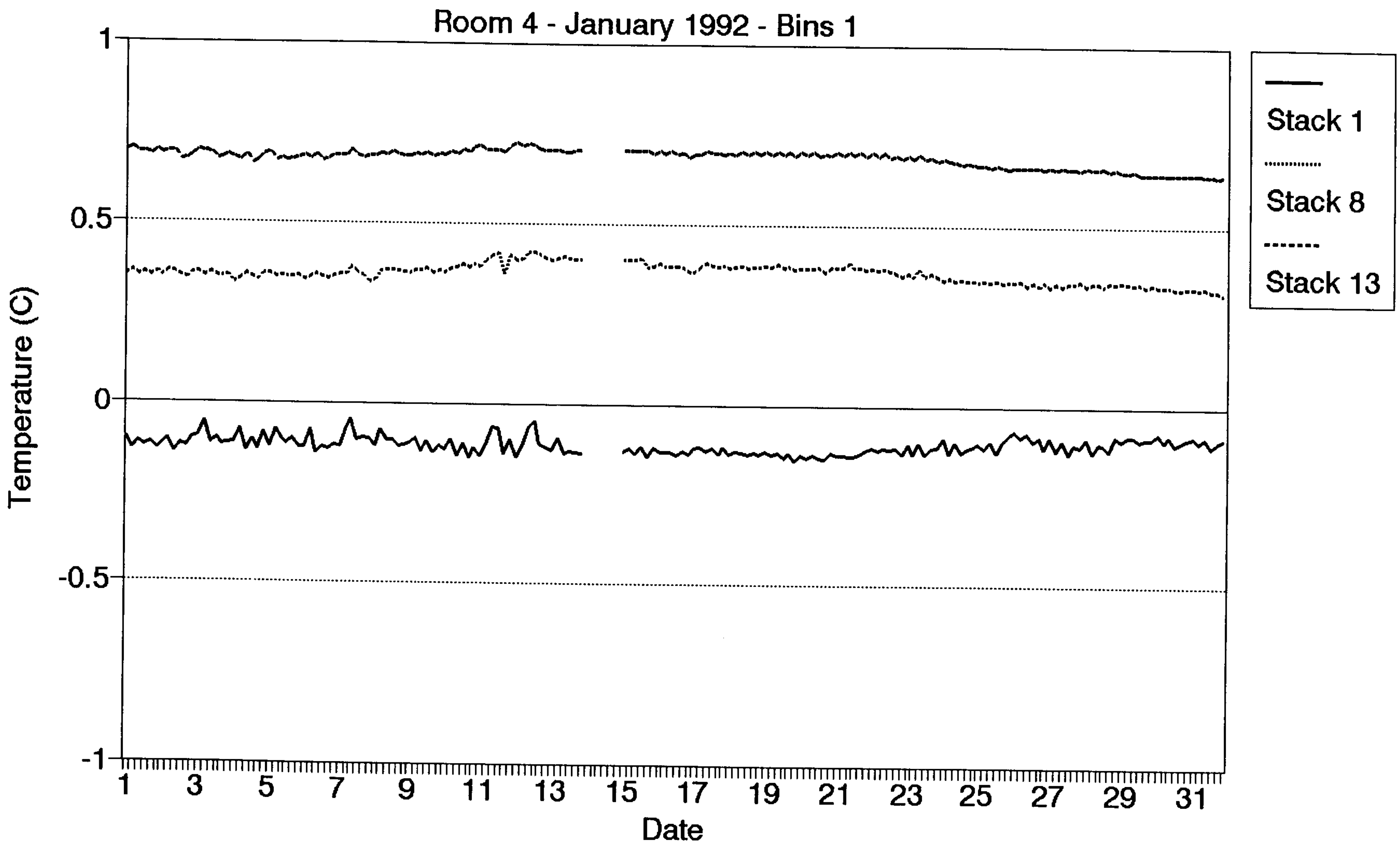


Figure 16. Horizontal Profile of Average Temperatures in Room 4 - Bins 1

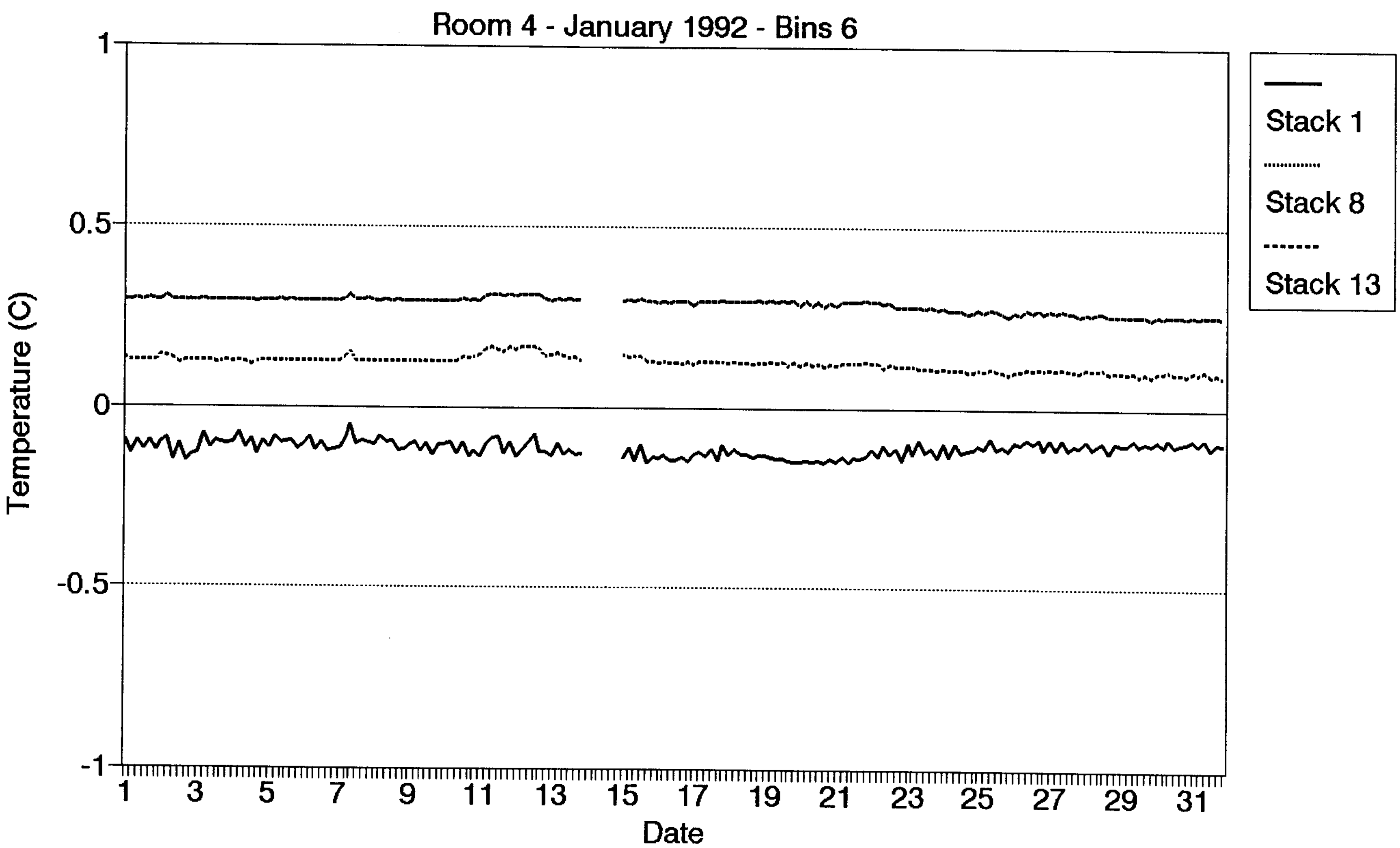


Figure 17. Horizontal Profile of Average Temperatures in Room 4 - Bins 6

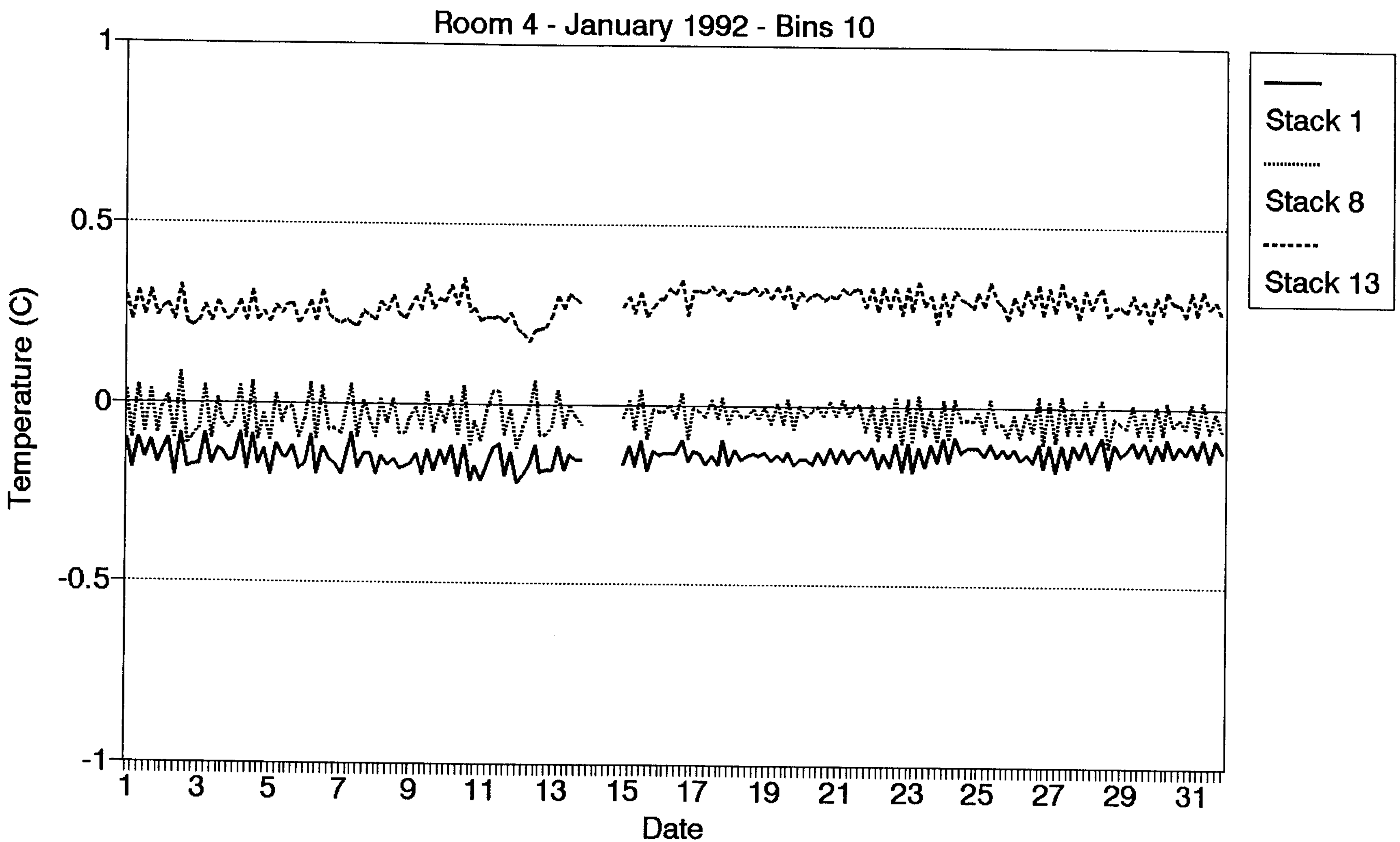


Figure 18. Horizontal Profile of Average Temperatures in Room 4 - Bins 10

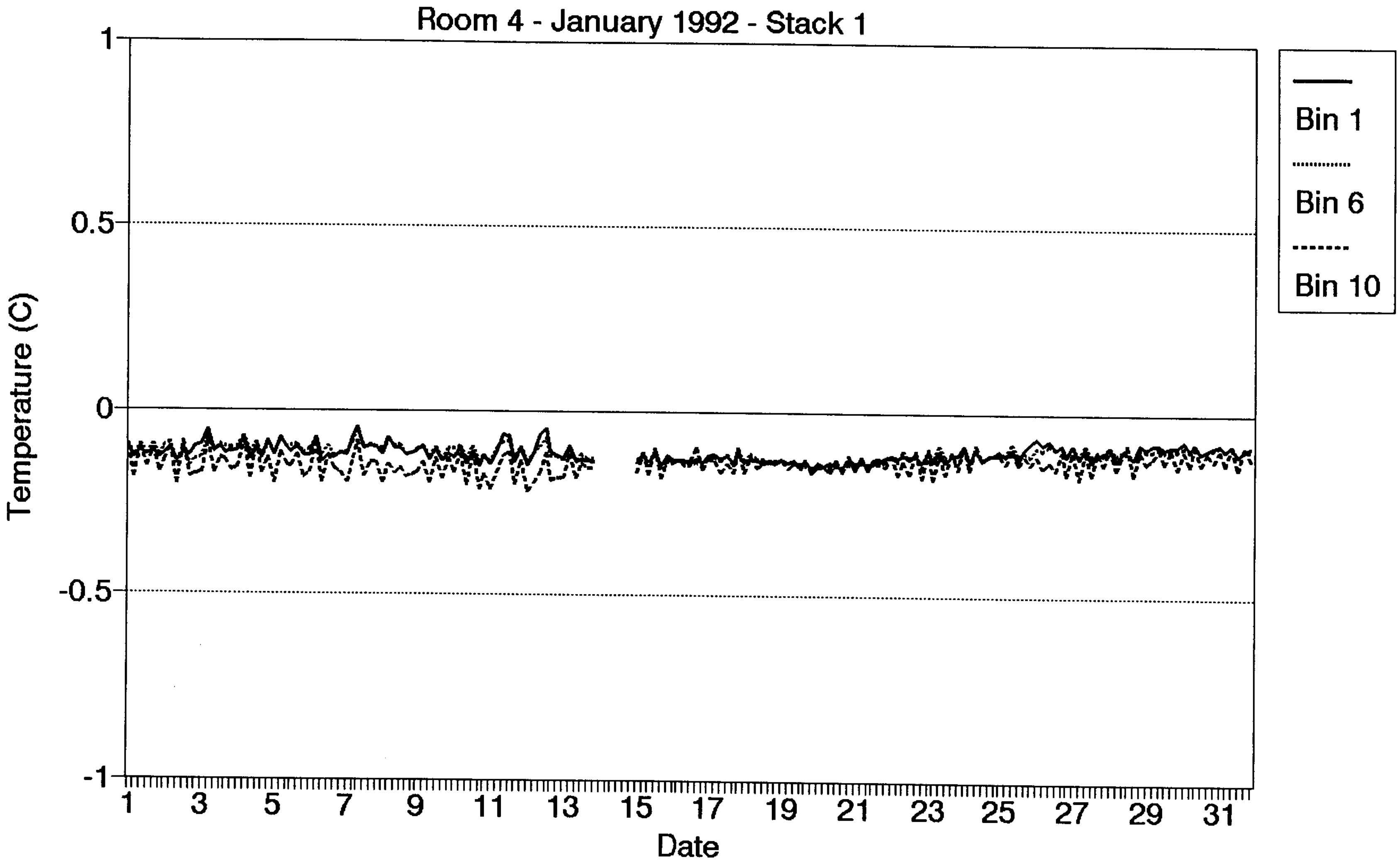


Figure 19. Vertical Profile of Average Temperatures in Room 4 - Stack 1

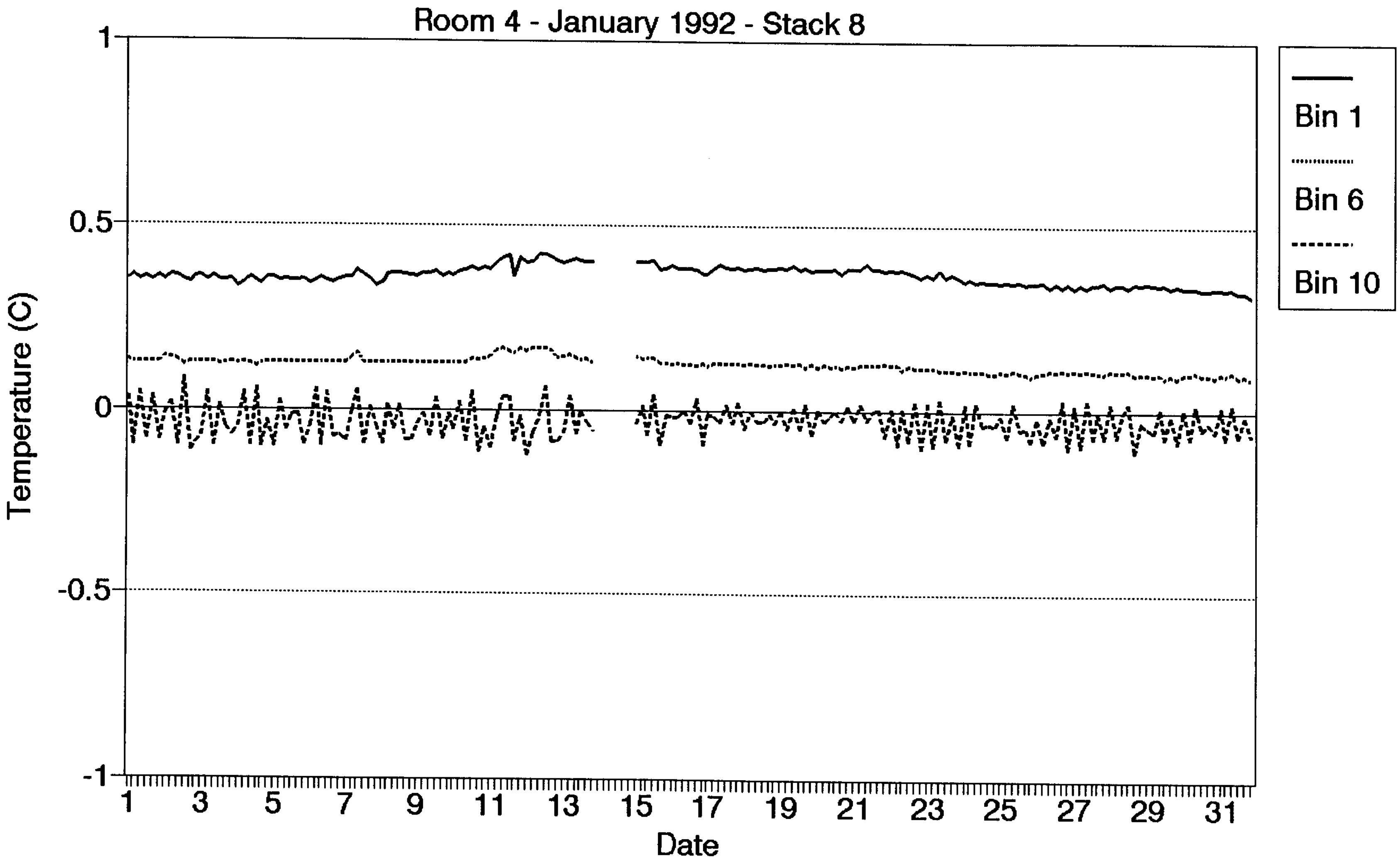


Figure 20. Vertical Profile of Average Temperatures in Room 4 - Stack 8

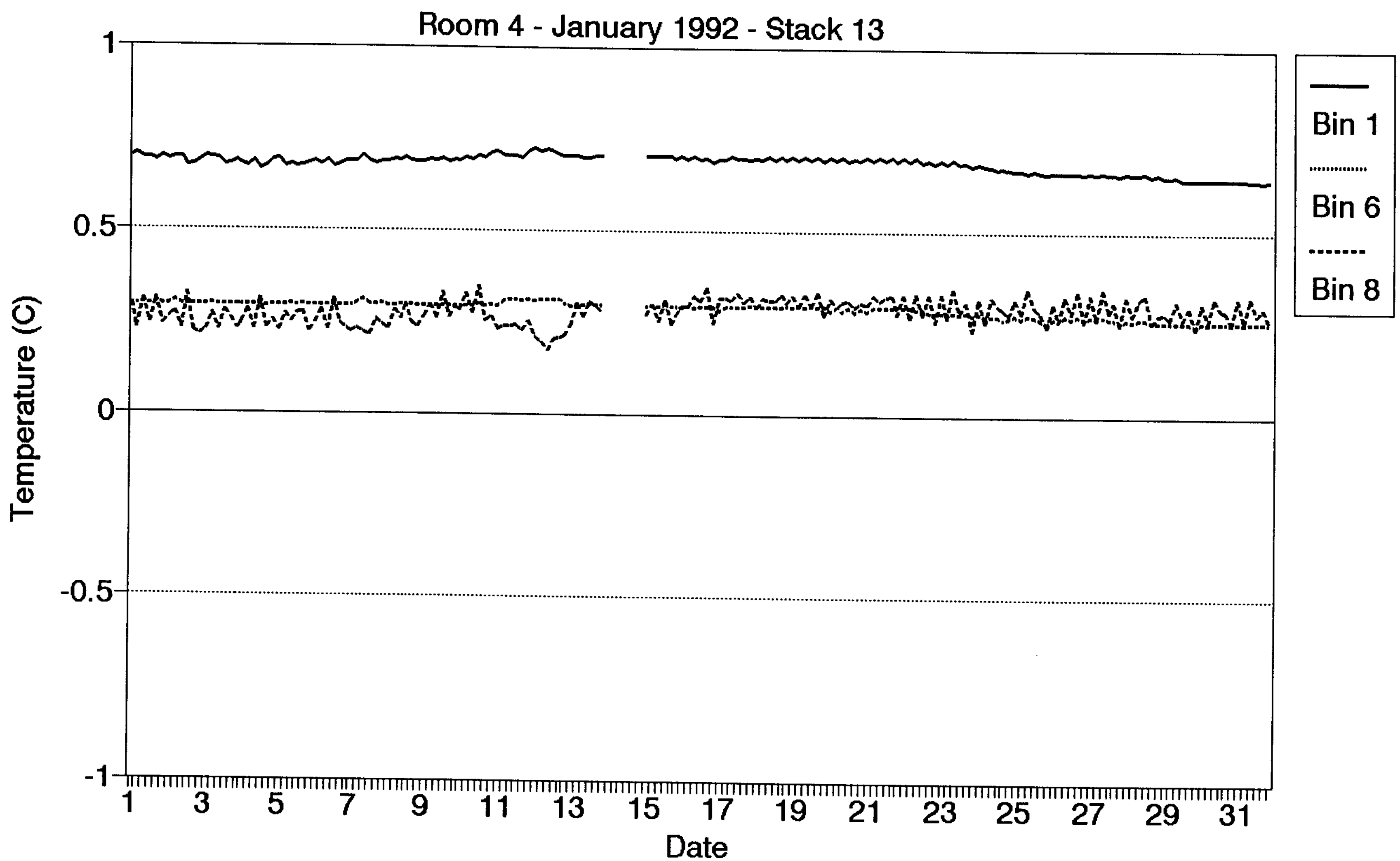


Figure 21. Vertical Profile of Average Temperatures in Room 4 - Stack 13

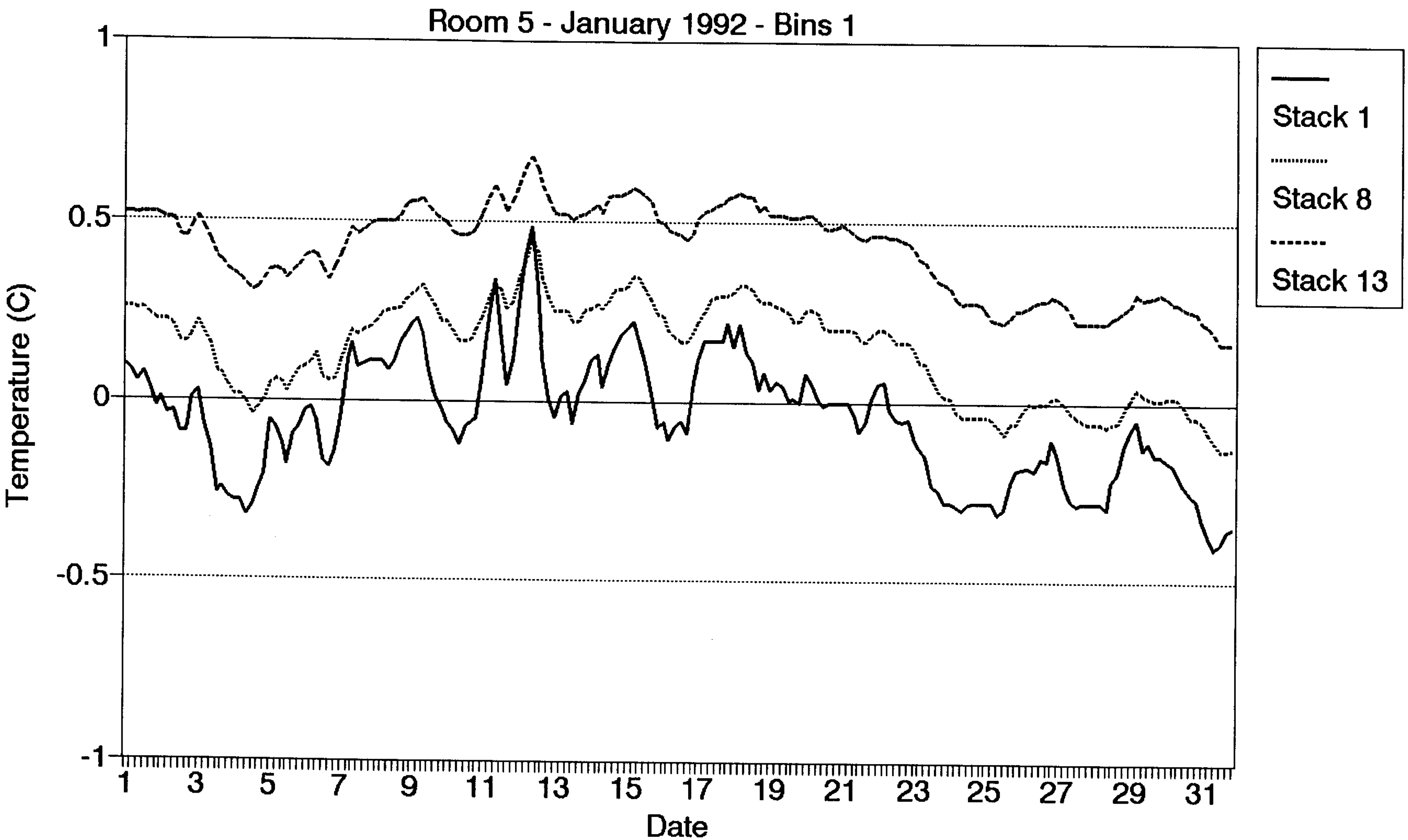


Figure 22. Horizontal Profiles of Average Temperatures in Room 5 - Bins 1

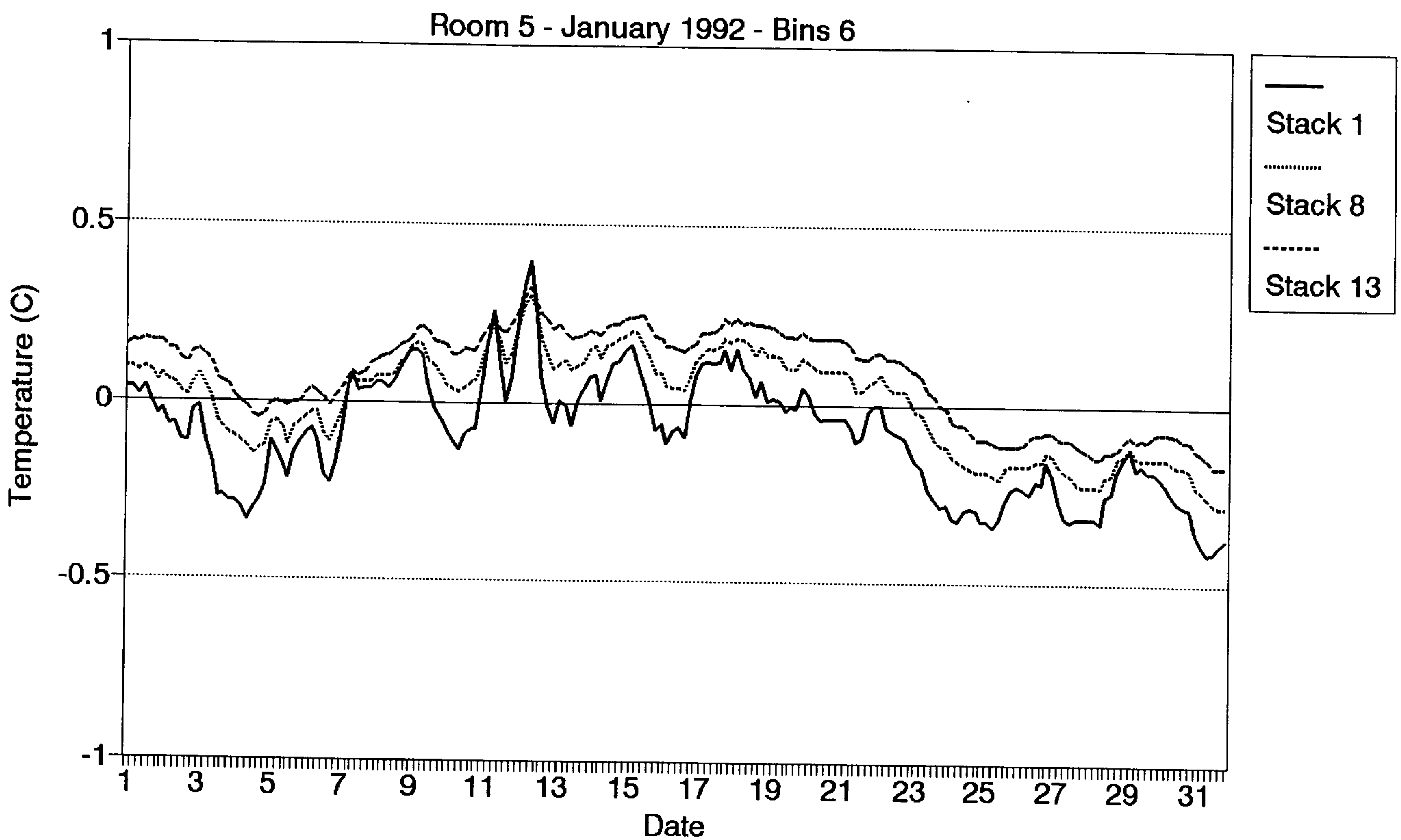


Figure 23. Horizontal Profiles of Average Temperatures in Room 5 - Bins 6

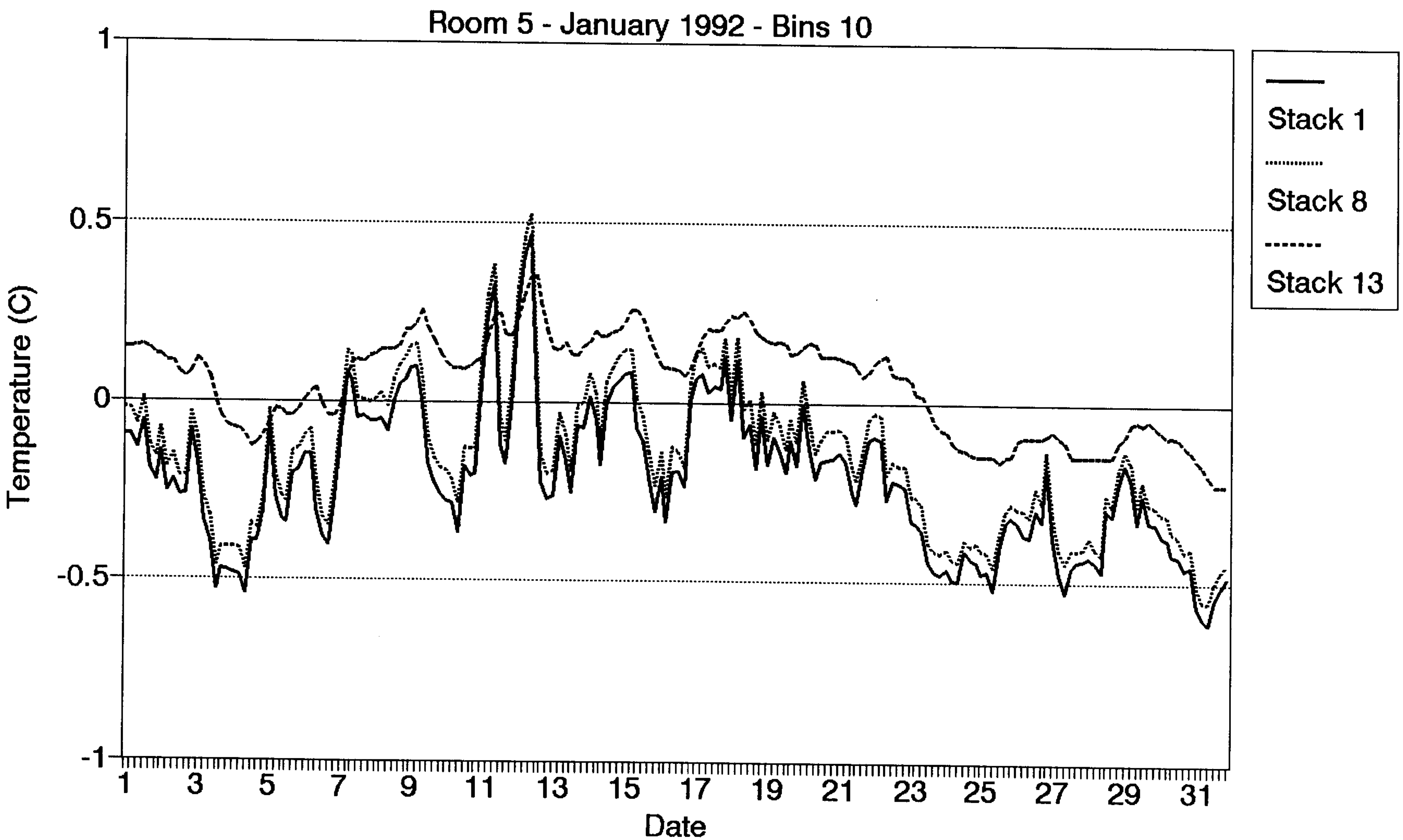


Figure 24. Horizontal Profiles of Average Temperatures in Room 5 - Bins 10

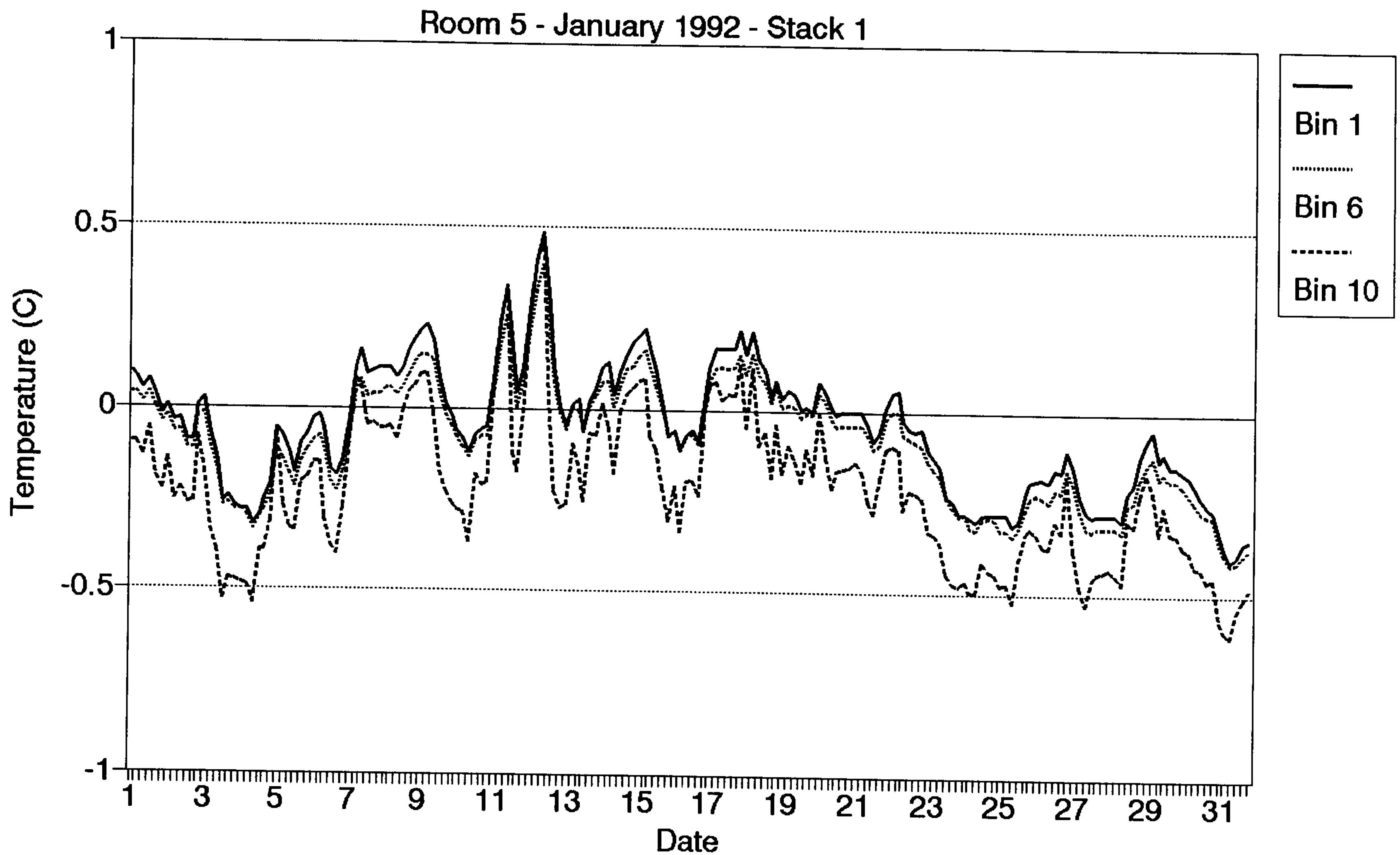


Figure 25. Vertical Profiles of Average Temperatures in Room 5 - Stack 1

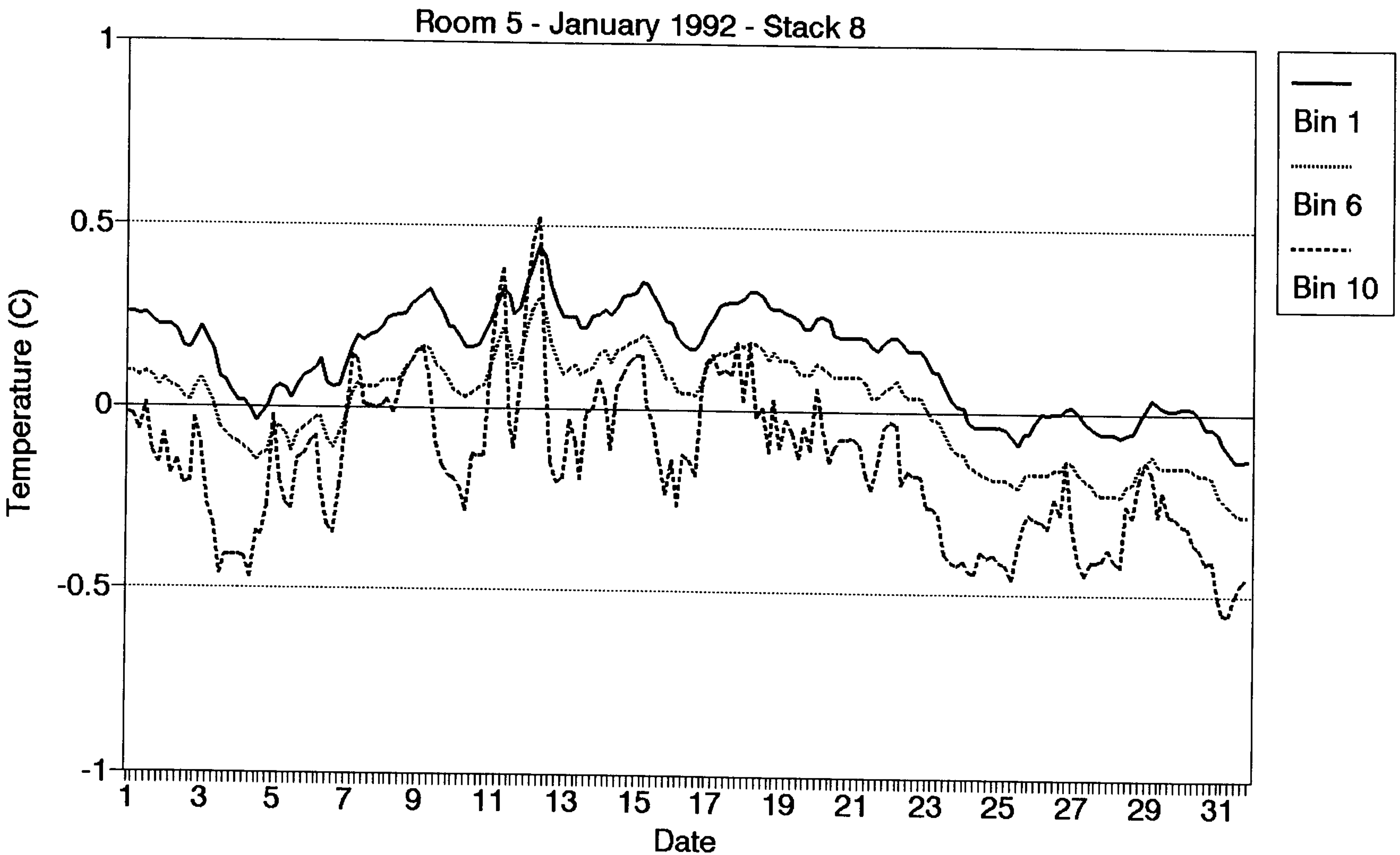


Figure 26. Vertical Profiles of Average Temperatures in Room 5 - Stack 8

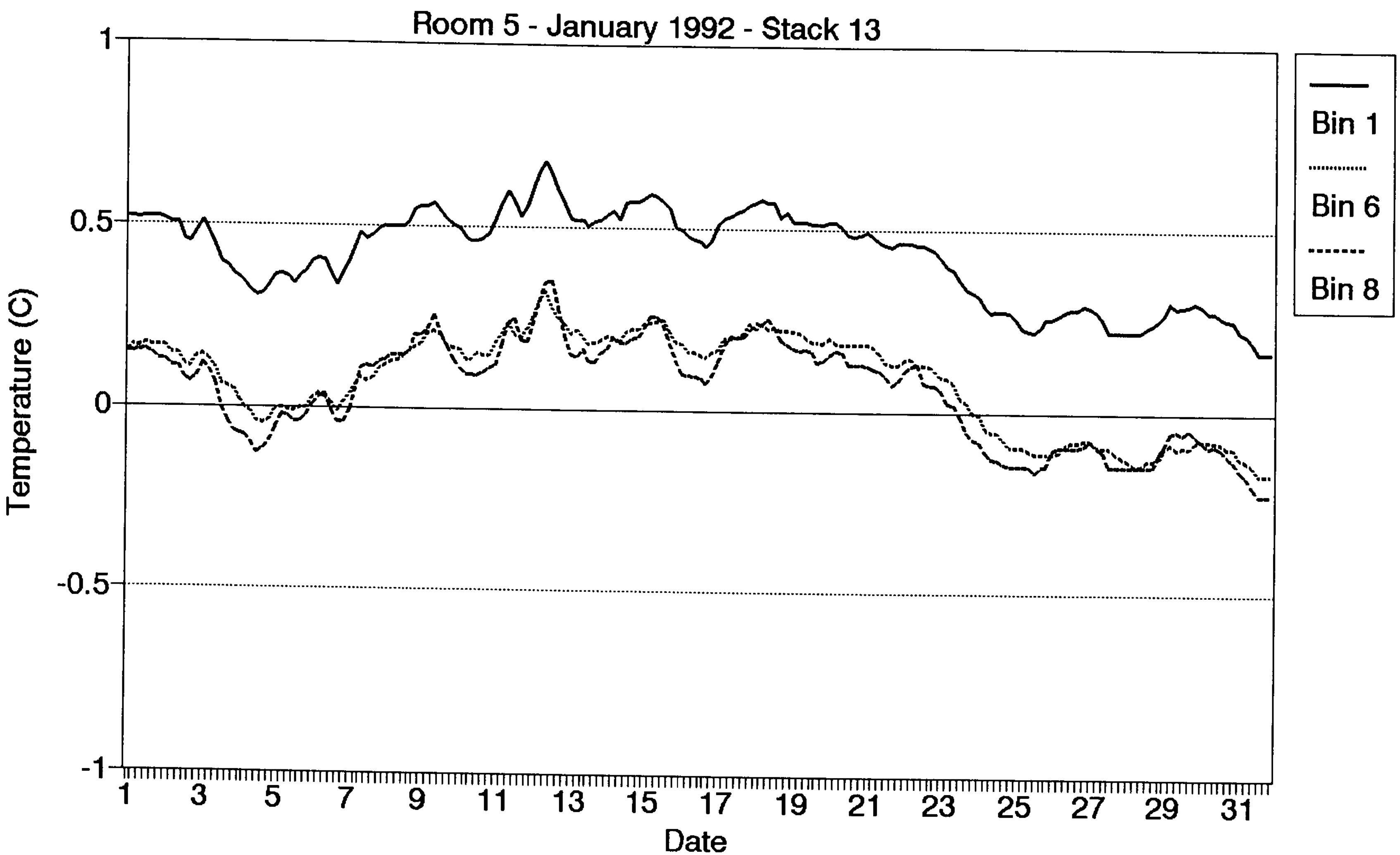


Figure 27. Vertical Profiles of Average Temperatures in Room 5 - Stack 13

Averaged temperatures recorded at all three locations in Stack 13 (Bins 1, 6, and 10) were consistently higher than respective temperatures recorded in Stacks 1 and 8 in both fan cycled and non-fan cycled rooms. Also, averaged temperatures recorded at all three locations in Stack 8 were consistently higher than temperatures recorded in Stack 1 (Figures 16-18 and 22-24). This was expected due to the air circulation pattern, bin stacking pattern, and room geometry. Cold air leaving the evaporator coils traveled across the tops of the bins to the front of the room, moved downward into the stacks, and returned at the rear of the room under the cooling units. Bins on the floor (Bins 1) nearly always had the highest temperature and bins on the top (Bins 10) nearly always had the lowest temperature in all four test rooms (Figures 19-21 and 25-27). Reduced air movement, especially during evaporator fan off-periods, caused this temperature stratification to develop. This unnatural temperature stratification was also partially due to heat transferred through the floor of the storage.

Minimum temperature variation between stacks occurred at the middle layer of fruit or Bins 6 (Figures 17 and 23). Maximum temperature variation between stacks occurred at the bottom layer of fruit (Bins 1) where airflow is low (Figures 16 and 22). Minimum temperature variation between bins occurred at Stack 1 in both rooms (Figures 19 and 25). Thus, airflow at the front of the room is adequate. Maximum temperature variation between bins occurred at Stack 13 in both rooms (Figures 21 and 27). Similar temperature histories were observed in Rooms 14 and 15, and during other months.

Further analysis of Figures 16-27 illustrates the difference in temperature stability in a room in which the BPR was controlled by a computer (Room 4) rather than by a thermostat (Room 5). Four hour average bin temperatures recorded in Room 4 fluctuated less than ±0.1°C during January 1992 (Figures 16-21). Four hour average bin temperatures recorded in the control room (Room 5) fluctuated ±0.5°C during the same period (Figures 22-27). The exceptionally uniform temperatures achieved in Room 4 as compared to Room 5 illustrate the need for and benefit of computer control of the BPR.

AIR TEMPERATURE HISTORIES

Air temperatures above the fruit bins fluctuated more than temperatures recorded within the fruit mass, as was expected. Air, having much less mass to dampen temperature fluctuations than the fruit, more rapidly reflected changes in operation of the refrigeration system. Figures 28 and 29 show air temperatures recorded at the evaporator coil inlet and outlet in Rooms 4 and 5, respectively, during January 1992. Data shown are four hour averages of five minute interval recordings. The spikes on Figure 28 correspond to the times when the evaporator fans were on and off. When the fans were on a difference of 0.3°C or less was observed between the inlet and outlet. Inlet and outlet temperatures were equal when the evaporator fans were off. The spikes on Figure 29 correspond to the daily defrost cycle. The evaporator inlet temperature was generally 0.2 to 0.6°C warmer than the evaporator outlet temperature.

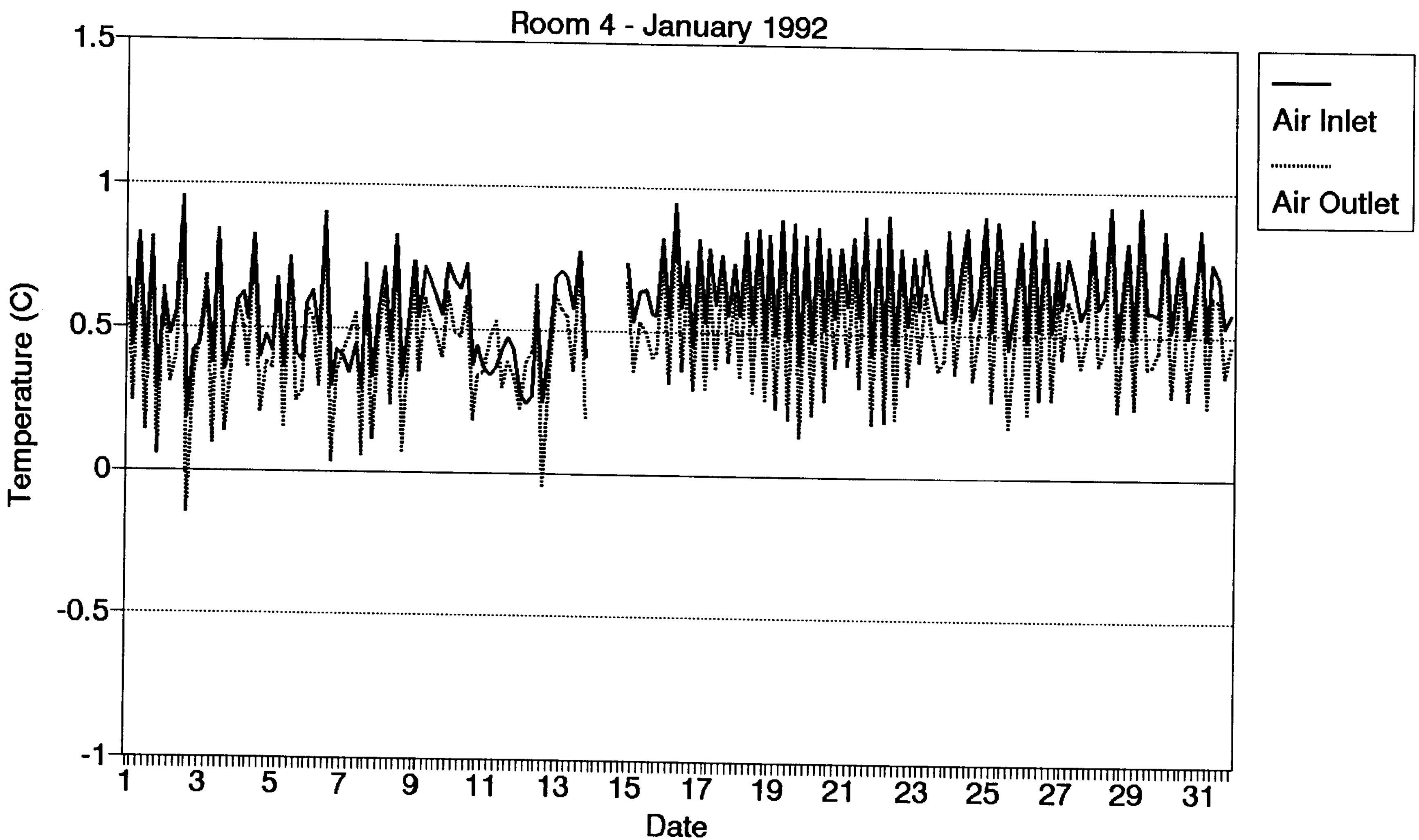


Figure 28. Evaporator Coil Air Temperature Histories in Room 4

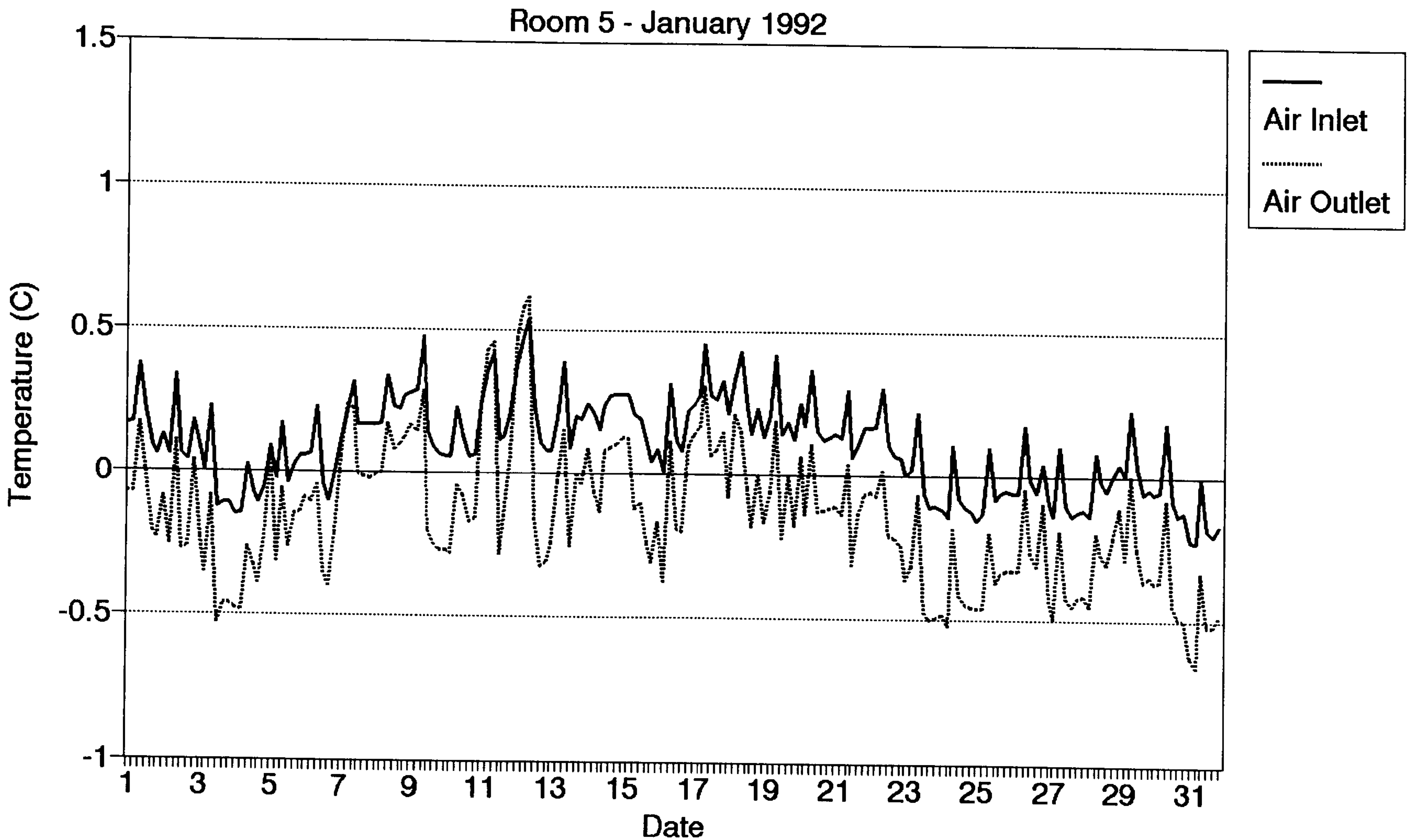


Figure 29. Evaporator Coil Air Temperature Histories in Room 5

Figures 30 and 31 show air temperatures recorded at 0.6 and 7.6 m below the ceiling at approximately the same location in Rooms 4 and 5, respectively, during January 1992. In the fan cycled room air temperature measured 0.6 m below the ceiling fluctuated more, and was slightly warmer than air measured 7.6 m below the ceiling (Figure 30). Air temperatures in Room 5 were approximately equal since evaporator fans operated continuously (Figure 31). Small spikes on Figure 31 correspond to the daily defrost cycle. Larger temperature fluctuations in Room 5 are a result of typical variation in the refrigerant temperature control method.

MASS LOSS AND MASS LOSS RATES

The total number of days apples were stored in each room varied; however, excellent overall mass loss values were achieved in each room. Actual mass loss during the 1990-91 storage season of the 20 numbered apples were 2.90, 2.32, 1.79, and 2.58% in Rooms 4, 5, 14, and 15, respectively (Table 5). All measured mass loss values were lower than industry expectations of 3-5%. Table 6 lists total storage time, initial mass, and mass loss rate data for each storage room. Mass loss rate accounts for differences in storage time and fruit mass analyzed. However, since a disproportionate amount of mass is lost during the cool down period, mass loss rate values for short term storage situations are larger and should not be compared to long term storage data.

Fruit mass loss values were dramatically improved in the fan cycled rooms and slightly improved in the control rooms during the 1991-92 storage season. Mass loss values for the fruit samples

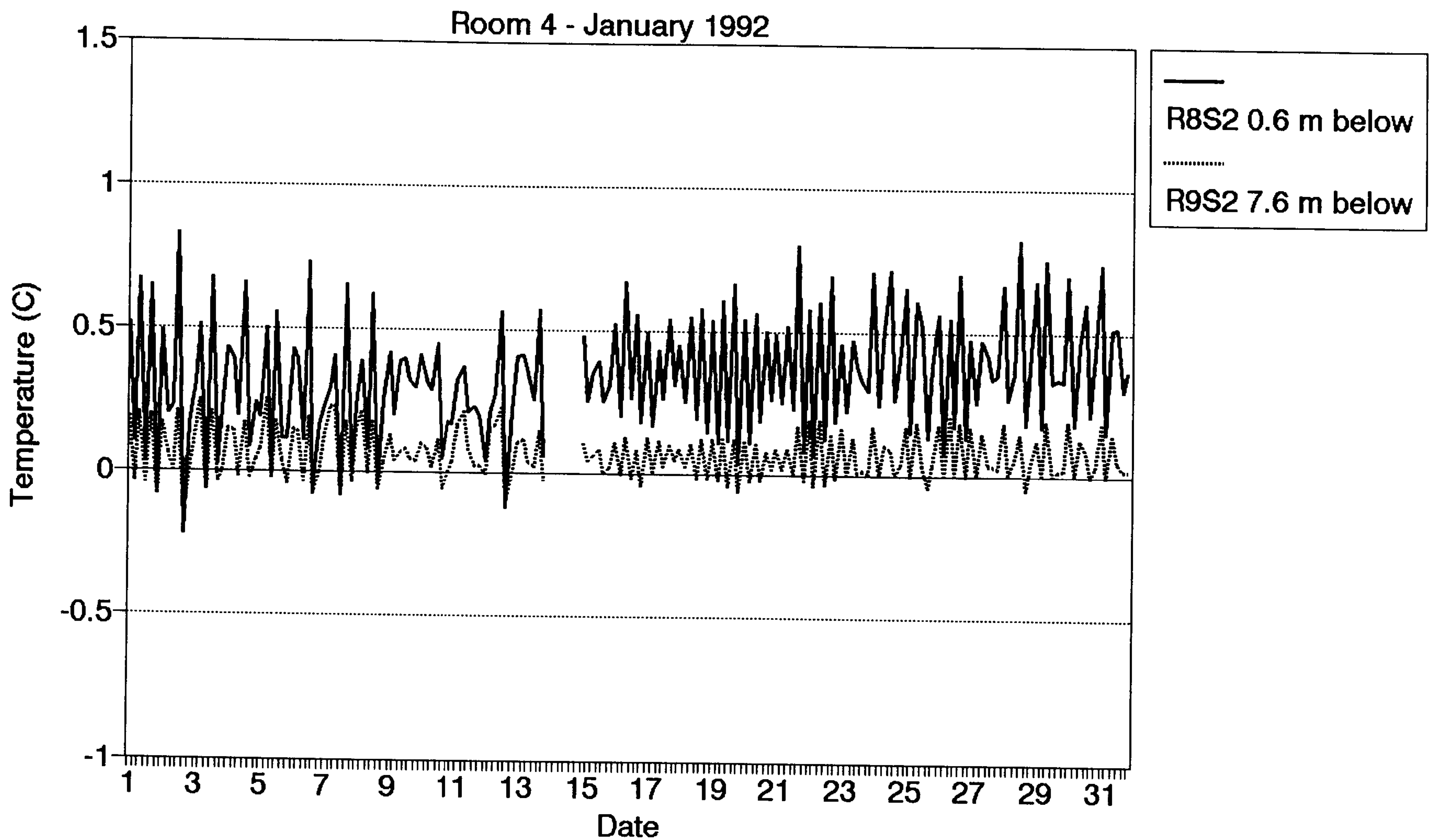


Figure 30. Room Air Temperature Histories in Room 4

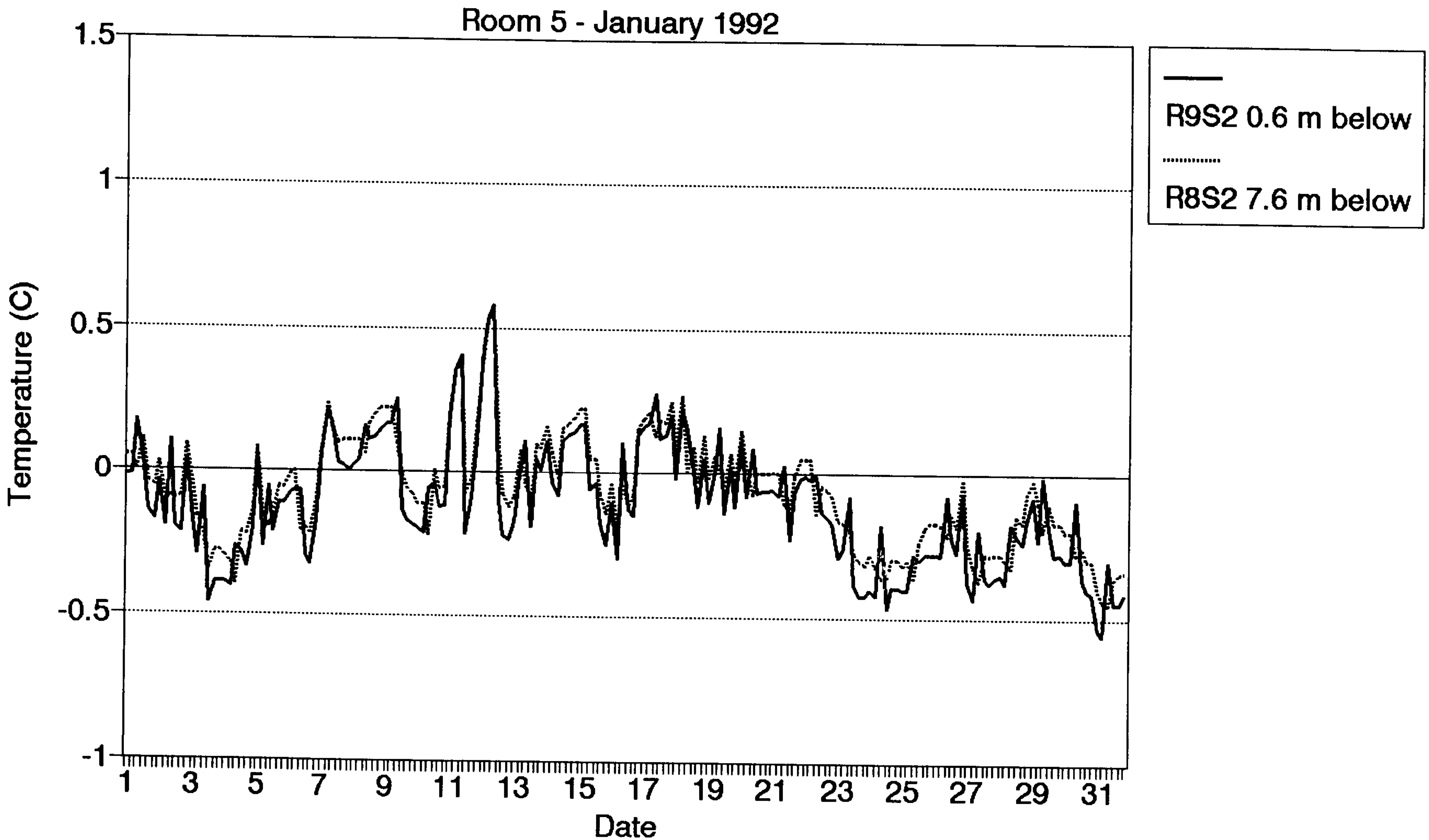


Figure 31. Room Air Temperature Histories in Room 5

Table 5. Product Mass Loss in Percent

	1990-91	1991-92		
	Wood Bin	Plastic Box	Wood Bin	Difference (95% c.i.)
Room 4	2.90	2.07	2.41	0.34 +/-0.15
Room 5	2.32	1.61	1.81	0.20 +/-0.12
Room 14	1.79	1.32	1.55	0.23 +/-0.11
Room 15	2.58	1.32	1.59	0.27 +/-0.12

Table 6. Storage Times and Mass Loss Rates

	1990-91			1991-92		
	Storage Time (days)	Initial Mass (kg)	Mass Loss Rate (mg/kg·hr)	Storage Time (days)	Initial Mass (kg)	Mass Loss Rate (mg/kg·hr)
Room 4	288	3.4464	4.19	300	14.6843	2.87
Room 5	294	3.4293	3.25	278	15.5527	2.42
Room 14	250	3.4349	2.95	107	15.4865	5.13
Room 15	255	3.9172	4.23	201	16.7321	2.73

placed in plastic boxes in Rooms 4, 5, 14, and 15 were 2.07, 1.61, 1.32, and 1.32%, respectively (Table 5). Table 6 shows total storage time, initial mass, and mass loss rate of fruit in the plastic box for each room. Mass loss rate in Room 14 appears high; however, only 1.32% of the initial mass was lost during a relatively short storage period. Since product mass loss is non-linear with storage duration, comparison of mass loss rates can be misleading.

Fruit placed in contact with the wooden bin during the 1991-92 study lost significantly more mass than those in the plastic box. The maximum difference between percent mass loss of fruit in contact with the wooden bin and those in the plastic box using a 95% confidence interval was 0.34 ±0.12, 0.20 ±0.12, 0.23 ±0.11, and 0.27% ±0.12% in Rooms 4, 5, 14, and 15, respectively (Table 5). Increased mass loss was expected as the wooden bins were stored outdoors during the growing season. The wooden bins absorbed moisture transpired from the stored product and from a postharvest drench until the moisture content of the wood reached an equilibrium with the storage environment. Waelti (1989) reported that wooden bins each absorbed 6 kg of water and Kupferman (1991) stated that wooden bins can gain 7-9 kg of water during the storage period.

LOAD CELL DATA

Technical difficulties precluded continuous measurement of fruit mass by the load cell systems during the 1990-91 storage season. Load cell drift during the second season prevented accurate comparison of total mass loss measured by the load cells to total mass loss determined from initial and final fruit weights. Comparison of load cell total mass loss values to values measured with the balance indicated the load cells underpredicted mass loss as measured with the balance in Rooms 4, 14, and 15 by 19, 104, and 182%, respectively, and overpredicted mass loss in Room 5 by 43%. Sustained periods of high room relative humidity levels contributed to the load cell measurement error. Percent mass loss and mass loss rate values presented in Tables 5 and 6 were calculated using overall mass losses measured with the balance. Additionally, mass loss data obtained during the cool down period was unavailable because of load cell sensitivity to temperature change.

FRUIT QUALITY

Fruit quality is influenced by genetic factors, preharvest environmental factors, harvesting, postharvest treatments, and interaction among the various factors listed above (Kader, 1985). Preharvest environmental factors include climate, soil type, nutrient and water supply, pruning, and thinning. Fruit maturity, ripeness, and physiological age at harvesting influence product quality. Postharvest handling, temperature, relative humidity, and atmosphere composition during storage, and duration between harvest and consumption also affect fruit quality (Kader, 1985).

Standard fruit quality parameters measured at the beginning and end of each storage period are presented in Tables 7 and 8. All values were obtained by averaging measurements from each bulk lot in each room. Average firmness values ranged from 77.4 to 81.9 N when the fruit was placed into storage in 1990 and from 76.4 to

Table 7. Fruit Quality Parameters Before and After 1990-91 Storage Season

	1990-91 SEASON							
	Before Storage					After Storage		
	Pressure (N)			Starch	Sol.Solids	Pressure (N)		
	High	Low	Avg	(1-6)	(%)	High	Low	Avg
Room 4	86.7	76.5	81.9	2.0	11.2	77.8	60.1	69.4
Room 5	85.4	76.1	80.1	1.9	10.8	80.1	60.5	70.3
Room 14	83.6	72.5	77.8	2.1	11.3	76.5	53.8	66.4
Room 15	81.9	73.4	77.4	2.1	11.2	76.5	57.4	66.7

Table 8. Fruit Quality Parameters Before and After 1991-92 Storage Season

	1991-92 SEASON							
	Before Storage					After Storage		
	Pressure (N)			Starch	Sol.Solids	Pressure (N)		
	High	Low	Avg	(1-6)	(%)	High	Low	Avg
Room 4	92.3	76.4	83.9	2.0	11.0	80.2	64.1	72.4
Room 5	86.0	76.8	81.0	2.0	13.5	76.3	63.3	70.1
Room 14	96.0	75.1	84.0	2.5	11.2	79.6	64.5	71.5
Room 15	86.4	68.7	76.4	1.9	10.8	71.1	55.1	63.7

Table 9. Transpiration Coefficients

	Km (mg/kg·s·kPa)	
	1990-91	1991-92
Room 4	0.027	0.034
Room 5	0.031	0.024
Room 14	0.030	0.035
Room 15	0.026	0.021

84.0 N at the beginning of the 1991-92 storage season. Soluble solids content ranged from 10.8 to 11.3% in 1990 and from 10.8 to 11.5% in 1991 when fruit was placed into storage. Starch content ranged from 1.9 to 2.1 in 1990 and from 1.9 to 2.5 in 1991 when fruit was placed into storage. Average firmness losses were 12.5, 9.8, 11.4, and 10.7 N in Rooms 4, 5, 14, and 15, respectively, during the first season, and 11.5, 10.9, 12.5, and 12.7 N, respectively, during the second season. No direct correlation can be determined between product firmness loss and loss of mass, or between firmness loss and evaporator fan operation.

TRANSPIRATION COEFFICIENT

Transpiration is fruit water loss through the processes of evaporation and diffusion. Transpiration coefficient of a fruit or vegetable is the mass of moisture transpired per unit mass of commodity, per unit environmental water vapor pressure deficit, per unit time. Overall transpiration coefficients were calculated using equations described by Chau et al. (1987), Gaffney (1985), and Sastry (1985). Sastry et al. (1978) reported an average transpiration coefficient of 0.042 mg/kg·s·kPa for all apple varieties. Transpiration coefficient values for apples in the literature ranged from 0.016 to 0.10 mg/kg·s·kPa (Sastry et al., 1978).

Transpiration coefficients (K_m) ranged from 0.026 to 0.031 mg/kg·s·kPa during the first storage season and from 0.021 to 0.035 mg/kg·s·kPa during the second season (Table 9). Several assumptions were made in calculating the transpiration coefficients. The total measured mass loss was assumed to be due to transpiration (i. e., carbon loss as a result of respiration was neglected). Also, the vapor pressure of the intercellular spaces in the product was assumed to be the vapor pressure of the surrounding air at saturation. Average dry-bulb and dew point temperatures used to determine transpiration coefficients are shown in Table 3.

REFRIGERANT FLOW

Refrigerant flow was measured in the liquid supply line between the solenoid and hand expansion valves (Figure 3). Refrigerant flow data was unavailable in all rooms during the first season and in Rooms 5 and 15 during the second season. Monthly average refrigerant flow values upon cool down during continuous fan operation ranged from 0.46 L/min in Room 14 to 2.1 L/min in Room 4. Refrigerant flow averaged 0.78 L/min during three months of fan cycling operation in Room 4 (December, January, and February). Wide variation in refrigerant flow between Rooms 4 and 14 prevented comparison of continuous evaporator fan operation and fan cycling operation. Lack of knowledge of the state of refrigerant exiting the evaporator coils complicated refrigerant flow data analysis. In flooded coil evaporators, refrigerant may exit the coil as a liquid, gas, or a combination of the two, depending on the level of refrigerant in the surge drum and heat load.

A room heat load of 9100 W (continuous fan operation) requires approximately 950 L/day of liquid ammonia refrigerant at -1°C. Evaporator fan cycling can potentially decrease fan motor heat input by 65%; therefore, total room heat load would decrease by

30% to 6300 W which requires 660 L/day of liquid ammonia refrigerant at -1°C.

REFRIGERATION SYSTEM PERFORMANCE IN ROOM 4

Figures 32 through 36 show evaporator fan electrical energy use, evaporator coil inlet and outlet refrigerant temperatures, evaporator coil inlet and outlet air temperatures, bulk fruit temperature at R9S8B10, R9S8B6, and R9S8B1, and relative humidity at R5S8B10 in Room 4 during February 1992. Electrical energy use was consistent throughout the month (Figure 32). Refrigerant temperatures ranged from 1.0 to -1.0°C (Figure 33) and evaporator coil inlet and outlet air temperatures each varied about 0.7°C (Figure 34). Figure 35 shows the temperature recorded by three thermistors placed among the fruit mass. Bulk fruit temperature was maintained within ±0.1°C throughout the month. Relative humidity was very stable and ranged from 94.0 to 95.5% (Figure 36). Figures 32-36 illustrate refrigeration system performance in Room 4 during February 1992 was excellent.

Refrigeration system performance was drastically different in Room 4 during June and July 1992. Figures 37-42 show evaporator fan electrical energy use, evaporator fan operation time, evaporator coil inlet and outlet refrigerant temperatures, evaporator coil inlet and outlet air temperatures, bulk fruit temperature at R9S8B10, R9S8B6, and R9S8B1, and relative humidity at R5S8B10 in Room 4 during July 1992. Electrical energy use from July 10 to July 29 was greater than normal (Figure 37). Data from July 7 and July 19 are missing. Figure 38 shows that the evaporator fans were on

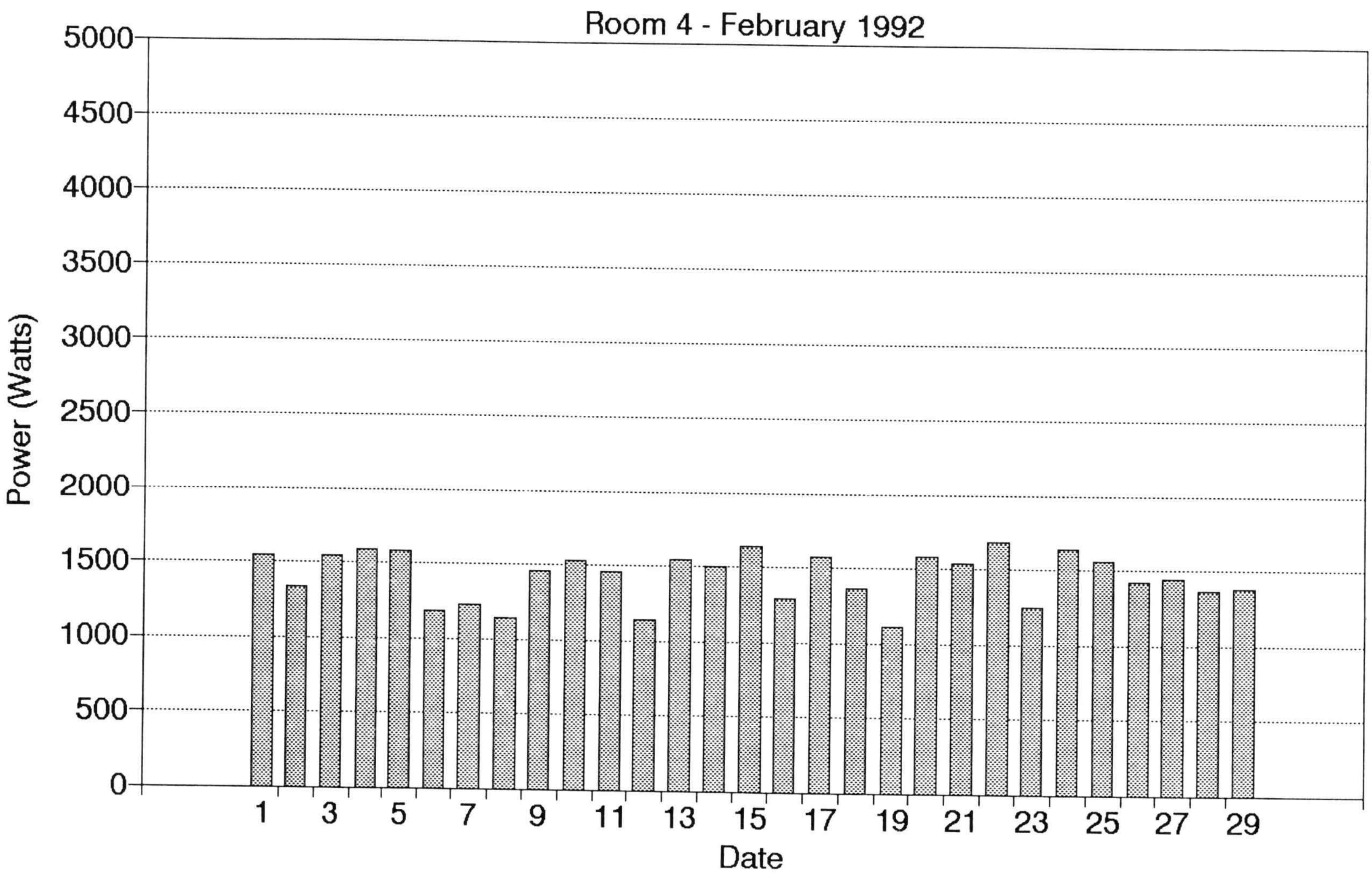


Figure 32. Evaporator Fan Electrical Energy Use in Room 4 During February 1992

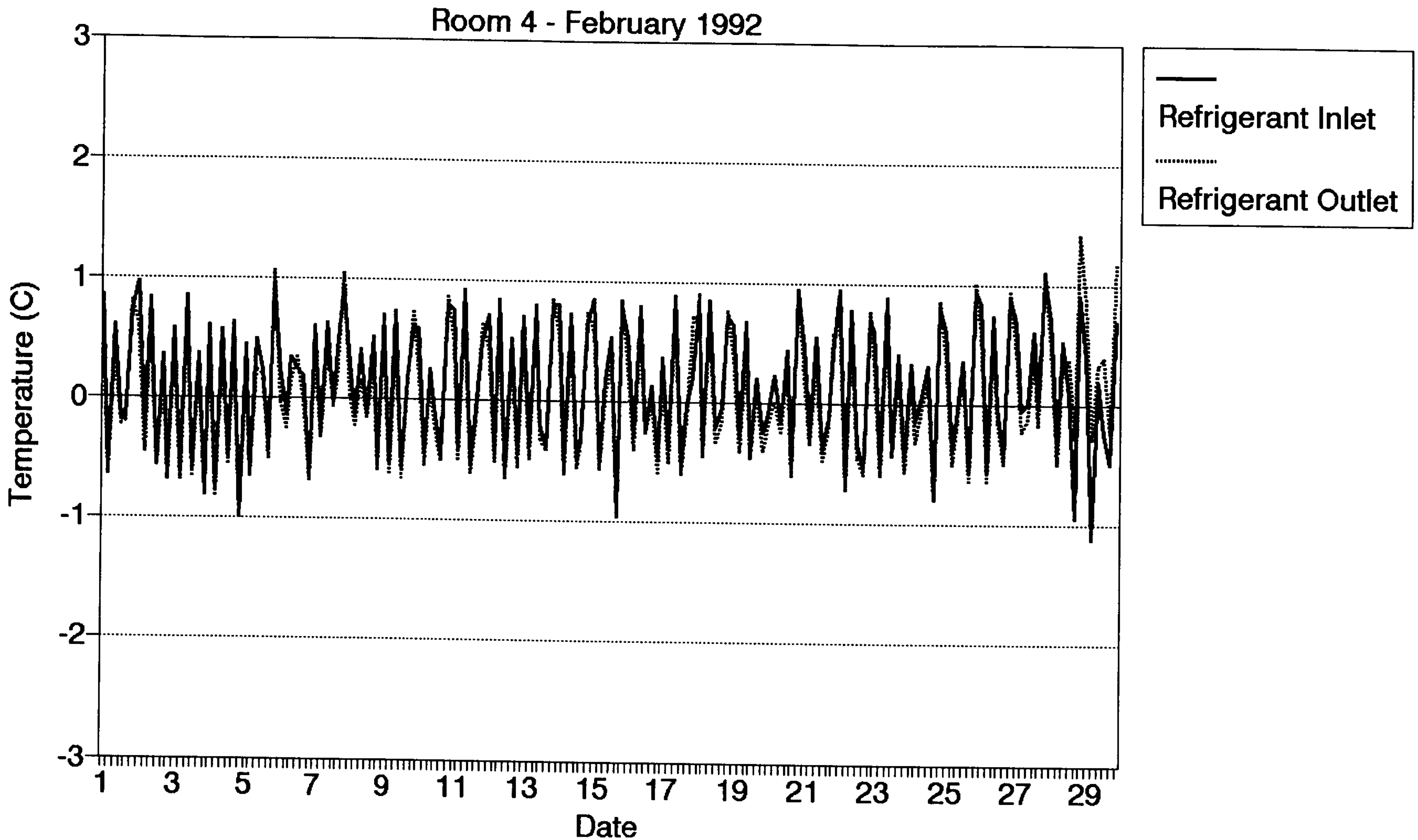


Figure 33. Evaporator Coil Refrigerant Temperature Histories in Room 4 During February 1992

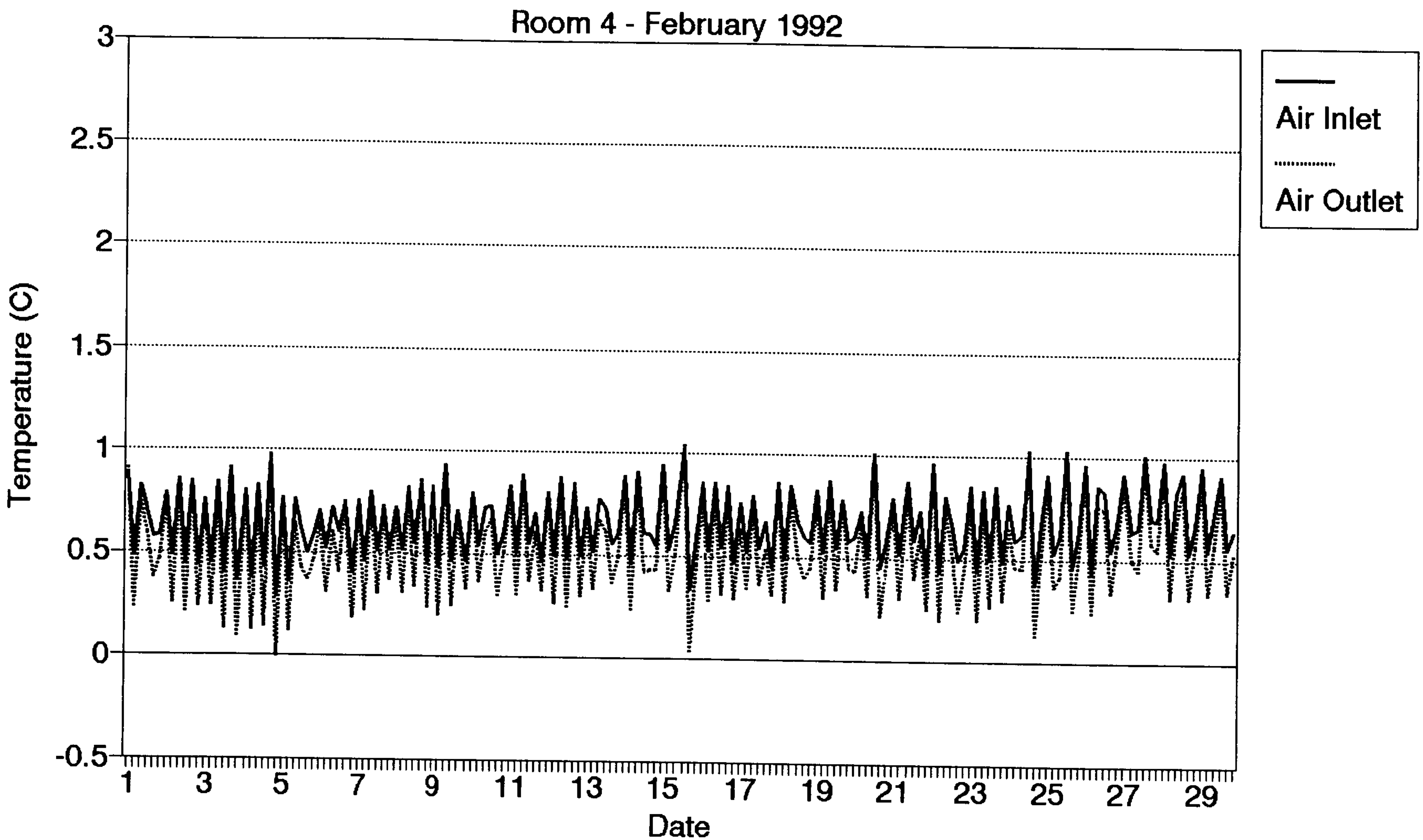


Figure 34. Evaporator Coil Air Temperature Histories in Room 4 During February 1992

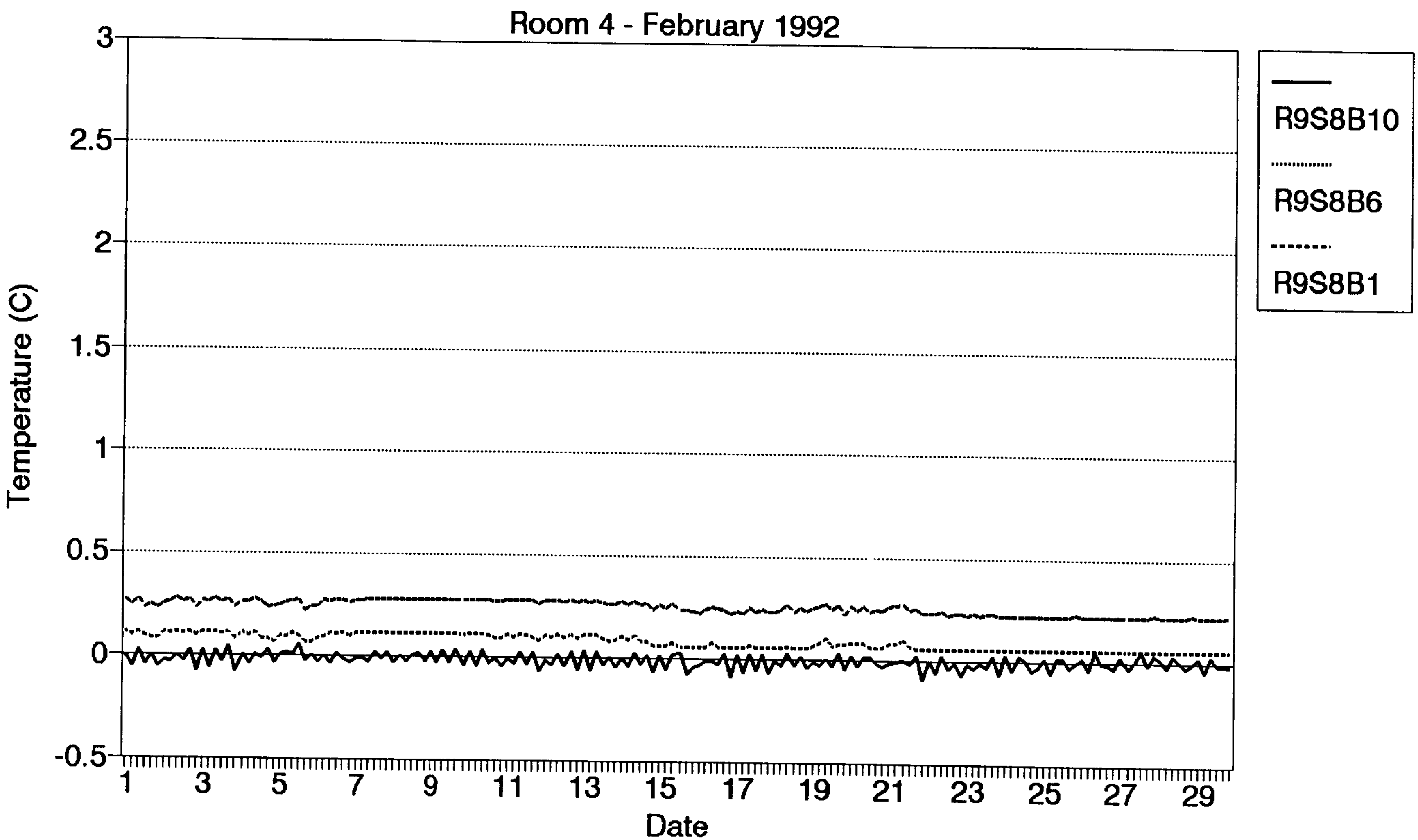


Figure 35. Fruit Bin Temperature Histories in Room 4 During February 1992

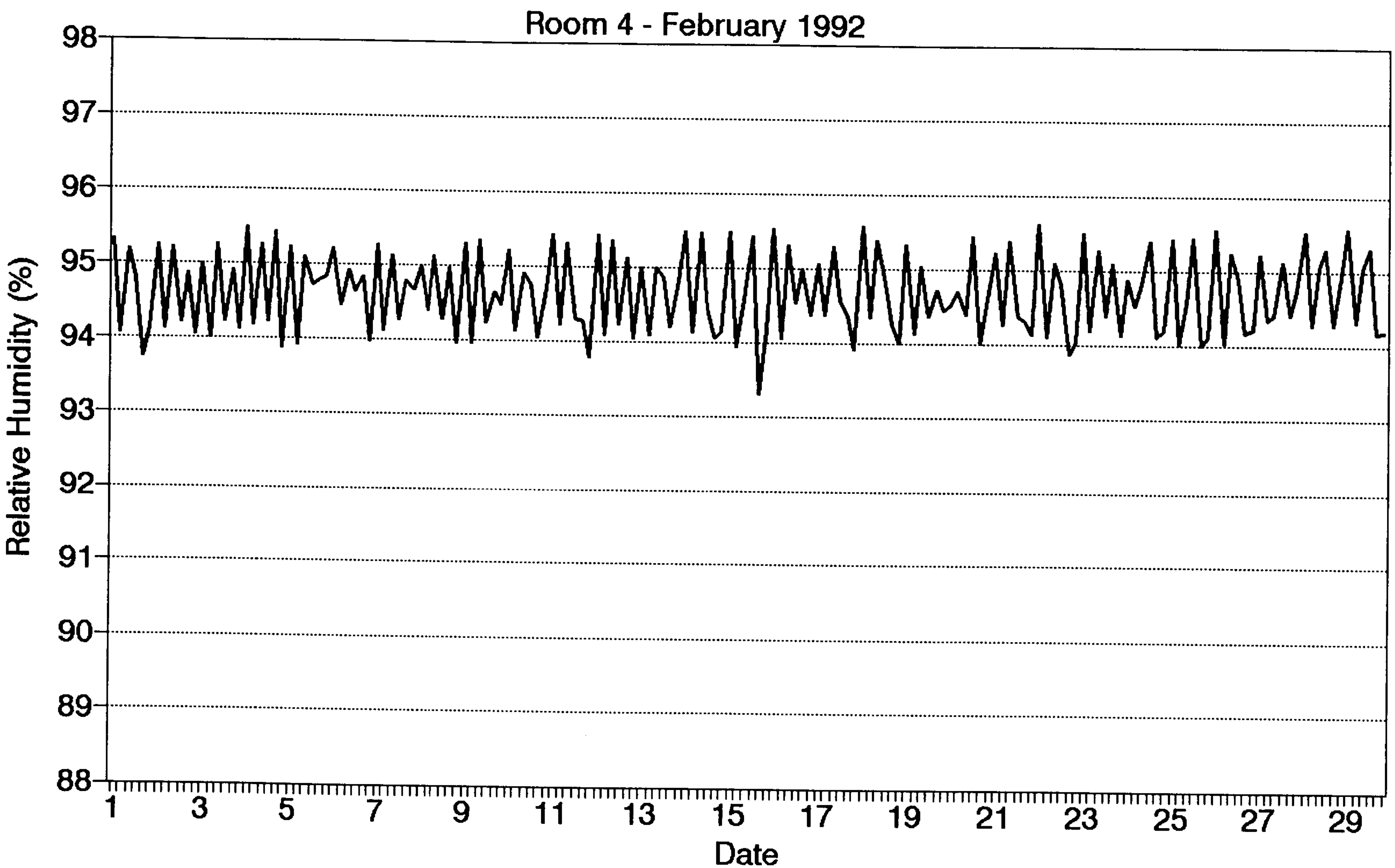


Figure 36. Relative Humidity in Room 4 During February 1992

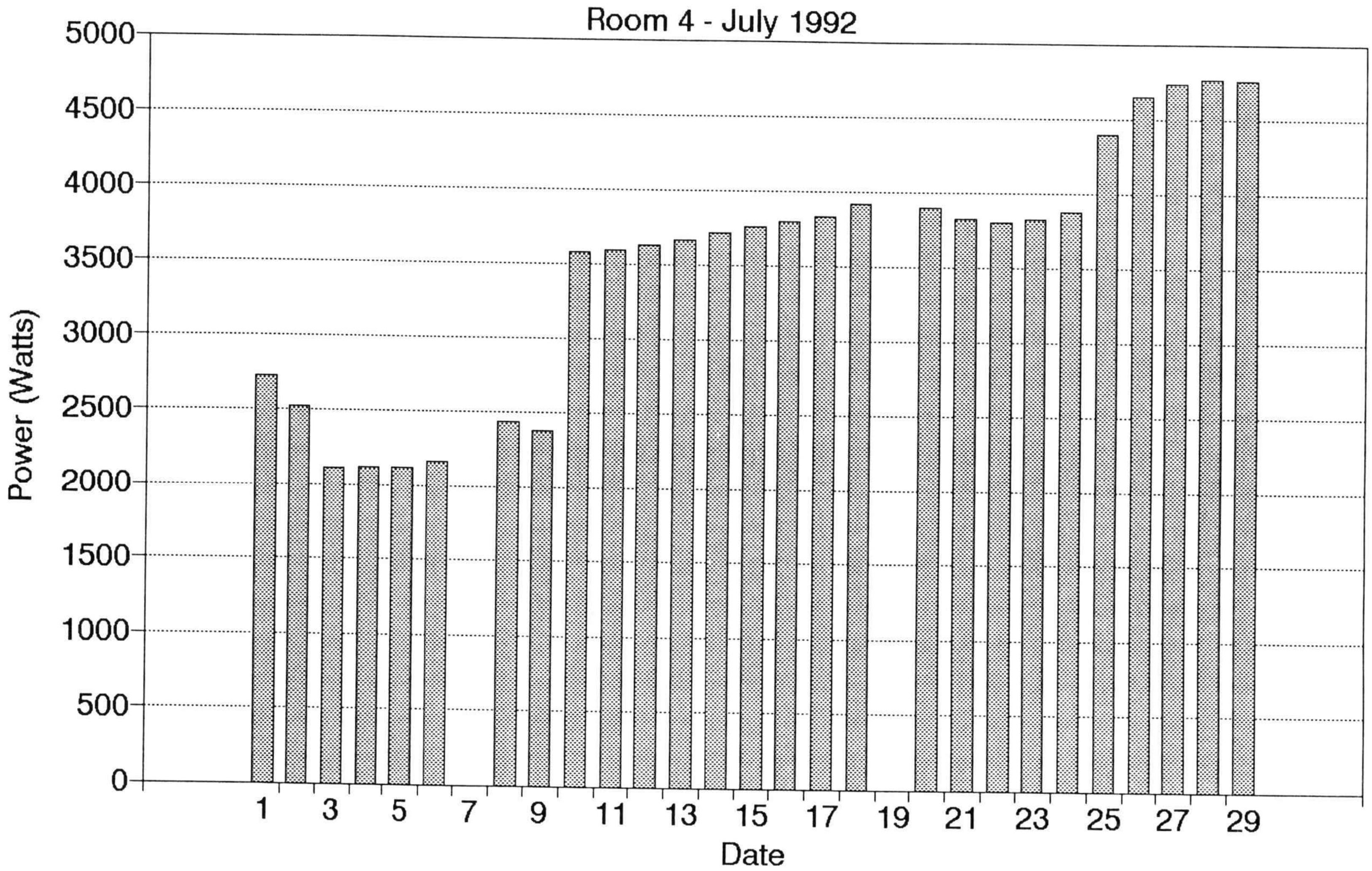


Figure 37. Evaporator Fan Electrical Energy Use in Room 4 During July 1992

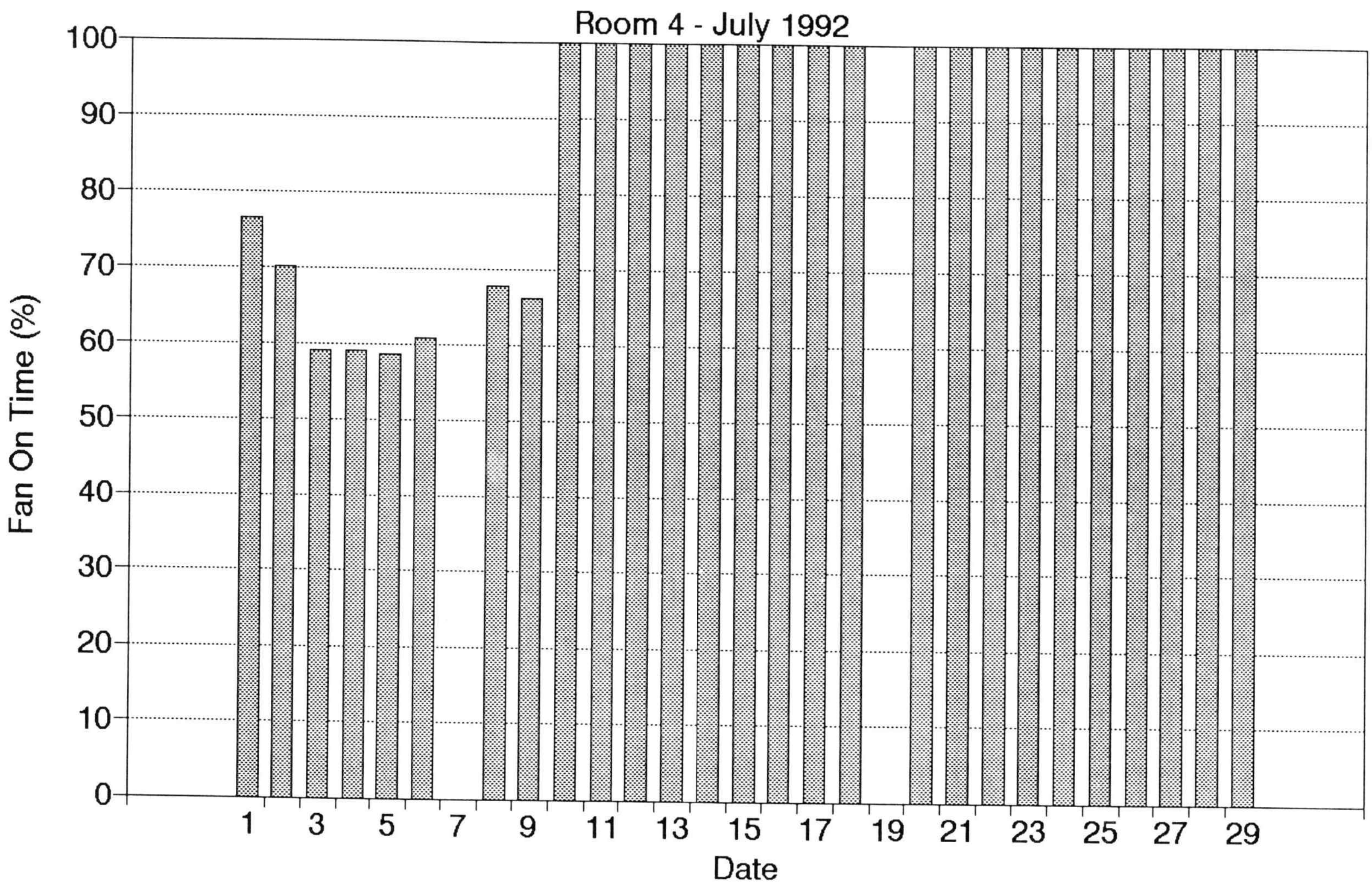


Figure 38. Evaporator Fan Operation Time in Room 4 During July 1992

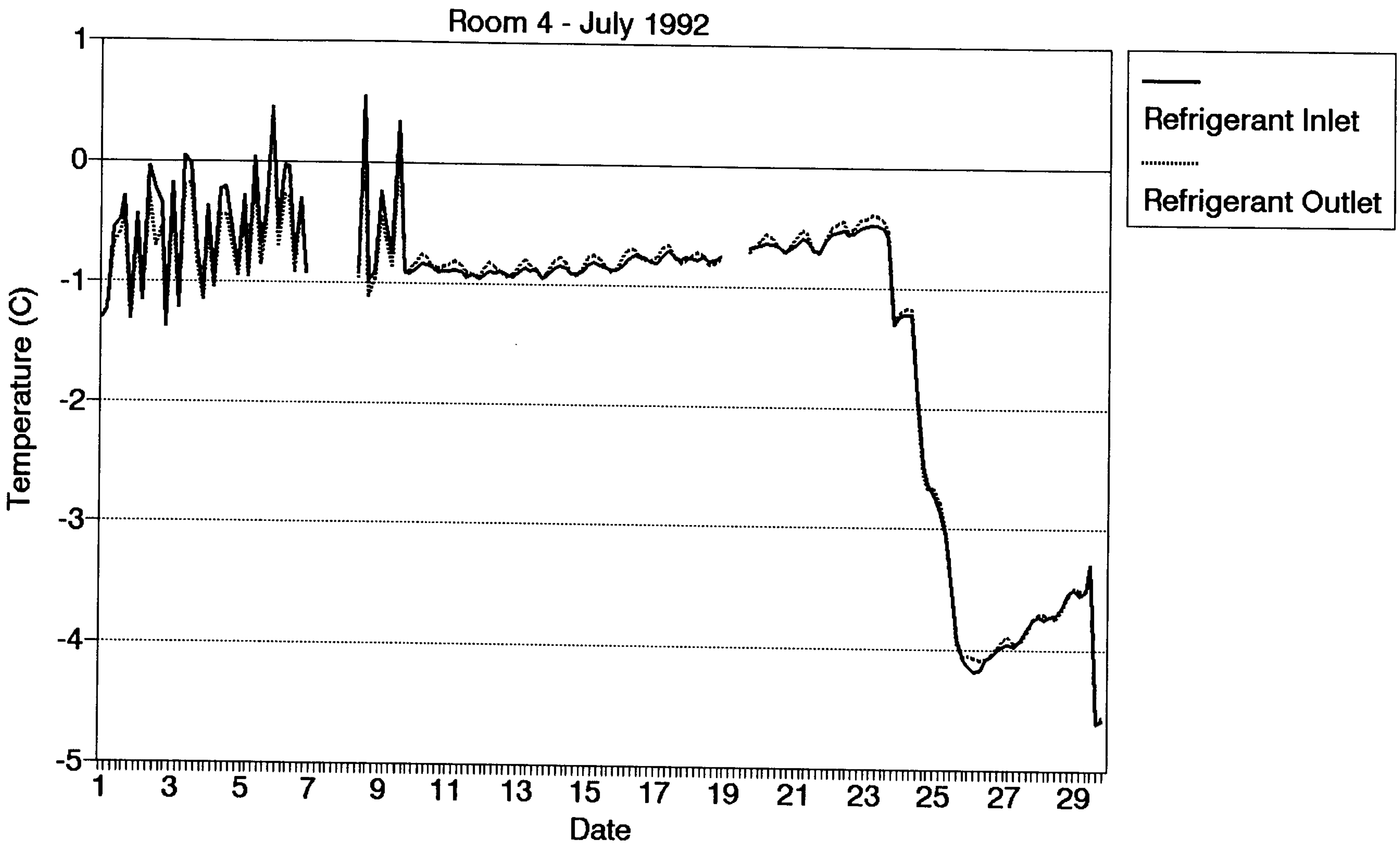


Figure 39. Evaporator Coil Refrigerant Temperature Histories in Room 4 During July 1992

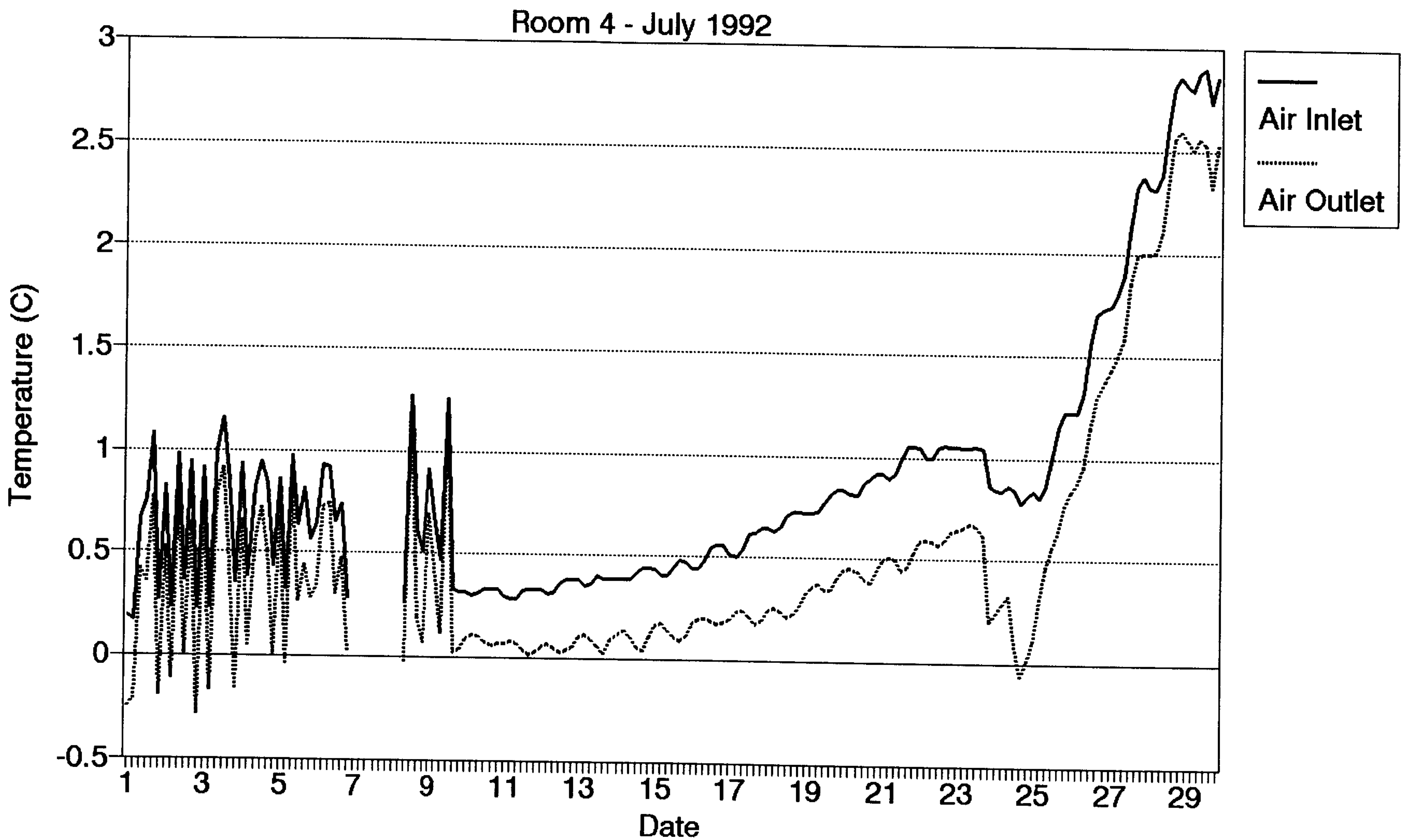


Figure 40. Evaporator Coil Air Temperature Histories in Room 4 During July 1992

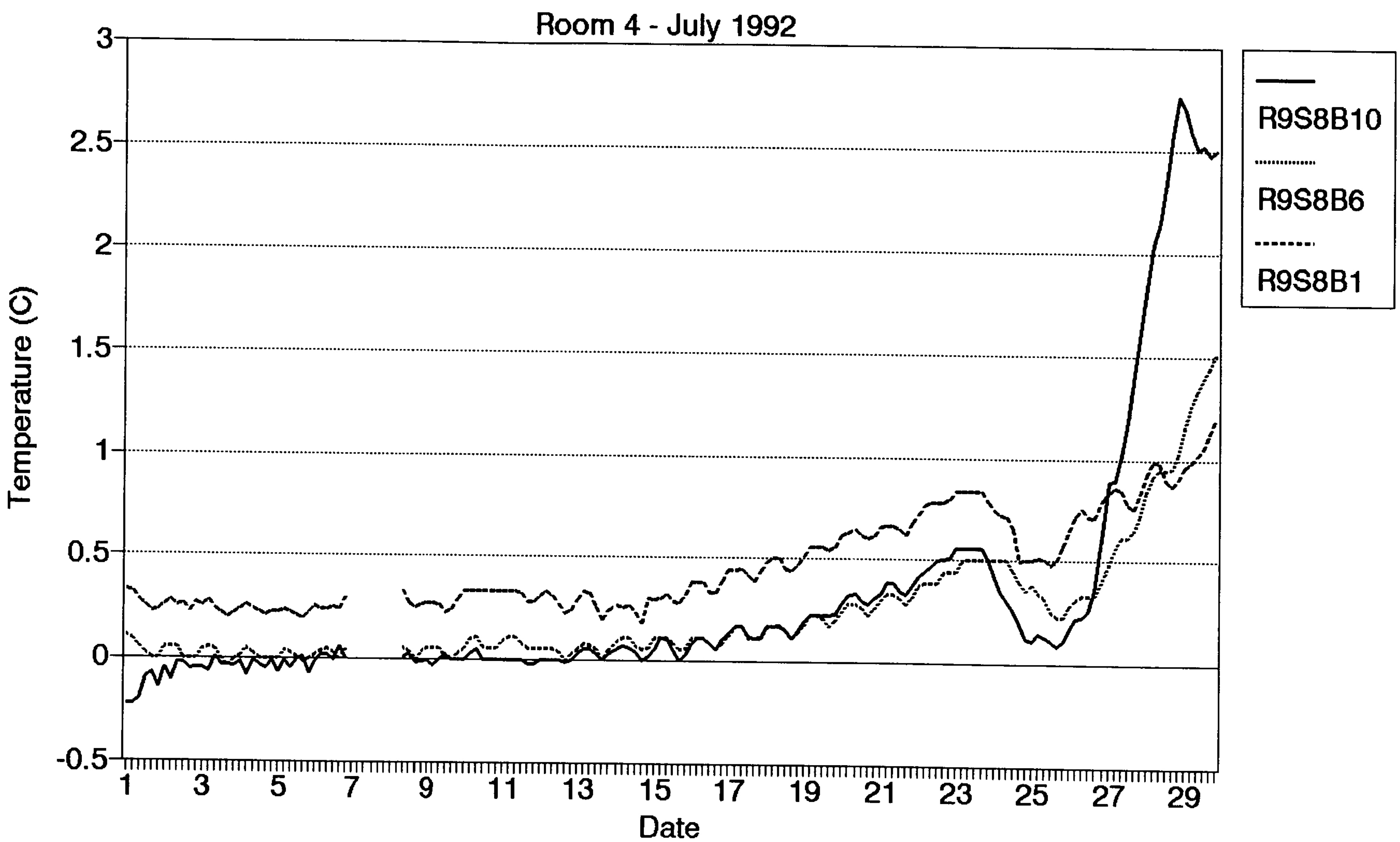


Figure 41. Fruit Bin Temperature Histories in Room 4 During July 1992

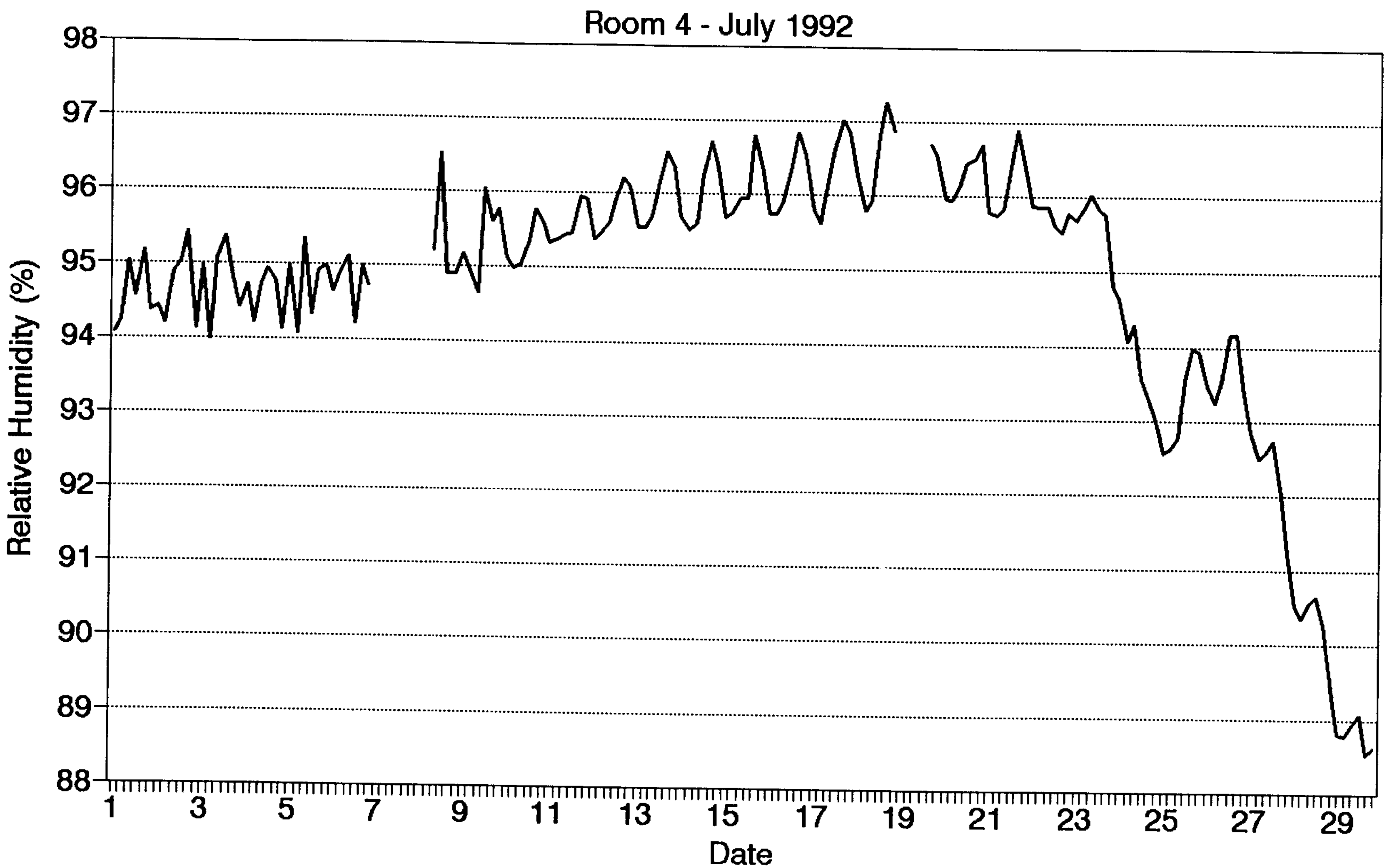


Figure 42. Relative Humidity in Room 4 During July 1992

continuously from July 10 to July 29. Figure 37 indicates that only a portion of the evaporator fans were in operation during this period since power levels were below 4368 W. Electrical energy consumption increased slightly each day as ice built up on the fins of the evaporator coil. Beginning July 10 the evaporator fans no longer cycled as indicated by the refrigerant and air temperatures entering and exiting the evaporator coil (Figures 39 and 40). On July 23 refrigerant temperatures began to drop to -4.2°C (Figure 39). This had little effect on room air and fruit temperatures since ice build-up on the evaporator coils prevented effective cooling. Room air and fruit temperatures increased to 2.8°C on July 28 (Figures 40 and 41). In addition, room relative humidity decreased from 96% to below 89% in seven days (Figure 42). Operating conditions such as these are detrimental to fruit mass loss and product quality. Ice build-up should have been detected within one to two days and defrost cycles manually executed to enable the refrigeration system to operate under design conditions.

POTENTIAL FACILITY ELECTRICAL ENERGY SAVINGS

This study has shown that evaporator fans can remain off 65% of the time or more after fruit cool down. Overall evaporator fan electrical energy savings approach 55% when a 30 day cool down period and a 170 day fan cycling storage period are considered. Evaporator fan energy savings of 218,000 kWh/year can be realized in the 18 room Snokist Complex B facility. Since evaporator fans are off 55% of the time, compressors do not need to perform work needed to remove 45.3 kW of otherwise produced fan motor heat. An overall compressor coefficient of performance of 3.95 results in an annual compressor electrical energy savings of 55,000 kWh due to fan cycling. Further energy savings are observed by the condenser and defrost equipment; however, the magnitude of these savings are small. Total annual electrical energy savings for Complex B as a result of evaporator fan cycling could approach 273,000 kWh or $12,300 ($0.045/kWh). The percent of total energy consumption that this figure represents varies and depends on the magnitude of component compressor loads (fan motors, heat of respiration generated by fruit, conduction, and infiltration).

POTENTIAL WASHINGTON STATE ELECTRICAL ENERGY SAVINGS

A projection based on storage capacity and the above energy savings indicates a potential 80 million kWh or $3.6 million of electrical energy can be saved annually by the Washington State apple storage industry as a result of evaporator fan cycling. This projection assumes storage facilities operated for the same number of days as stated above, on average. This projection also assumes all other storage facilities and refrigeration systems have equipment operating with the same efficiency as the warehouse monitored. The monitored complex is relatively new and contains modern equipment; therefore, energy consumption is more efficient than in the average storage facility and the potential electrical energy savings in Washington State may be even greater than the above stated amount.

CONCLUSIONS

State of the art flooded evaporator coil refrigeration systems constantly vary cooling rate by responding to air temperature sensed by a single thermostat. This thermostat is typically located near the inlet side of the evaporator coil. Placement of the sensor in this location causes the refrigeration system to react to the warmest air temperature in the storage space. The difference between the air temperature sensed by the thermostat and the set point temperature controls opening and closing of the evaporator coil back pressure regulator valve.

In this research, three computer control algorithms were used to regulate operation of the back pressure regulator in two experimental (fan cycled) rooms. Two algorithms, used during the 1990-91 storage season, resulted in unsatisfactory control of the back pressure regulator, which caused average relative humidity levels to remain approximately three to four percent lower than in the non-fan cycled rooms. Excessive opening of the back pressure regulator at the beginning of any fan-on period caused unnecessarily cold surfaces to be presented to the air being circulated past.

A third computer control algorithm, used in the two experimental rooms during the 1991-92 season, resulted in increased room relative humidity and lower mass loss rates. Figure 36 illustrates that relative humidity remained stable at nearly 95% in Room 4 during February 1992. The set point temperature used to control back pressure regulator valve opening during evaporator fan operation was provided by a weighted average of three thermistors located in the air. Computer software caused the back pressure regulator valve to close when the evaporator fans were off and ramp to its operating open position over a specified time period when the fans were turned on.

Room cooling demand and evaporator fan operation was controlled by the bulk fruit temperature in the fan cycled rooms. Although total fluctuations of air temperatures in all rooms were approximately equal, fruit temperature was approximately five times more stable in the computer controlled, fan cycled rooms (Figures 16-21) as in the thermostat controlled, non-fan cycled rooms (Figures 22-27). The exceptionally uniform temperatures observed in Figures 16-21 are highly desirable for long term storage. Bulk fruit temperature was effectively measured by placing the thermistor probes at least two, but not greater than three layers beneath the surface. Placement of thermistors used to control fan cycling on the surface of the fruit in a bin caused increased fan cycling and unstable refrigeration system operation. Inserting thermistors into fruit flesh or placement of thermistors below three layers of fruit excessively dampens refrigeration response, which may cause apples in surface layers to freeze when the fans are on, or warm excessively when the fans are off.

Data collected during this two-year project indicated 60-65% of the electrical energy used to operate evaporator fans can be saved upon fruit cool down. Figure 32 illustrates that evaporator fans remained off 66.6% of the time in Room 4 during February 1992. Overall annual evaporator fan energy savings approached 50-55% when the product remained in storage a typical length of time. An estimated 218,000 kWh of evaporator fan electrical energy can be saved annually in the 18 room, 9410 metric ton storage facility.

Electrical energy savings are realized by the compressor motors as a result of less fan motor heat input. An additional 55,000 kWh or 25% of the evaporator fan electrical energy savings can be realized in the 18 room facility. Projections based on total storage capacity indicate the Washington State apple storage industry can save 80 million kWh of electrical energy annually as a result of evaporator fan cycling. Based on current costs, this represents an annual savings of approximately $3.6 million.

ADDITIONAL OBSERVATIONS AND RECOMMENDATIONS

1. Fruit touching wooden bin surfaces lost significantly more mass than other fruit. Use of plastic bins may reduce product mass loss and improve fruit quality.
2. Relative humidity levels in all four rooms stabilized at 93-95% during the second season. Therefore, storage rooms with properly sized and designed evaporator coils do not require supplemental humidity injection.
3. Thermal expansion and contraction of the walls and doors leads to loss of an airtight door seal in some storage rooms, especially in south facing rooms. Re-establishing the door seal may be more cost effective than flushing the room with nitrogen gas to maintain the desired oxygen level.
4. Consistently higher temperatures recorded in bins on the floor indicates the need for additional floor insulation to reduce heat transfer into the structure.

5. Transpiration coefficient is determined from the association of vapor pressure differential and moisture mass loss rate. However, this relationship is not linear, especially as ambient conditions around the fruit are modified to achieve high humidity CA storage conditions. Transpiration coefficients are not as useful to the warehouse manager as mass loss rate data for a given commodity at at given set of environmental conditions.

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APPENDIX

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EQUIPMENT AND INSTRUMENTATION

Item	Manufacturer and Model[1]
Dew Point Sensor	General Eastern Instruments Humidity Instrument No. M-1 Auto Sensor Model No. 1111H
Evaporator Coils	Krack BTX-5-1300-FLA-WD
Load Cell	Hottinger Baldwin Messtechnik, Inc. Load Cell: Type U1T Amplifier: Model MGT232
Refrigerant Flow Meter	Omega Engineering, Inc. Flow Meter: FP-5200 Signal Conditioning Unit: FLSC-720
Scale	Mettler P1200
Thermistors	negative-temperature coefficient
Watt-Transducer	Rochester Instrument Systems Model PCE20P3E0C5X1F60W0Z1A2G1

[1] The use of trade names in this thesis does not imply endorsement of the product named, nor criticism of similar ones not mentioned.