Design, Fabrication, and Testing of a Returnable, Insulated, Nitrogen-Refrigerated, Controlled-Atmosphere, Shipping Container for Distribution of Fresh Red Meat

by

Christopher Glen Bailey

A thesis
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DESIGN, FABRICATION, AND TESTING OF A RETURNABLE, INSULATED, NITROGEN-REFRIGERATED, CONTROLLED-ATMOSPHERE, SHIPPING CONTAINER FOR DISTRIBUTION OF FRESH RED MEAT

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ABSTRACT

In today's market, packaging and cutting of fresh red meat occurs at both the wholesale and retail level. Greater economies could result if the wholesaler prepared all consumer cuts centrally. Despite advantages, meat storage life is inadequate for extensive centralized distribution, however, master packaging in combination with strict temperature control (-1.5 ± 0.5°C) increases storage life. Although master packaging technology exists, mechanical refrigerators usually fail to maintain this temperature range.

An insulated shipping and storage container was researched, designed, and tested for its suitability to distribute master packaged meat. A stainless steel rack was fitted in the container to support 36 master packages (508 x 381 x 60 mm) which were refrigerated by time-pulsed injections of liquid nitrogen (N_2). Electric fans provided uniformity of temperature within the container.

Thirty-six, saline water bags were used to simulate meat thermally and the temperatures of 20 bags were recorded. The container was tested at outside temperatures of 15, 0, and −15°C with four fans and at 30°C with two, four, and six fans. In all instances, bags cooled from an initial 10°C to an equilibrium temperature within 5.5 h. Minimum equilibrium temperatures during any 8 h trial were −2.6, −2.0, and −2.0°C for two, four, and six fans, respectively. Correspondingly, maximum temperatures were −0.2, −0.7, and −0.3°C. Initial chilling of the product required, on average, 19 kg of N_2, while equilibrium was maintained at a N_2 consumption rate of 5.5, 4.0, 2.6, and 0.93 kg/h at outside temperatures of 30, 15, 0, and −15°C, respectively with four fans. Usage for two and six fans was 5 and 6.3 kg/h, respectively at 30°C outside temperature.

For simulated power failure or the N_2-tank running dry, temperatures in the container rose 0.9 and 2.0 °C/h, respectively. When the door to the container was opened long enough to remove three trays, temperature was restored within 5 min.

Convective heat transfer coefficients between saline water bags and circulating N_2 were in the range of 80 to 100, 115 to 135, and 140 to 155 $W/(m^2 \bullet K)$ for two, four, and six fans, respectively. Heat transfer to meat will be limited by conduction in master packaged meat if similar convection coefficients prevail.

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I wish to express my gratitude to Carolyn Domytrak for the research she completed on this project, her design ideas, and for all the *leg work* she did. Dr. R.A. Holley's commitment to providing me with the appropriate background on meat science as well as packaging materials was excellent and as such was much appreciated.

Being as this was a design thesis, I owe my success to the Messrs. Dale Bourns, Matt McDonald, and Jack Putnam who provided me with invaluable design ideas and saw the project through to completion. Thank-you.

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1. INTRODUCTION

Highly perishable food products (e.g. meat) present marketing problems which are unique and challenging. Some preservation techniques, such as smoking or pickling can enhance the flavour and quality of food as well as facilitate distribution. Nevertheless, in most markets a demand still exists for the *fresh* product form, and often consumers will pay a premium price for this freshness. The challenge from scientific and engineering perspectives is to develop the packaging and shipping technologies to market these highly perishable fresh foods.

Master packaging is a modified atmosphere packaging (MAP) technique used to extend the storage life of fresh red meat (referring to chilled beef, lamb, pork, or veal). In combination with strict temperature control, master packaging technology is able to provide a consistent, fresh consumer product. Although appropriate technology exists to facilitate use of master packaging in industry, most refrigeration systems are incapable of maintaining the narrow range of product temperatures desired (Gill and Jones 1992; Gill and Phillips 1993a, 1993b; Gill et al. 1995). If a simple, economical method of adequately refrigerating meat during transport and retail storage could be developed, grocery chains will then be able to offer a wide variety of fresh meat, consistently meeting consumer demand with reduced spoilage.

The objectives of this thesis were:

1. to research, design, and fabricate an insulated, controlled-atmosphere, refrigerated container for use in the distribution of fresh red meat to domestic and international markets by method of the master packaging technology.

2. to evaluate the container for its ability to chill and maintain meat temperatures under various operating conditions.

3. to determine thermal characteristics of the system so future researchers have a basis for engineering design and comparison to traditional refrigeration systems.

4. to develop a finite element model to predict heat flow from meat stored inside a master package to aid future researchers in design and shipping considerations.

2. LITERATURE REVIEW

2.1. Meat Distribution Systems

Many small processing and packaging plants exist in today's red meat industry and, therefore, wholesale distribution occurs over relatively short distances. In this system, live animals are delivered to a slaughter house, where they are processed into primal, sub-primal, and consumer cuts. Primal and sub-primal cuts are large in size (15-50 kg), while consumer cuts are retail-ready. Most meat leaves the packaging plant as either primal or sub-primal cuts, requiring further fabrication at retail outlets. Meat is distributed by means of refrigerated tractor trailers and rail cars.

Today's meat processing and distribution system is inefficient and would become more profitable through the implementation of centralized processing and packaging of retail-ready meat cuts (Scholtz *et al.* 1992). In the new distribution system, all meat would be delivered from a large, centralized packaging plant to distant retailers as consumer cuts. Benefits would accrue to wholesalers, retailers, consumers, and the agricultural community.

Wholesalers would realize greater profits in a centralized meat distribution system because of economies of scale. This economic principle is explained by the following quotation:

> "Some advantages for the wholesaler of central packaging include better utilisation of resources, because of better inventory control and less waste through utilisation of fat, bone and trimmings; the possibility of using automated equipment and more efficient use of manpower and space; better quality control; and improved profits" (Cole 1986 and Sains 1988, as cited by Scholtz *et al.* 1992).

Retailers have shown considerable interest in the central preparation of retail-ready cuts because of lower in-store labour costs and ready-to-go convenience of products packed by suppliers (Creighton 1996). Floor space within the market place can then be devoted to merchandising of cuts, rather than to their fabrication. Consequently, retail stores would realize considerable profit if they were able to obtain chilled meat as retail-ready items (Farris *et al.* 1991, as cited by Gill and Jones 1994).

Consumers would capitalize on the centralized packaging of retail cuts through economic savings passed down to them by the wholesaler and retailer. Since only two-thirds of a beef carcass is useable meat, central processing would prevent the costly refrigeration and transportation of low value cuts or waste (Taylor 1985). Consistency and variety of products is another major benefit.

Centralized processing and packaging of meat is also desirable for Canadian livestock producers. A system dedicated to distributing meat over greater distances with increased efficiencies will undoubtedly expand markets for Canadian meat - both domestically and internationally. As well, a local means of value added processing is very attractive to farmers of the Canadian Prairies because of the recent abolishment of the Western Grain Transportation Act.

Disadvantages to centralized processing and packaging of meat also exist. Greater efficiencies within the industry will mean relocation of jobs and layoffs for meat cutters. Moreover, consumers will lose the personalized service which they have come to expect from meat retailers. Consequently, acquiring specialized cuts will be difficult.

2.2. Storage Life of Meat

2.2.1. Reasons for Wanting Meat with Extended Shelf Life There are many advantages to centralized processing and packaging of retail-ready meat cuts making it a desirable system. However, centralized processing it is not widely accepted by industry because distribution of meat over greater distances, increases the length of time between preparation and purchase by the consumer. Meat packaged in conventional retail packs has an extremely short shelf life and the time available for distribution and display of the product is severely limited (Gill and Jones 1994). Consequently, to realize the benefits of centralized packaging, a means of extending the storage life of meat must be found.

Two other reasons also exist for wanting an extended life. Currently, retail managers content themselves with ordering 90% or less of the pork cuts they hope to sell, because surplus has to be down-graded, reducing profit (Joll 1981, as cited by Buys *et al.* 1993). Another reason for extending the shelf life of meat is to enable surface transportation of meat products to export markets. Although domestic consumption of meat remains constant, export markets continue to grow - primarily exports to Mexico, Asia, and the Pacific rim countries (Anonymous 1995).

2.2.2. Microbiological Safety The most important factor affecting the shelf life of meat is microbiological safety. Modified atmosphere packaging (MAP) is a widely adopted method of slowing microbial growth and encouraging greater storage life. However, with MAP of meat a major concern is the growth of toxin producing bacteria before visible signs of spoilage are present (Ooraikul 1991). The same author reports that even though all necessary precautions are taken by the manufacturers, the possibility of thermal abuse by distributors and consumers cannot be ignored. Fortunately, the storage life of chilled meat packaged in a modified atmosphere (MA) is usually limited by deterioration of colour rather than excessive growth of bacteria (Shay and Egan 1990).

2.2.3. Colour Changes Another factor limiting the shelf life of meat is colour changes. Penney and Bell (1993) report that besides cost and culinary considerations, colour is arguably the most important factor used by consumers for selecting red meat products.

Researchers (Holley *et al.* 1993; Penney and Bell 1993; Taylor 1985) state that the colour of meat is largely determined by the oxidation state of the muscle pigment myoglobin. Raw meat stored in the complete absence of oxygen (O_2) will assume the purple colour of deoxymyoglobin. Under high partial pressures of O_2, purple myoglobin is rapidly oxygenated through a reversible process to the bright, red pigment oxymyoglobin. This pigment gives meat the colour which consumers commonly associate with freshness. Under low O_2 concentrations and with time, myoglobin will favour the formation of brown metmyoglobin. This process is largely irreversible and gives meat a colour that is typically associated with spoilage.

2.3. The Relationship Between Storage Temperature and Storage Life

To prevent undesirable changes in the meat, it is necessary to maintain a low microbiological load within the product. Because the growth of spoilage bacteria is proportional to the meat temperature, maximum storage life is achieved when fresh red meat is stored at the minimum temperature that can be maintained indefinitely without product freezing (Gill and Jones 1992). According to Gill *et al.* (1988, as cited by Gill and Jones 1992), the optimum temperature for packaged meat is $-1.5\,^{\circ}C$.

Despite various methods of packaging meat, the proportional loss of storage life with increasing temperatures is approximately the same (Gill and Phillips 1993a). At 0, 2, 5, and 10°C the storage life of meat is respectively 70, 50, 30, and 15% of the storage life obtained at the optimum temperature of −1.5°C (Gill and Phillips 1993a). As well, an increase in temperature from −1.5 to −1.0°C results in a 10% decrease in storage life. Therefore, a narrow temperature range is desirable to achieve maximum storage life of the product.

Various studies have been completed to assess the preservative capabilities of current storage and distribution processes for fresh red meat (Gill and Jones 1992; Gill and Phillips 1993a, 1993b; Gill *et al.* 1995). These studies, which concentrated on collecting temperature histories of the stored product, indicate that while some storage and distribution processes are satisfactory, many are not. Gill and Phillips (1993a) state:

> "... management of product with respect to time and temperature factors is commonly poor. That leads to product achieving a storage life that is at best much less than the possible maximum. Moreover, the storage life is often not only short but also uncertain. A shortened and variable storage life restricts the marketing possibilities for product, tightly constrains production and distribution schedules, and is a major cause of customer dissatisfaction or outright rejection of product. Proper management of time/temperature factors in storage and distribution will therefore substantially enhance product stability and reliability."

2.4. The Relationship Between Packaging Techniques and Storage Life

2.4.1. MAP with Carbon Dioxide (CO_2) By introducing or encouraging the formation of an atmosphere containing CO_2 in the absence of O_2, the shelf life of meat can be greatly increased (Holley *et al.* 1993). The shelf life of meat has been extended up to

15 times that in air (Gill and Penney 1988, as cited by Holley *et al.* 1994). Taylor (1985) states that "Not only can a storage life of several weeks be achieved, but during this time the meat is improving in tenderness."

There are two different methods used to obtain a CO_2 enriched environment. The first method, referred to as vacuum packaging, has been called the greatest innovation in meat handling in the last 20 yr (Taylor 1985). It involves drawing a vacuum within the meat package and sealing it. The respiration of bacteria and meat tissue quickly consume the residual O_2, releasing CO_2. Oxygen concentrations are reduced from 21 to less than 1%, while CO_2 concentrations increase from < 1% to 10-40% (Holley *et al.* 1993). The second method of obtaining a CO_2 enriched environment involves evacuation of air out of the package and back flushing it with CO_2. This packaging option is costly, but can offer improved storage life.

The mechanisms by which CO_2 extends shelf life are poorly understood. However, it is known that the presence of CO_2 in packaged meat produces an inhibitory effect on microbial growth (Holley *et al.* 1993; Stiles 1991). In the presence of O_2, the microbial flora is dominated by fast growing aerobic bacteria. When these bacteria reach sufficient numbers, they become organoleptically detectable (usually in the form of off-odours). With the introduction of CO_2 concentrations greater than 20%, a shift in the microflora from aerobic bacteria to slower-growing, facultatively aerobic bacteria occurs (Holley *et al.* 1993). The same authors report that the metabolic products of these spoilage bacteria are generally less offensive. The extended shelf life of meat packaged in an enriched CO_2 atmosphere may therefore be explained by the lag-time associated with the shift in microflora and because facultative aerobic bacteria grow slower and produce less offensive byproducts.

While MAP or vacuum packaging greatly extends the shelf life of meat, a major disadvantage is the purple, anoxic colour of the packaged meat. To achieve consumer acceptability, the meat can be repackaged with an O_2 permeable film before display. This allows oxygenation of the meat, producing the bright-red colour that consumers deem essential. However, this requires the use of expensive labour and space at retail outlets to produce the final display package.

2.4.2. MAP with High O_2 Atmosphere The colour of meat from MAP under anoxic conditions has proven to be unacceptable for most consumers (Taylor 1985). Thus, high-O_2 MA have been used in combination with CO_2 to create atmospheres which extend the storage life of the meat, yet maintain a fresh-red meat appearance. Work by Gill and Jones (1994) indicates a MA of O_2 + CO_2 will extend the shelf life of beef to approximately 2 wk while maintaining an acceptable colour. However, if packaged in an anoxic atmosphere of CO_2, the storage life is approximately three times greater (Gill and Jones 1994). A study by Gill and McGinnis (1993, as cited by Gill *et al.* 1994) of current meat distribution systems (from packing plant to retailer) in North America indicates a storage life approaching 3 wk is required for convenient marketing of beef because the product is stockpiled at both the wholesale and retail levels to meet the fluctuating demands of consumers. Therefore, the storage life achieved by MAP with high O_2 concentrations is not of sufficient length.

2.4.3. MAP with N_2 The effect on meat of MAP with N_2 is similar to that of vacuum packaging (Taylor 1985). The author also states that colour stability tends to be better than vacuum packaging since N_2 dilutes the residual O_2 and thereby reduces the formation of brown metmyoglobin. Unfortunately, N_2 also dilutes the CO_2 formed from the respiration of bacteria and meat tissue and reduces the inhibitory effect of CO_2. Experiments by Gill and Jones (1994) show that the storage life of steaks packaged in N_2 are usually limited by microbial spoilage rather than colour changes after a period of 4 wk. This is compared to the 7 wk storage period determined for meat packaged in a CO_2 MA.

When packaged with N_2, the meat assumes the purple colour of deoxymyoglobin. Therefore, this type of atmosphere is undesirable, unless the meat is repackaged at the retail outlet in a package that allows it to bloom. This requires expensive in-store labour and space for final preparation of the retail package.

2.4.4. Master Packaging of Meat All packaging techniques discussed previously had undesirable characteristics. The storage life was either too short or the colour unacceptable for consumers. Master packaging is a technology which overcomes both of these disadvantages (Gill and Jones 1994).

Master packaging is the use of two packaging layers. Meat cuts are individually overwrapped as retail-ready cuts with an O_2/CO_2 permeable film (Figure 2.1.). Numerous consumer trays are placed on a master tray, which is packaged with a film that is O_2/CO_2 impermeable. Before sealing, the master pack is evacuated and back flushed with a CO_2 enriched atmosphere. The O_2/CO_2 permeable films on the retail packaging allow the ingress of CO_2 into the meat. Therefore, master packaged meat has storage characteristics similar to that discussed for MAP with CO_2.

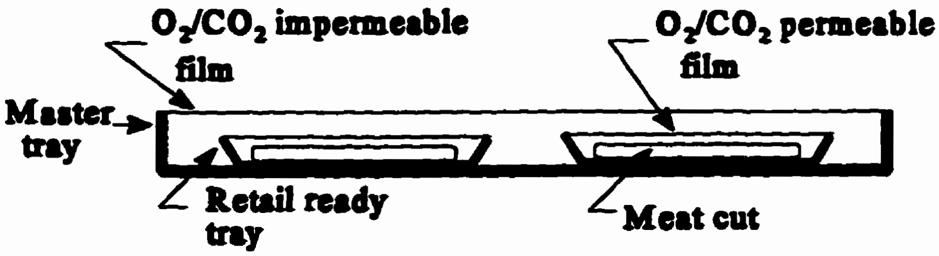


Figure 2.1. Schematic of a master pack used for packaging fresh red meat

Master packs are received by the retailer and stored until needed; then the enveloping O_2/CO_2 impermeable film is removed, allowing O_2 to initiate bloom by permeating through the packaging of the retail cuts. Oxygenation is completed in 0.5-2.0 h (Taylor 1985; Gill *et al.* 1994), giving the bright-red colour that consumers desire.

Scholtz *et al.* (1992) and Gill *et al.* (1994) have investigated the general acceptability of master packaged meat and found the results promising. Excessive exudate constituted a complaint from consumers, however, Gill *et al.* (1994) state that free exudate in packs could be collected by an absorbent of adequate capacity.

2.5. Summary

The literature reviewed indicates great benefits could be realized by wholesalers, retailers, consumers, and the agricultural community if the meat industry made a move towards centralized processing and packaging. However, this move has not and will not be made until an acceptable method is found of increasing meat storage life for transportation and distribution over greater distances.

Modified atmosphere packaging has been used successfully for many years to increase the storage life of meat. Packaging in atmospheres of CO_2 or N_2 produce products with suitable storage life for the centralized distribution of meat, but these methods require expensive in-store repackaging of the meat with O_2 permeable films to gain a consumer acceptable colour. Although packaging with an O_2 rich atmosphere encourages appropriate meat colour, the storage life is inadequate.

Master packaging is a MAP technique which has the merits of providing excellent shelf life and colour stability, without the need for expensive assembly of the final retail package at consumer outlets. This makes it ideal for the centralized distribution of meat.

To maximize the storage life of meat during distribution from a central location, it is imperative that product temperatures be strictly controlled at $-1.5 \pm 0.5°C$. Although some conventional refrigeration systems are capable of maintaining this temperature range, few do. This is a major barrier to centralized distribution by the meat industry. The remainder of this thesis will present the research, design, and testing of a refrigeration system which was developed to combine master packaging technology with strict product temperature control.

3. EXPERIMENT DESIGN

3.1. An Overview of the Insulated Container and Refrigeration System

Les Contenants Xactics Ltée, a company in Joliette, PQ, manufactures insulated containers used by the food industry. A container (Model: C-54) was purchased from the company and equipped with a stainless steel shelving system (see Appendix A for dimensioned drawings) suitable for holding 36 master trays (Figure 3.1.) at nine levels, with four master trays at each level. Trays were made to slide out of the shelving unit, allowing individual display at retail outlets.

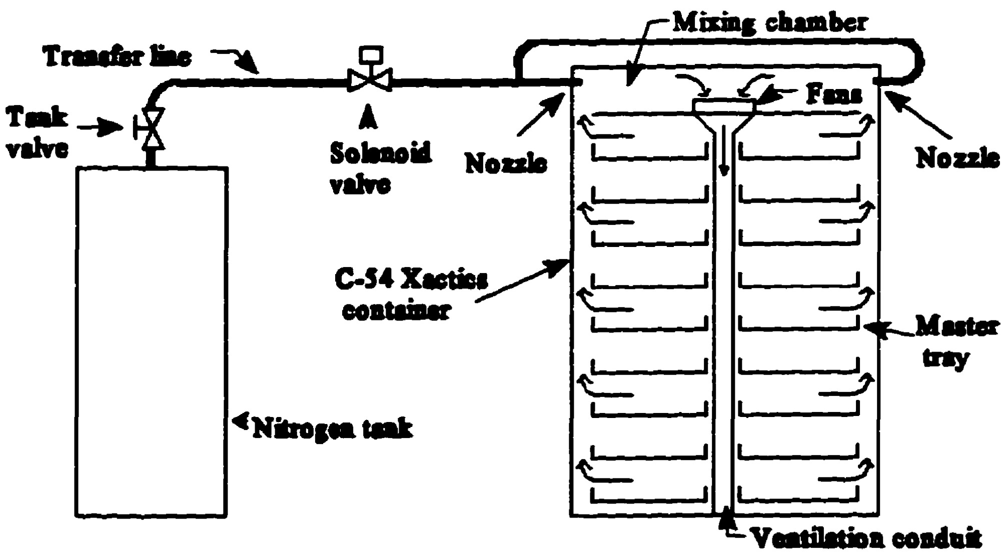


Figure 3.1. Schematic of the container (Model: C-54, Les Contenants Xactics Ltée, Joliette, PQ) equipped with master tray shelving unit and N_2 refrigeration system (arrows in container display N_2 flow patterns).

Polyethylene master trays with maximum dimensions of 508 x 381 x 60 mm can be accommodated in the current design. To achieve the full bacteriostatic effect of CO_2, master packs must be sized to contain sufficient quantity of CO_2 to fully saturate the meat and maintain a head space of excess CO_2 at atmospheric pressure (Gill and Penney 1988). The master tray dimensions were chosen to provide the recommended 2 L of CO_2 per kilogram of meat (Penney and Bell 1993).

Refrigeration was accomplished by injecting liquid N_2 from a pressurized source into the mixing chamber. A computer program was used to maintain the container temperature at $-1.5 \pm 0.5\,°C$ by toggling a solenoid valve on and off (which controlled the flow of N_2 to the container). Six fans (300 mm diameter and rated at 0.14 m^3/s in free air) located at the top of the ventilation conduit (which was constructed of 6 mm polyethylene), mixed the injected N_2 with circulated air to achieve a uniform, acceptable temperature. The fans can be operated in combinations of two, four, and six when the holes of non-operating fans are covered over. The N_2-air mixture travels down the centre conduit, across the master trays, and then returns to the mixing chamber. The designed air flow path keeps heat conducted through the walls away from the stored product.

3.2. An In-Depth Discussion of the Design and Design Decisions

3.2.1. Mechanical Refrigeration vs. Expansion of a Cryogenic Fluid Mechanical refrigeration systems have largely proven inadequate at maintaining product temperatures within the narrow range required for maximizing the shelf life of meat (Gill and Jones 1992; Gill and Phillips 1993a, 1993b; Gill *et al.* 1995). With the designed liquid N_2 system, the solenoid can be cycled on and off to achieve a temperature control that is not usually achieved by mechanical refrigeration systems. As well, the simplicity of the system enhances its reliability.

Another advantage of liquid N_2 refrigeration is lower fixed costs as compared to mechanical refrigerators (Guilfoy and Mongelli 1971). The weight of a liquid N_2 refrigeration system is almost half that of a similarly equipped mechanical refrigerator. This is an extremely important factor when the high costs of intercontinental shipments are considered. Moreover, current weight restrictions on tractor trailers can limit the quantity of boxed beef to a little over half of the trailer volume (Gill and Jones 1992).

The designed system is also very flexible. The same container will be used for chilling, transporting, and storing meat at retail outlets. Current meat distribution systems use separate refrigeration sources for each task.

Many mechanical refrigeration systems still use chlorinated fluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs) as refrigerants. The use of these substances is increasingly more regulated as scientists discover their detrimental effects to the environment. In contrast, N_2 is non-toxic to humans and the environment, with 78% of the natural atmosphere composed of it (Cameron 1989). As well, it is nonflammable and nonexplosive. Noise pollution and emissions from diesel engines used to power mechanical refrigerators, are also undesirable characteristics.

3.2.2. Liquid N_2 vs. CO_2 At the outset of this project, a decision was made to use liquid N_2 instead of CO_2 as the cooling medium for the refrigerated container. Both products are used extensively in the food industry. Although liquid N_2 has greater cooling potential per unit mass, it is considerably more expensive (see Appendix B for a thermodynamic and cost comparison between CO_2 and N_2). However, the decision to use liquid N_2 instead of CO_2 was made because it is difficult, if not impossible to avoid the formation of snow when using CO_2. Snow results when large pressure drops occur e.g.

high use of CO_2 can cause storage tank pressure drops; or pressure fluctuations may result from opening the solenoid valve. Snow can cause freezing or damage to the solenoid valve or it may plug the injection nozzles. Also, snow would accumulate in the upper portion of the container following injection from the nozzles, resulting in uneven distribution of the cooling medium. None of these problems are experienced with N_2 making it very desirable for the designed system.

3.2.3. Injection Nozzles The container was equipped with two injection nozzles orientated as shown in Figure 3.2. To promote even distribution of N_2 by each of the six fans, a 110° spray pattern was required. Because nozzles specifically designed for the injection of liquid N_2 were not found, various agricultural sprayer nozzles were tested in preliminary trials to determine their effectiveness. By trial and error, a Teejet nozzle (Model: 11006, Spraying Systems, Chicago, IL) was discovered to be the most effective. This nozzle is designed to give a 110°, flat, "V" pattern spray when agricultural chemicals are supplied at low pressures and flow rates. The nozzles were angled slightly above the horizontal plane (towards the roof of the container) to minimize exposure of the fans to extreme temperatures.

In preliminary trials, a system of four nozzles (two on opposing sides of the line of fans) was tested and found to give uneven spray distribution from the front to the back of the container. Two-phase flow may explain why N_2 was not distributed evenly to the four nozzles.

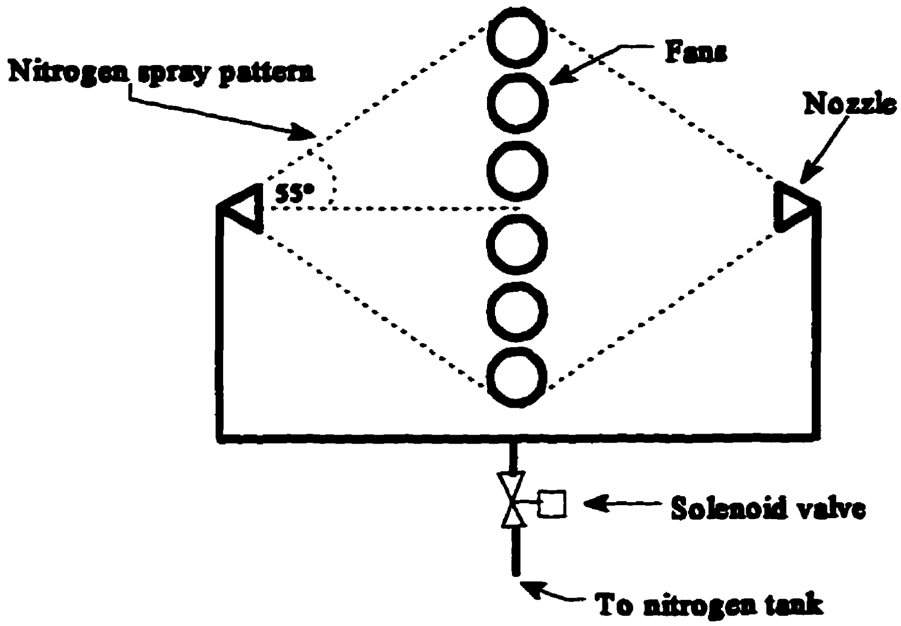


Figure 3.2. Top view of the N_2 injection system showing the location of nozzles, fans, and the predicted spray patterns of the nozzles.

3.2.4. Liquid N_2 Storage Tanks Liquid N_2 is stored as a boiling liquid in specially designed cryogenic tanks because of its extremely low temperature. All tests for this thesis were conducted with a DURA SERIES N_2-tank (capable of holding approximately 115 - 120 kg of N_2) manufactured by Minnesota Valley Engineering (MVE), New Prague, MN. To prevent excessive boil off of N_2, tanks are double walled, vacuum and multilayer insulated. Their rugged design makes them suitable for transportation.

Vapours from the boiling N_2 generate pressure within the tank and therefore no pump is required to transfer liquid N_2.. Each N_2-tank is fitted with a relief valve which cracks at either 152 or 241 kPa (these are the two low pressure N_2-tanks used). The DURA SERIES tank is also equipped with an internal vaporizer that can build pressure within the tank when the influx of heat does not create adequate vapour pressure.

3.2.5. Transfer Line The cold nature of liquid N_2 makes design and construction of the liquid transfer line very important. Minimizing heat influx to the N_2 was of primary importance because N_2 loses approximately half its cooling potential when it changes phase. Vacuum insulated lines are desirable for this reason (Croft 1970). However, for ease of testing and modification to the design, transfer lines were insulated with extruded polystyrene by imbedding them in the centre of a 200 mm block.

Transfer lines should be of low mass to minimize the amount of N_2 required to cool the lines (referred to as flash losses). Thin-walled stainless steel or copper pipe is often used as the inner tube in vacuum insulated lines (Croft 1970). In this study, copper tubing was used.

All copper tubing connections were made by means of brass compression fittings, although a permanent method of connection such as welding or silver soldering is recommended (Croft 1970). Compression fittings facilitate quick and easy design modification. As well, the thermal coefficient of linear expansion is greater for brass than copper (Salisbury 1950) indicating that joints are likely to remain leak-free, even at cryogenic temperatures.

Safety must be exercised when designing transfer lines so as to prevent the possibility of confining liquid N_2 in the line. Liquid N_2 can reach pressures of 100 MPA when warmed to room temperature in a confined space (ASHRAE 1994). This far exceeds the bursting pressure of the thin walled transfer lines. Therefore, a pressure relief valve must be located in any transfer line where the possibility of confining N_2 exists. The transfer line designed for this thesis, was equipped with a relief valve with a cracking pressure of 345 kPa located between the solenoid valve and the N_2-tank valve. As an additional precaution, the selected solenoid valve is designed to open and vent pressure differentials in excess of 690 kPa.

Preliminary testing revealed that copper tubing of 9.5 mm diameter was adequate for the flow of N_2 required. Obviously it is advantageous to design a transfer line of the smallest possible diameter so as to minimize heat infiltration, however design equations for two phase flow tend to be empirical and not applicable to cryogenic fluids. Therefore, optimizing the diameter of the transfer line is a trial and error process and was not done in the designed system.

3.2.6. The Solenoid Valve The ASHRAE handbook (1994) states that flow control of liquid N_2 can be accomplished by either on-off cycling of solenoid actuated valves or by special flow-metering valves. I incorporated the former because of its simplicity and availability. No supplier of cryogenic flow-metering valves was found. As well, injection nozzles may not function adequately under reduced flow caused by a metering valve.

The solenoid valve (Model: 8263G206 LT, ASCOlectric Ltd., Brantford, ON) is an electrically actuated valve suitable for connection to a 9.5 mm transfer line. In the event of a power failure, the valve is normally closed, preventing the uncontrolled release of N_2 into the container.

3.2.7. Simulation of Master Packaged Meat It was not practical to test the performance of the container during the design phase with meat because of its expense. Therefore, a saline solution was prepared and packaged in 36 bags (229 x 559 mm) composed of a single laminate of nylon and polyethylene. The 10% concentration (by mass) of table salt, has a freezing point of −6.6°C (ASHRAE 1993) and possesses similar heat transport characteristics as meat. The thermal conductivity of the solution is 0.50 W/(m•K) while that for lean beef is approximately 0.47 W/(m•K) (ASHRAE 1993). Thermal conductivities of pork are similar despite the fact they are dependant on moisture content, the direction of heat flow, the amount of fat, and the location of bones.

The heat storage capacity of each saline water bag was the same as one master tray of meat (where *heat storage capacity* is defined as the product of specific heat and the total mass of the product). Based on the dimensions of the current master trays, it is possible to package approximately 3.6 kg of meat. Therefore, the mass of each saline water bag was 3.0 kg to achieve the same heat storage capacity as the meat because the specific heats for the saline solution and beef are 3.72 and 3.11 kJ/(kg•K), respectively (ASHRAE 1993). The same reference indicates the specific heat of pork tends to be a bit lower with an average value of 2.71 kJ/(kg•K).

For reasons of expense, polyethylene master trays were not constructed. Instead, 18 pieces of mahogany panelling (3 mm thick) were cut to fit on the stainless steel ledges and to act as shelves. Each shelf supported two saline water bags.

3.2.8. Instrumentation An instrumentation system was assembled for measuring temperatures within the container and monitoring the use of N_2 from the tank. All data were logged to a UNISYS 300 (an 80286 PC) by a QuickBASIC program and stored on the hard drive for further analysis.

Nitrogen use was measured by placing the N_2-tank on a floor scale (Model: 2136, Mettler-Toledo Inc., Burlington, ON). This scale measures masses up to 500 kg at 0.2 kg increments. Through an RS 232 connection, mass data were transferred from the scale to the hard disk by the QuickBASIC program.

For temperature measurement, it was deemed necessary that the instrumentation system be accurate to ±0.1°C (see Appendix C which describes instrumentation error and the calibration procedure) because of the narrow temperature ranges required within the container (i.e., -1.5 ± 0.5°C). This degree of accuracy is almost exclusively obtained by using thermistors. Thermistors (Model: 44007, OMEGA Engineering Inc., Stamford, CT) were purchased.

With an increase in temperature, the resistance of thermistors decreases exponentially and as such, to maintain a high degree of instrumentation accuracy, a good ohm metre must be used. A multimeter (Model: HP 34401A, Hewlett-Packard Co., Loveland, CO) was used to measure resistance because of its accuracy and remote logging capabilities. The multimeter was connected to the computer through a serial port.

The Hewlett Packard multimeter is capable of measuring only one channel of resistance at a time. For this reason, a 24 channel multiplexer was built and connected to the multimeter. Channel activation was computer controlled.

3.2.9. Use of Multiplexer Channels Channel one of the multiplexer was devoted to controlling the solenoid valve. Thus, this channel was never patched through to the multimeter. Channel two was occupied by two thermistors wired in series (called the control sensors). One of these sensors was centrally located in the air space above saline water bags 13 and 14 and the other above bags 15 and 16 (Figure 3.3.). Multiplexer channels three and four were devoted to measuring air temperatures outside the container, and the remaining channels (5 to 24) measured temperatures of the saline water bags. These sensors were located under the midpoint of each bag (between it and the mahogany shelf).

3.3. Developing a Temperature Control Algorithm

3.3.1. The Purpose of the Temperature Control Algorithm The fundamental objective of this thesis was to research and design a refrigerated container which will maintain a product temperature of $-1.5 \pm 0.5°C$. This container may also be used for chilling meat, and therefore the purpose of the temperature control algorithm was two fold: 1) it must chill the product from a higher temperature to $-1.5°C$, and 2) it must maintain the product temperature at $-1.5 \pm 0.5°C$. The control algorithm must regulate temperatures by cycling a solenoid valve, which governs the quantity of cooling medium (N_2) injected into the container.

3.3.2. A Non-intrusive Method of Estimating Product Temperature It is crucial for the control algorithm to accurately approximate the temperature of the stored product so it can regulate injections of N_2. Inserting a temperature probe into the packaging is an unacceptable means of measuring the temperature because it contaminates the meat, destroys the controlled atmosphere, and is labour intensive. Therefore, a non-intrusive method of estimating product temperature is required.

Product temperatures can be predicted if the designed refrigeration system maintains a constant container temperature of $-1.5°C$. In this situation, the temperature of the stored product is expected to converge to that of its surroundings. However,

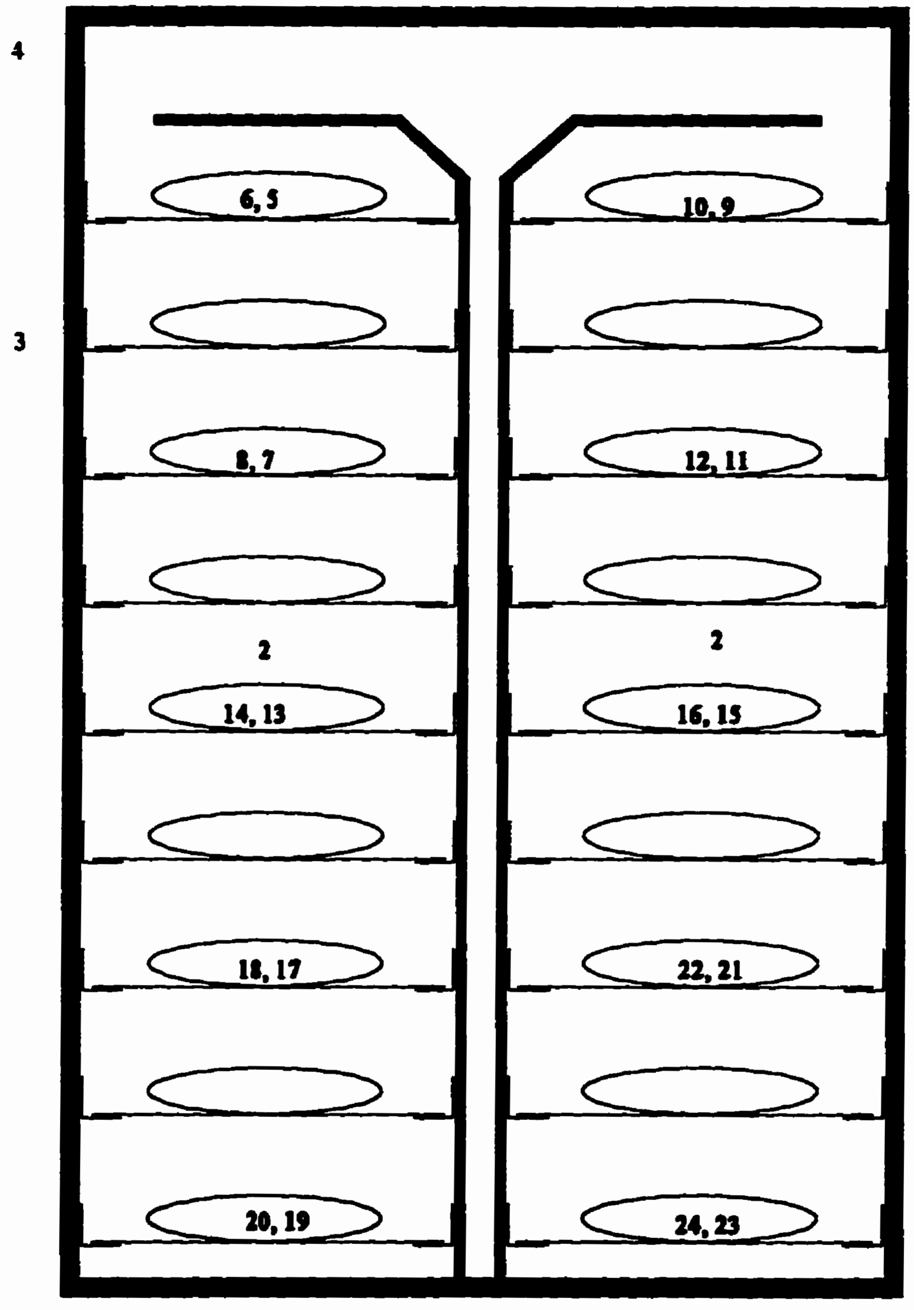


Figure 3.3. **Container equipped with mahogany shelves, water bags, and numbers indicating the location of temperature sensors (odd and even numbers represent sensors at the front and back of the container, respectively. The number 2 indicates the location of control sensors and 3 and 4 indicate the location of outside temperature sensors).**

difficulties arise in employing this ideology to the designed container because temperatures within the container are quite dynamic following an injection of N_2.

Although air temperatures in the container are not constant, the temperature of the product is quite static over short periods of time because of its large specific heat and the low rate of convective heat transfer with the surroundings. If injections of N_2 are quite frequent and result in similar cyclic mean temperatures (where *cyclic mean temperature* refers to the time-weighted average air temperature over the interval between N_2 injections), then the temperature of the product will converge to a steady state value equivalent to the cyclic mean temperature of the surrounding air. When these conditions are met, the control algorithm can reliably estimate the product temperature by a non-intrusive method.

3.3.3. Location of the Control Sensors The control sensors were used to measure the cyclic mean air temperature. To minimize cost, two thermistors were wired in series so as to occupy only one channel of instrumentation (Channel 2). Wiring the control sensors in this manner causes measured temperatures to be averaged.

Temperature variations exist throughout the container and therefore questions are raised about where to locate the control sensors so they best reflect the temperature of the stored product. Testing during the developmental stages indicated that air temperatures throughout the container varied by up to 1.5°C, and often in excess of 0.5°C from the front to back at any one tray level. However, little temperature variation was noticed between sensors located on opposing shelves at the same level. In attempts to reflect the average temperature of the container, the control sensors were centrally located (from top to bottom and from front to back, with one on each side of the air distribution conduit).

3.3.4. Control Sensor Temperatures Following a N_2 Injection Air temperatures inside the container were very dynamic following an injection of N_2 (Figure 3.4.). When the solenoid was energized, N_2 gas entered the container causing the temperature to drop slowly. As time progressed, the phase of the N_2 became predominantly liquid and the temperature plummeted - even 5 to 6 s after the solenoid was de-energized. The lag time reflects two things: 1) the time required for the temperature of the thermistor to equilibrate with the temperature of the surrounding air, and 2) the time required for dispersal of the cooling medium throughout the container. As the injected N_2 mixed with the container air, temperature began to climb linearly due to the heat produced by the fans, heat removed from the stored product, and heat conducted through the container walls.

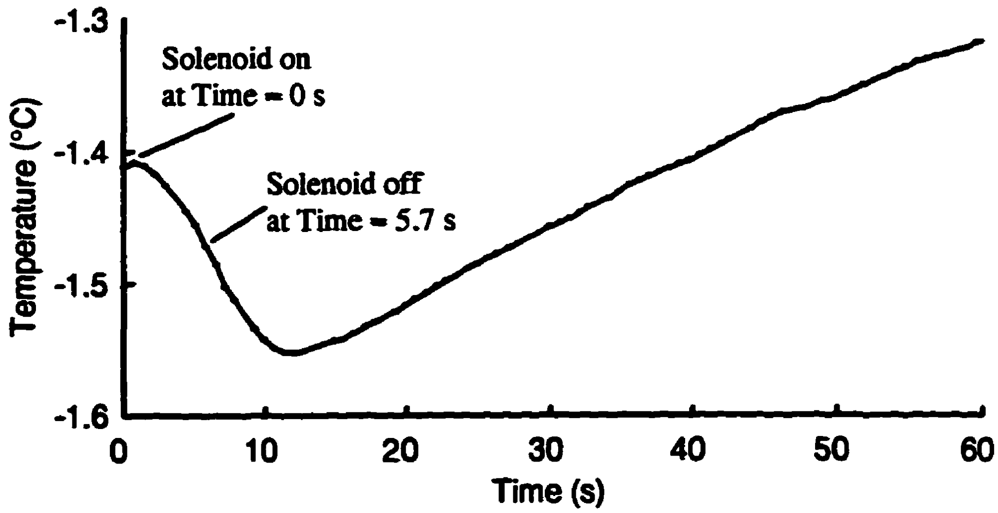


Figure 3.4. Temperature of control sensors following an injection of N_2.

3.3.5. Using Control Sensors to Cycle the Solenoid The key to accurate prediction and maintenance of product temperatures was the ability of the control algorithm to produce constant cyclic mean temperatures with controlled injections of N_2. In the designed container, the control algorithm governs solenoid cycling strictly on temperature data from the control sensors.

When N_2 is injected into the container, control sensor temperatures drop. Because a relationship must exist between the temperature at which the solenoid shuts off during an injection and the resulting cyclic mean temperature, the control algorithm makes a record of both. For future decisions, the control algorithm uses stored information and simple ratios to approximate a shut off temperature for the solenoid that will result in a cyclic mean temperature of -1.5°C.

During preliminary testing, I attempted to control cyclic mean temperatures by timing the length of N_2 injections. This method of temperature control failed because N_2 enters the container in two phases - each phase having different thermodynamic and flow properties. Because it is impossible to know what proportion of N_2 will exist as a gas or liquid at any given time, it is also impossible to predict the cooling potential of a N_2 injection from time measurements.

3.3.6. Regimenting a Fixed-Length, 60 s Time Cycle As previously discussed, two conditions are required by the control algorithm for successful prediction and control of product temperatures. The first is frequent injections of N_2 in order that the product temperature is stable, and the second provision is the consistent reproduction of cyclic mean temperatures equal to -1.5°C. Regimenting a fixed-length, 60 s time cycle between

N_2 injections encouraged the achievement of both goals.

Product temperatures are considered stable during 60 s intervals because heat transfer by convection is not significant enough to cause a change in temperature of the stored meat, which has a high specific heat. Another benefit of a 60 s cycle is the container will require servicing only once a year under continuous operation because the expected operating life of the solenoid is 0.75 to 1×10^6 cycles (K. LeFave, Field Sales Technician, ASCOlectric Ltd., Winnipeg, MB).

A fixed-length time cycle is defined as a time cycle with a constant time period between N_2 injections, whereas a variable-length time cycle may have 60 s between one set of injections and 30 s between the next. A fixed time cycle is required by the control algorithm because shut off temperatures are calculated based on cyclic mean temperatures and these in turn are dependent on the length of the time cycle. Therefore, the control algorithm will not function correctly on a variable-length cycle.

3.3.7. Forty Second Maximum Time Limit for N_2 Injections The solenoid can be energized during the first 40 s of any time cycle. At the end of this period, the solenoid is shut off regardless of whether the target temperature is reached. Two reasons exist for designing the temperature control algorithm this way: 1) continuous operation of the solenoid does not allow for temperature equalisation and results in an extreme range of temperatures throughout the container, and 2) approximately 20 s of each cycle are required by the instrumentation system to measure the temperature of the saline water bags within the container. In a commercial system, product temperatures would not be measured, but it is required in this instance to evaluate the performance of the container.

3.3.8. Complete Description of the Temperature Control Algorithm The reasoning and logic behind individual sections of the control algorithm have been discussed. What follows, is a complete description of how these sections function together within the control algorithm. The complete QuickBASIC program is included in Appendix D.

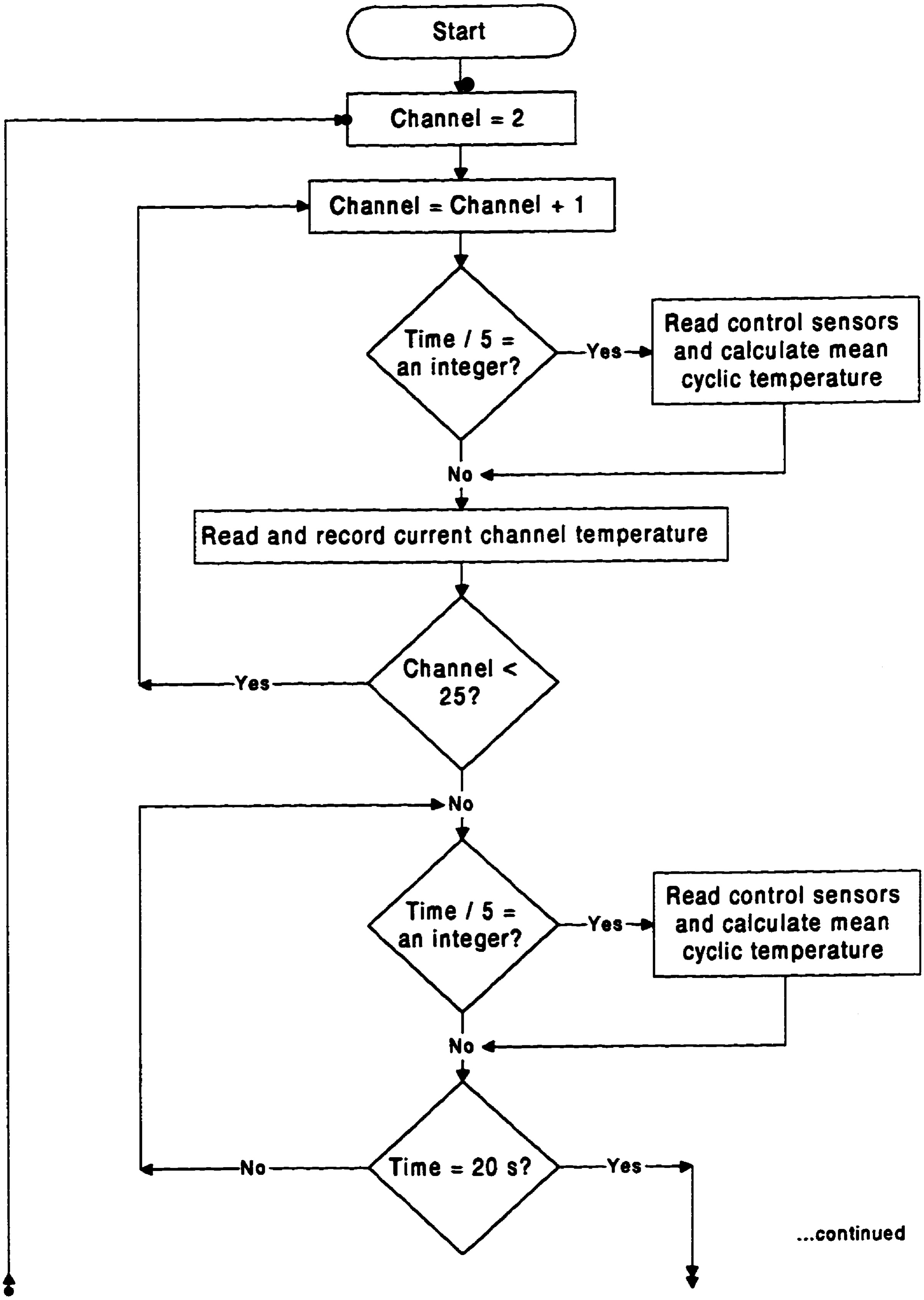
Many actions of the control algorithm are triggered by the clock (Figure 3.5.). The first 20 s of each cycle were reserved for measuring product temperatures, with the control sensors read once every 5 s (to determine the cyclic mean temperature). At Time = 20 s, the mean cyclic temperature was reset and calculation of a new mean began. A shut off temperature for the solenoid was also determined. If the control sensors measured a temperature greater than the shut off temperature, the solenoid was energized. Control sensor readings then became continuous as the control algorithm sought to de-energize the solenoid at the moment the shut off temperature was ascertained. As well, every 5 s a reading from the control sensor was used in contribution to the mean cyclic temperature. When the shut off temperature was achieved, the solenoid was de-energized and the temperature was stored for future reference in control decisions. At Time = 60 s the solenoid was automatically de-energized and a new cycle started.

In calculating the shut off temperature for the solenoid valve, the control algorithm used data from three or six previous temperature cycles. The control algorithm stored the ratio of the solenoid shut off temperature to the resulting cyclic mean temperature. Averaging the value of these ratios from the three or six previous cycles and using simple ratios, the algorithm calculated a shut off temperature in attempts to achieve a mean cyclic temperature of −1.5°C for the following injection of N_2.

3.4. Finite Element Model

A transient heat conduction and convection model was constructed by adapting the three dimensional model of Alagusundaram (1989) to a two dimensional model. A computer program was written in FORTRAN (see Appendix E for the program listing) and compiled on a UNIX system to solve problems discretized into linear triangular elements. To ensure correct functioning of the program, a four-element sample problem was created and solved by hand and the results were compared with the output of the program.

A model to simulate heat flow from the saline water bags was developed. The mesh consisted of 228 nodes and 382 elements (see Figure E.1. in Appendix E). As well, a model of master packaged meat was developed with a mesh consisting of 555 nodes and 1008 elements (see Figure E.2. in Appendix E).


Start
Channel = 2
Channel = Channel + 1
Time / 5 = an integer?
Yes
Read control sensors and calculate mean cyclic temperature
No
Read and record current channel temperature
Channel < 25?
Yes
No
Time / 5 = an integer?
Yes
Read control sensors and calculate mean cyclic temperature
No
Time = 20 s?
No
Yes
...continued

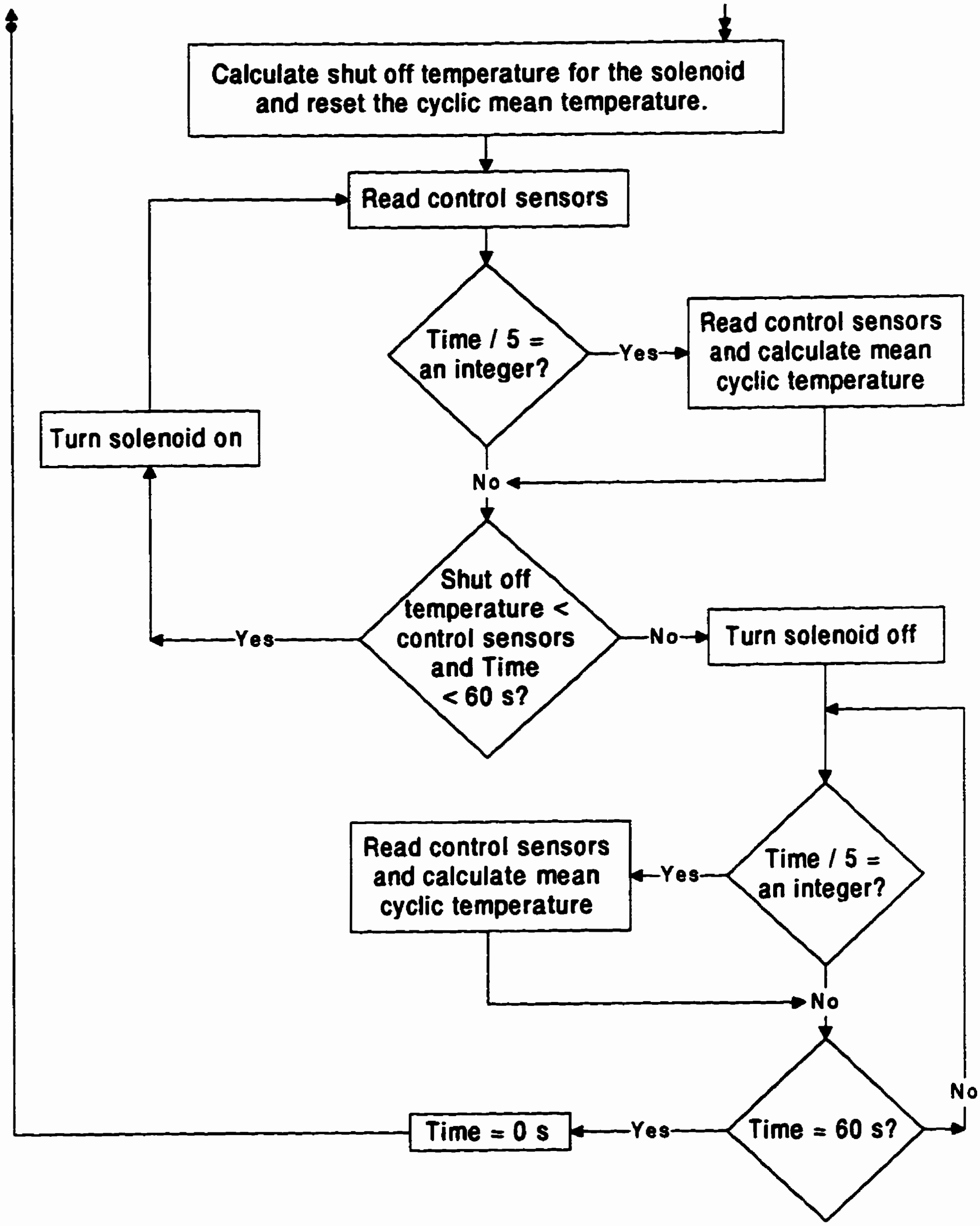


Figure 3.5. Flow chart of the temperature control algorithm.

4. EXPERIMENTAL METHOD

4.1. Tests to Evaluate the Container

For this design thesis, the purpose of testing was to determine the feasibility of the prototype - more specifically, the ability of it to maintain design temperatures of -1.5 ± 0.5°C and a controlled N_2 atmosphere. Because the feasibility of the container will be heavily influenced by economics, it was necessary to measure N_2 consumption so the cost and benefits of this type of refrigeration can be assessed. Tests of a practical nature were also necessary. They included determining the effects on container temperatures when the door was opened, or when a N_2 tank ran dry, or when the control algorithm and solenoid experienced a power failure.

Testing was also required to determine the air flow needed in the container to maintain appropriate product temperatures and to cool the stored product in a suitable fashion. Theoretical calculations of this parameter are complex, if not impossible because the stored product was situated in an area of transitional and underdeveloped air flow.

4.2. Determining the Thermal Resistance of the Container

The thermal resistance of any insulating wall can be determined by exposing one wall to a known heat source and recording the temperatures on both sides of it. The four circulation fans were used as the heat source and their true root mean square (RMS) power usage was measured with a power meter (Model: 2101, Valhalla Scientific Inc., San Diego, CA). Ten temperature sensors were spaced around the inside and outside of the container (no saline water bags were in the container during these tests).

All testing of the container was conducted within a controlled environment chamber (Model: CONVIRON C1010, Controlled Environments Ltd., Winnipeg, MB), which is capable of controlling temperatures to within ±0.5°C in the temperature range from −20 to 50°C. The CONVIRON chamber was set to −15, 0, and 15°C for three separate tests. Temperatures in and around the container were monitored and collected until a steady-state temperature gradient existed between the outside and inside of the container.

4.3. Temperature and N_2 Use Tests

The CONVIRON chamber was used to cool the container and saline water bags to an initial temperature of 10 ± 1.5°C. This was the maximum temperature expected of freshly butchered and packaged meat entering the container. Once this temperature was achieved, the door to the container was closed and the environmental chamber was set to the desired operating temperature. When this was achieved (it usually took less than 0.5 h), the valve to the N_2 tank was opened, and the computer program was started. Each test lasted 8 h, with time, temperature, and N_2 use data recorded by the computer. With four circulation fans, three replicates were completed for outside temperatures of −15, 0, and 15°C. As well, three replicates each were completed for tests with two, four, and six fans at an outside temperature of 30°C.

The first replicate used Program1 (Appendix D) as the controlling program. This program used temperature histories from three previous solenoid cycles to make solenoid cycling decisions. Second and third replicates were completed using Program2 (Appendix D), which used the six previous temperature histories to control the solenoid.

4.4. Failure Testing

4.4.1. Nitrogen Failure Using the CONVIRON chamber, the temperature of the saline water bags was brought to a temperature of -1.5 ± 1.5°C. Once this temperature was achieved, doors to the container were closed and the temperature control program started. The temperature of the environmental chamber was then ramped up to 30°C and maintained at this temperature. When the computer program had ran for a total time of 0.5 h, the valve to the N_2 tank was closed. Temperatures of the stored saline bags were recorded for the next 7.5 h. Three replicates of this test were completed.

4.4.2. Power Failure This test was set up and conducted in a similar manner to the tests which experienced a N_2 failure. The only difference between the two tests was at Time = 0.5 h; instead of closing the tank valve, the power was cut to both the fans and solenoid. Three replicates of this test were completed as well.

4.5. Container Door Openings

Using the CONVIRON chamber, the temperatures of the water bags and container were brought to a temperature of -1.5 ± 1.5°C. Program2 was then started and allowed to establish a temperature equilibrium, while the chamber temperature was ramped up to 21°C. Once temperature equilibriums were established, the door of the container was opened to 90° from its original position. All door openings occurred 40 s into the time cycle i.e. 20 s before the computer collected the temperature data of the saline water solutions. The door remained open for a time interval of 15 s. Three replicates of this test were completed, each allowing approximately 0.5 h between tests for a temperature equilibrium to re-establish. In the same manner, testing was completed for door openings of 25 and 35 s. The time intervals 15, 25, and 35 s correspond to the approximate times required to remove one, two, and three master packages from the container, respectively.

4.6. Atmospheric Composition Testing

All gas samples were withdrawn through a 4 mm diameter plastic tube, which was centrally located along the right hand side wall of the container (as viewed from Figure 3.3). Gas samples were analysed with a HP 5890A gas chromatograph and a HP 3396 integrator (Hewlett-Packard, Avondale, PA) using helium as the carrier gas. The gas chromatograph was calibrated at two points with pure N_2 gas obtained from the N_2-tank and with a standard gas source consisting of 20.0% O_2 and 19.9% N_2. The calibration was confirmed at the beginning and ending of each replicate by analysing gas samples from the ambient atmosphere (these samples were expected to be 20.9% O_2 and 78.0% N_2 (Cameron 1989)).

To start gas sampling tests, the saline water bags were brought to a temperature of 10 ± 1.5°C using the CONVIRON chamber. Once this temperature was achieved, the door to the container was closed while the door to the environmental chamber was left ajar for the remainder of the test (this was done to ensure that the O_2 concentration did not drop below 20.9% within the CONVIRON chamber). Once the chamber reached a room temperature of 21 ± 2°C, the valve on the N_2 tank was opened, and the computer program started. Two gas samples were withdrawn and analysed every 15 min for the 8 h duration of the test. Temperature and N_2 use data were also collected for the three replicates of this test. The first replicate used Program1 to control solenoid cycling, and the second and third replicate used Program2 (Appendix D).

4.7. Modelling

4.7.1. Determining a Convective Heat Transfer Coefficient Typically, finite element models are used to predict temperatures within an object based on a knowledge of pertinent thermal characteristics. However, in this thesis temperatures of saline water bags were determined experimentally and knowledge of the convective heat transfer coefficient was desired. To determine it, an inverse heat transfer method of finite element modelling was used (Weres *et al.* 1994). Various convection coefficients (increments of 5 W/(m^2•K)) were tested in the finite element model of saline water bags in attempts to minimize the objective function:

$$S = \sum_{i=1}^{NT} \left[T_{pred} - T_{\exp}\right]^2 \tag{4.1}$$

where:

S	= objective function	NT	= number of time steps
T_{pred}	= predicted temperature (K)	T_{exp}	= experimental temperature (K)

Control sensor readings (location 2 in Figure 3.3) from previously conducted experiments, were used as inputs to the finite element model for convection fluid temperatures. As well, sensors 13 through 15 (these water bags were closest in proximity to the control sensors) were averaged and used as the basis of comparing experimental and predicted temperatures.

Convection coefficients were determined from experimental data collected for two and six fans, with a temperature outside the container of 30°C (two trials each). As well, coefficients were found from test data for four fans and outside temperatures of 30, 15, 0, and −15°C (one trial each).

4.7.2. Modelling the Master Package Both the minimum and maximum convection coefficient determined for saline water bags were substituted into the finite element model for master packaged meat. Although convection coefficients are determined by boundary layer development around an object (and this layer is influenced by the object's surface geometry), coefficients determined for the saline water bags were used to approximate temperatures within master packaged meat stored in the container.

5. RESULTS

5.1. Thermal Resistance of the Container

5.1.1. Experimental Results and Discussion To ensure an equilibrium temperature was attained by the container, each of the tests was run for an extended period of time (the minimum was 30 h). Testing was terminated when a steady-state temperature had been maintained for a minimum of 10 h. Data were imported into a Quattro Pro spreadsheet where the average temperature difference between the inside and outside of the container was determined. The last 10 h of data from each of three tests were graphed (Figure 5.1).

Immediately obvious in Figure 5.1. are large temperature spikes for tests conducted at outside temperatures of −15 and 0°C. These result every 3 h because the CONVIRON chamber must defrost its evaporating coils. Despite this, a constant temperature difference occurs between the interior and exterior of the container during all three trials. The average temperature differences for the container exposed to outside temperatures of 15, 0, and −15°C were 16.6, 17.7 and 16.5°C, respectively. This equates to an average temperature difference of 16.9°C for all three tests.

Variations among the three temperature differences may be explained by the fluctuating power consumption of the fans. Four fans consumed anywhere from 119 to 126 W of power depending when measurements were taken. Power usually fluctuated 2 to 3 W during any given reading. The average power consumption by the fans was approximated to 123 W.

The thermal resistance of the container can be determined from the above experimental data by using the following equation:

$$R_{total} = \frac{\Delta T}{\dot{q}} \tag{5.1}$$

where:

R_{total} = thermal resistance of container (K/W) ΔT = temperature difference (K)
$\dot{q}$ = source of heat (W)

For the container, the total thermal resistance was found to be 137×10^{-3} K/W.

5.1.2. Explanation of Theoretical and Experimental Results The theoretical resistance value of 181×10^{-3} K/W (Appendix F) differed from the measured value by 32%. A variety of errors may have contributed to this difference. As previously mentioned, power usage of the fans had an uncertainty. Another contribution to the error was an inexact knowledge of the thermal conductivities of the polyethylene shell and polystyrene insulation because of differing densities and grades of materials. Also, as mentioned in Appendix F, the thickness of the wall varied considerably. However, I believe the majority of the difference results because theoretical calculations were based on a *clear wall*. In reality, the wall is fitted with frames to maintain its rigidity and to mount the door. As well, theoretical calculations do not consider interface details (wall/door, roof/door, and floor/door), which are suspected of being areas of heat leak.

5.2. Analysis and Discussion of Temperature and N_2-Use Data

5.2.1. How Data were Analysed The output files from Program1 and Program2 (Appendix D) were similar. Every 60 s, each program recorded the time, the current mass of the N_2 tank, and the temperatures of sensors 2 through 24. Data files were analysed by importing them into Quattro Pro, where for each time increment the minimum, maximum,

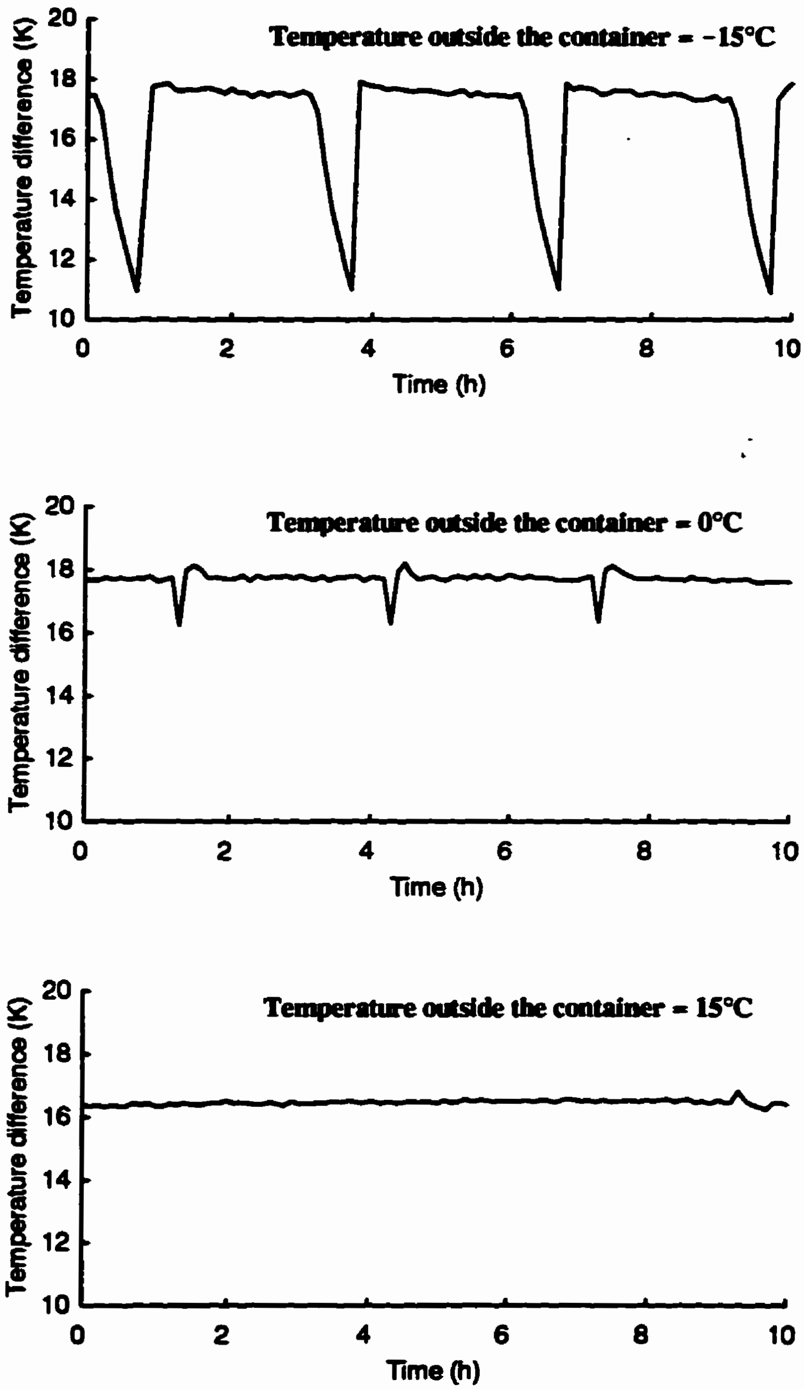


Figure 5.1. The temperature difference between the inside and outside of the container when the heat of four fans causes the interior temperature to equilibrate at an elevated value.

and average temperatures of sensors 5 through 24 were determined as well as the total consumption of N_2. These parameters were graphed and included in Appendix G (Graphs G.1. to G.36.).

The saline water bags reached a temperature equilibrium by approximately 5.5 h into testing (Appendix G). Therefore, the overall minimum, maximum, and average temperatures of all saline water bags were determined for the time period between 5.5 to 8 h (Table 5.1.). The locations of the warmest and the coolest temperature sensors were also recorded.

Nitrogen usage was considered to be of two forms - fixed and variable. The term *fixed usage* refers to that quantity of N_2 required to cool the contents of the container (e.g. the steel shelving unit, the plastic walls of the container, the plastic walls of the air distribution duct, and the saline water bags) to its operating temperature. It is a one time input of N_2 and was thus referred to as fixed. *Variable N_2 usage* removes heat conducted into the container and transfer line, and heat produced by the fans. This N_2 is needed on a continuous basis to maintain the operating temperature of the container and was thus referred to as variable usage.

Experimental data (N_2 use data) were analysed by separating it into both fixed and variable forms of usage. This was done by performing a linear regression on the last 2.5 h of data. During this time period, N_2 use was constant and thus the slope of the line represents variable usage and the y-intercept represents fixed usage. Table 5.2. shows the experimental and theoretical values (procedure to calculate theoretical values is given in Appendix H) for fixed and variable N_2 usage.

5.2.2. Temperature Data in Relation to the Number of Fans Using two or six circulation fans within the container did not maintain temperature at −1.5 ± 0.5°C and temperatures climbed to −0.2 and −0.3°C, respectively (Table 5.1.). However, a four fan combination resulted in temperature control which was only slightly outside of the desired temperature range. In three trials at 30°C (the worst case scenario), temperatures ranged from a low of −2.0°C to a high of −0.7°C (Table 5.1.).

It was unexpected that four fans would outperform six because I thought greater air flows would mean better removal of heat (from the walls), and therefore minimize its influence on the stored product. In attempts to explain this anomaly, Appendix I explains how air velocities were measured at the distribution duct with combinations of two, four, and six fans. In general, measured velocities were progressively greater in the order of two, four, and six fans (as expected). As well, all fan combinations demonstrated uneven air flow at the top of the duct; this was probably the result of a turbulent region immediately below the fans. However, the air velocity data do not suggest any reason why four fans outperformed six. In fact, in the regions of temperature sensor 20 and 23 (the coolest and warmest locations), six fans had a high and uniform velocity distribution (which should have been ideal).

Another reason six fans may not have performed as well as four was because of uneven distribution of the cooling medium i.e. fans at the back of the conduit may have circulated more of the cool N_2 gas than fans at the front. However, there was little difference between average temperatures for six and four fan combinations (Table 5.3). Therefore, it is suspected that both fan combinations distributed the cooling medium just

Table 5.1. **The minimum, maximum, and average temperatures of saline bags within the container, and the warmest and coldest sensor location.**

Number of fans	Trial number	From Time = 5.5 to 8.0 h				
		Temperature			Sensor locations[a]	
		Min (°C)	Avg (°C)	Max (°C)	Coldest	Warmest
		Temperature outside the container = 30°C				
2	1	-2.3	-1.4	-0.4	20	23
	2	-2.6	-1.5	-0.3	20	23
	3	-2.5	-1.5	-0.2	20	23
4	1	-1.9	-1.4	-0.8	20	7, 23
	2	-2.0	-1.4	-0.7	20	23
	3	-1.9	-1.4	-0.7	20	23
6	1	-1.9	-1.4	-0.3	20	23
	2	-2.0	-1.4	-0.6	20	23
	3	-1.9	-1.4	-0.6	20	23
		Temperature outside the container = 15°C				
4	1	-1.8	-1.4	-0.9	20	7, 23
	2	-1.9	-1.4	-0.9	20	23
	3	-1.9	-1.4	-0.8	20	23
		Temperature outside the container = 0°C				
4	1	-1.7	-1.5	-1.2	20, 22, 24	7, 13, 23
	2	-1.9	-1.5	-1.2	20	5, 7, 23
	3	-1.8	-1.5	-1.1	20, 22, 24	5, 7, 23
		Temperature outside the container = -15°C				
4	1	-1.7	-1.5	-1.4	20, 22, 24	5, 7, 11, 13, 15, 17
	2	-1.9	-1.7	-1.4	20, 22, 24	7, 13, 15
	3	-1.7	-1.5	-1.3	20, 22, 24	5, 7, 13, 17, 19, 23

[a]For sensor location numbers refer to Figure 3.3.

Table 5.2. Experimental and theoretical values of fixed and variable N_2 usage.

Number of fans	Trial number	Fixed usage of N_2 (kg)		Variable usage of N_2 (kg/h)	
		Experimental	Theoretical [b]	Experimental	Theoretical [c]
		Temperature outside the container = 30°C			
2	1	19.7	17.7	4.5	3.0
	2	18.6	17.5	5.5	3.0
	3 [a]	16.7	17.4	5.0	2.9
4	1	18.8	18.3	5.3	3.6
	2	22.1	16.7	5.6	3.6
	3	20.1	18.1	5.7	3.6
6	1	16.0	18.5	5.9	4.1
	2 [a]	13.7	18.3	6.3	4.1
	3	22.6	19.7	6.7	4.1
		Temperature outside the container = 15°C			
4	1	18.2	17.7	3.8	2.5
	2	18.9	19.6	3.8	2.5
	3 [a]	20.9	18.4	4.4	2.5
		Temperature outside the container = 0°C			
4	1	17.9	17.5	2.4	1.4
	2	22.3	18.9	2.6	1.4
	3 [a]	21.2	17.9	2.8	1.4
		Temperature outside the container = −15°C			
4	1	23.3	17.8	0.5	0.09
	2	22.5	18.1	1.2	0.11
	3	22.8	17.7	1.1	0.10

[a] liquid N_2 storage tank at 152 kPa instead of the usual 241 kPa.

[b] data from Table H.1. in Appendix H.

[c] data from Table H.2. in Appendix H.

as effectively and it is unknown why four circulation fans were the best combination.

Table 5.3. Minimum, maximum, and average temperatures of the saline water bags at the front and back of the container resulting when two, four, and six circulation fans were used and the container was exposed to an outside temperature of 30°C.

Number of fans	Trial number	Temperature					
		Front of container			Back of container		
		Min (°C)	Avg (°C)	Max (°C)	Min (°C)	Avg (°C)	Max (°C)
2	1	-1.4	-1.0	-0.4	-2.3	-1.8	-1.5
	2	-1.3	-1.0	-0.3	-2.6	-2.0	-1.8
	3	-1.4	-0.9	-0.2	-2.5	-2.0	-1.7
4	1	-1.3	-1.1	-0.8	-1.9	-1.6	-1.4
	2	-1.3	-1.1	-0.7	-2.0	-1.7	-1.5
	3	-1.3	-1.1	-0.7	-1.9	-1.7	-1.4
6	1	-1.3	-1.0	-0.3	-1.9	-1.7	-1.5
	2	-1.3	-1.1	-0.6	-2.0	-1.7	-1.6
	3	-1.4	-1.2	-0.6	-1.9	-1.7	-1.5

5.2.3. Temperature Data in Relation to Location The sensor location 23 was consistently one of the warmest locations while sensor 20 was the coolest (Table 5.1.). Both these sensors measure the temperature of saline water bags on the bottom shelf of the container (Figure 3.3.). Sensor 20 was located at the back of the container while sensor 23 was located at the front. These results were consistent with Table 5.3. which indicates that bags located towards the front of the container were on average warmer than their counterparts at the back. The average temperature differences between bags located at the front and the back were 0.9, 0.6, and 0.6°C for two, four, and six fans, respectively. Therefore, the temperature ranges experienced at each shelf level (from front to back) account for a large portion of the temperature range experienced between all 20 sensors. As well, data indicate there were little difference in the temperature of sensors located on opposing sides of the air distribution conduit.

5.2.4. Temperature Data in Relation to Outside Temperature The range of temperatures inside the container progressively narrows with dropping temperatures outside the container. This would seem to indicate that conduction of heat into the container was a major factor influencing product temperatures. Although temperatures may be more uniform within the container at lower test temperatures, it is still evident that the front of the container experiences warmer temperatures when compared to the back regions. This may be explained by the increased wall thickness at the back of the container and therefore greater resistance to heat conduction.

It is also noteworthy that the container can operate at temperatures well below zero and still not risk freezing the stored product because heat produced by the circulation fans was able to warm the container (see Appendix H for more details).

5.2.5. Nitrogen Data - Fixed Usage Fixed usage of N_2 was expected to be constant throughout all testing because of similar starting temperatures ($\approx 10°C$) and the same target temperature ($-1.5°C$). Equations derived in Appendix H explain why fixed usage of N_2 was not related to temperatures outside the container or to the number of fans used.

Table 5.2. shows the experimental and theoretical values of fixed N_2 usage during testing. As expected, the measured fixed usage was approximately equivalent during all tests, with an average of 20.3 and 18.1 kg used from N_2 tanks at 241 and 152 kPa, respectively. The corresponding theoretical values were 18.1 and 18.0 kg, which compares favourably with the experimental values. However, two notable exceptions

occurred during testing where N_2 usage was well below that predicted. Two trials with six fans had a fixed usage of 13.7 and 16.0 kg. It is unknown why these particular tests behaved differently from the majority.

Because of differences in the thermodynamic properties of N_2 stored at 241 and 152 kPa, it was expected that fixed N_2 usage would decrease with tests performed at tank pressures of the latter. However, whether this was the case or not is impossible to determine with just the four trials conducted at tank pressures of 152 kPa.

5.2.6. Nitrogen Data - Variable Usage Table 5.2. also displays the variable usage of N_2. Unlike fixed usage, variable usage was expected to increase from one test to the next with increasing outside temperatures and number of fans (see Appendix H). This trend is quite evident from the data. However, predicted variable usage was considerably lower than that experienced during testing. In attempts to locate the source of the difference, Table 5.4. was compiled from data in Table 5.2. and Table H.2. (Appendix H).

By observing how N_2 usage changes from test to test, it was possible to determine the effect of individual parameters. For instance, N_2 consumption was reduced 0.8 kg/h when the number of fans was decreased from six to four (Table 5.4.). Another reduction of 0.5 kg/h occurred from four to two fans. The only parameter change between testing was the number of fans and therefore it can be concluded that two fans consume on average 0.65 kg/h. This compares favourably with the theoretical value of 0.6 kg/h. Therefore, the large difference between predicted and actual usage of N_2 in Table 5.2. was not caused by errors in calculating the N_2 consumption per fan.

Table 5.4. Theoretical and experimental differences between variable N_2 usage when the number of fans and the outside temperature was varied.

Description of tests compared	Reduction in variable N_2 use [a] (kg/h)	
	Experimental [b]	Theoretical [b]
Tests with six and four fans (30°C outside)	0.8	0.6
Tests with four and two fans (30°C outside)	0.5	0.6
Tests at 30 and 15°C (4 fans)	1.5	1.1
Tests at 15 and 0°C (4 fans)	1.4	1.1
Tests at 0 and −15°C (4 fans)	1.7	1.3

[a] the reduction in N_2 use from six, to four, to two fans was determined when the temperature outside the container was held constant; and the reduction in N_2 use was determined when the temperature outside the container was reduced from 30, to 15, to 0, to −15°C with the constant use of four fans.
[b] all three trials of experimental and theoretical data were averaged to determine the change in N_2 usage between tests, despite different source pressures of N_2 (which should make only 2% difference in usage).

Change in N_2 usage between tests at different temperatures (with four fans) indicate how the temperature outside the container affected conduction of heat into the container and the transfer line. From Table 5.4., it is evident that a large difference exists between measured and predicted values. Unfortunately, there was no conclusive way of determining whether this difference was attributable to the conduction equations for the transfer line or the container. However, being as the resistance value of the container was determined experimentally (Section 5.1.), little error was expected in the theoretical heat conduction equation. By the process of elimination, it can be said that either difference arises from calculations of heat flow to the transfer line or an unknown variable exists.

A number of assumptions were made in calculating the theoretical heat flow into the transfer line. For instance, the insulation was considered to be 200 mm in diameter, but during testing frost often forced a gap open between the sheets of polystyrene and no doubt caused increased heat flow to the transfer line. As well, the solenoid and pressure relief valve were situated on the transfer line, encouraging greater heat flow through poorer insulated areas. When energized, the solenoid valve produces 16.7 W of heat, and a portion of it must find its way into the transfer line. Any one of these factors could be a major source of error in predicting heat flow to the transfer line.

It is also hypothesized that a hidden variable exists causing difference between predicted and measured usage of N_2. It is known that a large upper portion of the N_2 storage tank frosted over during N_2 use. Although the tanks themselves were well insulated, evidently the transfer line out of the tank was not. This was difficult to account for in calculations. As well, the cooling properties of liquid N_2 are closely related to its storage pressure and it was questionable how accurately the tanks maintained intended pressures of either 152 or 241 kPa. Tank pressure gages often indicated pressures different than these. Whether this was a malfunction of the gage or pressure relief valve is unknown.

5.3. Analysis and Discussion of Failure Data

Temperature data were collected every 60 s for a period of 8 h during the power and N_2 failure tests. Minimum, maximum, and average temperatures were calculated and graphed using a Quattro Pro spreadsheet (see Graphs G.37. to G.42. in Appendix G). Linear regressions were performed on the average temperatures of the water bags from time 0.5 to 8 h (Table 5.5.) to determine the rate of temperature increase following a failure.

Temperatures increased at a greater rate during a N_2 failure than a power failure

(Table 5.5). This was caused by the heat from the fans, which functioned during a N_2 failure but not during a power failure.

Table 5.5. Average rate of saline water bag temperature increase following a failure (container was exposed to an outside temperature of 30°C).

Trial number	Average rate of temperature increase (°C/h)	
	Power failure	Nitrogen Failure
1	0.9	1.9
2	0.9	2.0
3	0.9	2.0

During a N_2 failure, the temperature range of the stored product was narrow (less than 1°C) whereas a power failure caused temperature stratification within the container i.e. the top shelf reached temperatures over 5°C warmer than the bottom shelf at the end of the 8 h test.

5.4. Analysis and Discussion of Door Opening Tests

Temperature data for the door opening tests were collected by Program2 and analysed in Quattro Pro to determine the minimum, maximum, and average temperatures of the saline water bags at each 60 s time interval. The results were graphed and included in Appendix G (Graphs G.43. to G.45.).

The reason for conducting these tests was to ensure that the temperature of the stored product did not warm significantly for an extended period of time during a door opening. In all cases, the temperature of the stored product returned to its equilibrium temperature within 5 min of the door being closed.

5.5. Analysis and Discussion of O_2 Concentrations

Oxygen and N_2 concentrations of container gas samples were manually recorded from the HP integrator and transferred into a Quattro Pro spreadsheet. The composition of the two samples was then averaged and graphed for each of the three trials (See Graph G.46. in Appendix G).

The three trials resulted in very similar atmospheric compositions. The lowest O_2 concentration reached was 1.7%. This was obtained during the first hour of operation when large quantities of N_2 were used to cool the container and its contents. By approximately 3 h into testing, the O_2 concentration levelled off at between 13 to 15%. The N_2 usage (even numbered Figures from G.2. to G.36.) was also constant by approximately 3 h into testing.

5.6. Program1 vs. Program2

After the first replicate of temperature tests, I noticed N_2 was not always injected once per minute. As mentioned previously, a fixed length, 60 s time cycle is crucial to the ability of the control program to accurately control temperatures. Therefore, Program2 was established. Its control algorithm differs from Program1 in that it considers the temperature history of six, instead of three previous time cycles when making solenoid cycling decisions. As well, it forces the solenoid valve to be open for a minimum of 3 s during any one minute.

Program2 resulted in slightly tighter temperature control than Program1 (Figure G.7. and G.9. in Appendix G). Figure G.7. is a graph of temperatures collected from the container when solenoid cycling was under the control of Program1. The minimum, maximum, and average temperature lines were wavy because the solenoid was not cycling every 60 s as it was designed to do; when compared to Figure G.9., where solenoid cycling was controlled by Program2, temperatures were less wavy because injections of N_2 occurred once per minute.

5.7. Discussion of Finite Element Testing

Approximate convection coefficients were determined from experimental data for the saline water bags. Table 5.6. is a summary of convection coefficients determined and Graphs E.3 to E.10 (in Appendix E) indicate how well the predicted and actual temperatures correlate. Graphs of relative error in Appendix E were determined according the formula of Weres *et al.* (1994):

$$Relative\ error(i) = 100 \times \frac{|\ T_{pred}(i) - T_{\exp}(i)\ |}{T_{\exp}(i)} \qquad i = 1,...,NT$$

where:

T_{pred}	= predicted temperature (K)	T_{exp}	= experimental temperature (K)
i	= the number of the iteration	NT	= total number of time steps

The relative errors experienced were high at the beginning of each test, but subsequently dropped as time progressed. A similar pattern was observed by Weres *et al.* (1994).

There was an increase in convection coefficients with an increase in the number of fans (Table 5.6.), because of an increase in gas velocities around the saline bags. The temperatures outside of the container had no influence on the convection coefficient (which was expected). The magnitude of the convection coefficients was considerably greater than theory would predict. However, theoretical equations have short falls in solving problems of this nature because the package was situated in an area of transitional flow immediately next to an air jet. Although equations exist to solve problems where the product is exposed to an impinging jet, no equations were found to predict heat transfer from a jet which forces the air parallel to the surface of the object. Therefore, theoretical calculations were extremely limited in this instance.

Table 5.6. Estimated convective heat transfer coefficients for saline water bags.

Number of fans	Trial number	Estimated convection coefficient $W/(m^2 \bullet K)$
2[a]	1	95
	2	85
4[b]	1	130
	2	120
	3	125
	4	125
6[a]	1	145
	2	150

[a] during both trials the container was exposed to an outside temperature of 30°C.
[b] the temperatures outside the container were 30, 15, 0, −15°C for trials 1, 2, 3, and 4, respectively.

Convection coefficients of 80 and 155 $W/(m^2 \bullet K)$ were input into the master packaged meat model to determine the temperature at the midpoint of the meat over time. These convection coefficients were to represent the extremes of using either two or six circulation fans within the container. Figure 5.2. indicates that almost no temperature difference exists between the two simulations, implying that conduction is the limiting factor to heat transfer for master packaged meat.

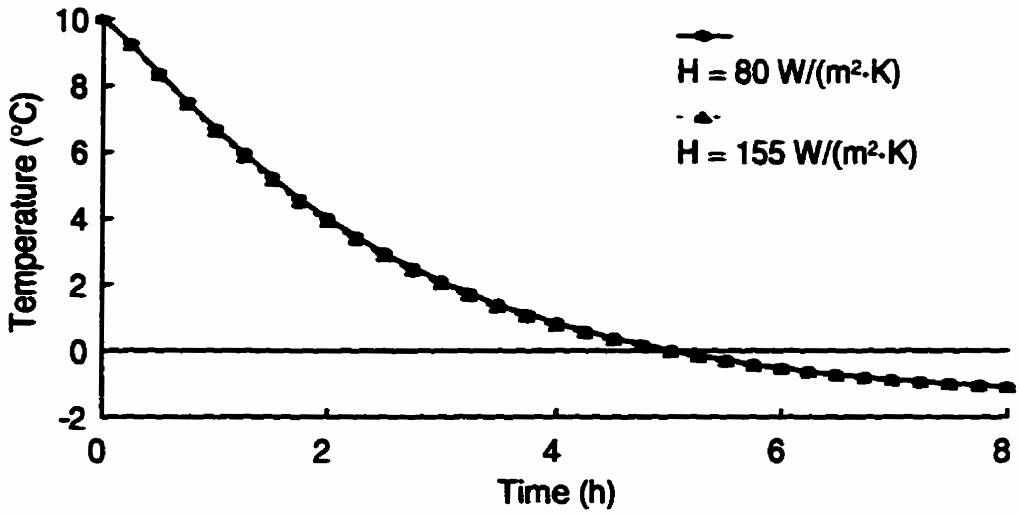


Figure 5.2. **The effects of two convection coefficients on the midpoint temperature of master packaged meat.**

6. CONCLUSIONS

The following conclusions can be drawn from this study:

1. Cooling of the container with time-pulsed injections of liquid N_2 was a successful and reliable means of refrigeration. As well, the shelving unit and air distribution system proved effective.

2. Temperature control was most effective by relating the control sensor temperature at the point in time when the solenoid was de-energized, to the resulting cyclic mean temperature of the container. Through a history of such information and simple ratios, the control algorithm made solenoid cycling decisions for future N_2 injections.

3. The container operated slightly outside the temperature range of -1.5 ± 0.5°C. The four fan combination resulted in the best temperature control with a minimum and maximum product temperature of -2.0, and -0.7°C respectively.

4. Temperature of the stored product was most influenced by the amount of air flow within the container and conduction of heat through the walls.

5. Measured variable N_2 consumption was considerably higher than the theoretically calculated value. Using a process of elimination, the difference resulted from a poor knowledge of the heat flow into the transfer line and the N_2 tank.

6. Convection coefficients between circulated air and saline water bags can be determined using an inverse, heat transfer, finite element model.

7. Simulated heat transfer within master packaged meat showed that heat flow into the master packaged meat was limited by conduction rather than convection.

7. RECOMMENDATIONS FOR FUTURE WORK

There were two main goals in conducting this research. The first was to collect data from the container to facilitate a financial assessment of its merits. Necessary data are now available to enable future researchers to make judgements regarding whether the cost and benefits of such a refrigeration system warrant further research. The second goal was to collect data on the operating characteristics of this container and to determine fundamental thermal properties necessary for future design work. Some recommendations for design improvements are:

1. The transfer line was cause for considerable inefficiency in N_2 consumption. Two modifications will improve its performance: a) Move the solenoid and relief valve from the transfer line to the mixing chamber inside the container so insulation of the transfer line will be more effective. Note: During preliminary testing of the container, the solenoid and relief valve were located inside the container, but subsequently they were moved outside the container to "prevent the solenoid from freezing open." The problems were later discovered to be software related. b) The transfer line could be vacuum and powder insulated. The current system of insulation was employed because of ease of design refinement, but now that a working system has been established a permanent, vacuum-insulated transfer line may be warranted.
2. Data indicated conduction of heat into the container may be a major reason behind the temperature extremes within. This problem could most easily be resolved by increasing the insulation around the container.

3. Additional research is needed to determine why the temperature of stored products varied so much from the front to the back of the container.

4. A large amount of N_2 was consumed just to overcome the heat produced by the fans. Therefore, reducing the number of circulation fans would definitely be an asset. The finite element simulation of master packaged meat indicates the cooling rate of the meat was limited by conduction rather than convection for air flow rates provided by two fans. Therefore, a need does not exist for any more than two fans if air distribution in the container can prevent large temperature extremes. Evidently, the current air distribution system was not effective for two fans and there is a need to investigate other possibilities. Ideas of jacketed storage were discarded in preliminary designs because high air flow were believed to be needed to encourage rapid cooling. Because simulations indicated high air flow was not necessary, jacketed storage becomes an alternative that should provide more effective temperature control by removing heat conducted through the container walls before it comes in contact with the stored product.

5. The mass of the shelving unit should be reduced (if it is to be used commercially) by making the container walls part of the structure supporting the master trays, and thereby reducing the amount of steel required.

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APPENDIX A - DRAWINGS OF SHELVING UNIT

A.1. Front View

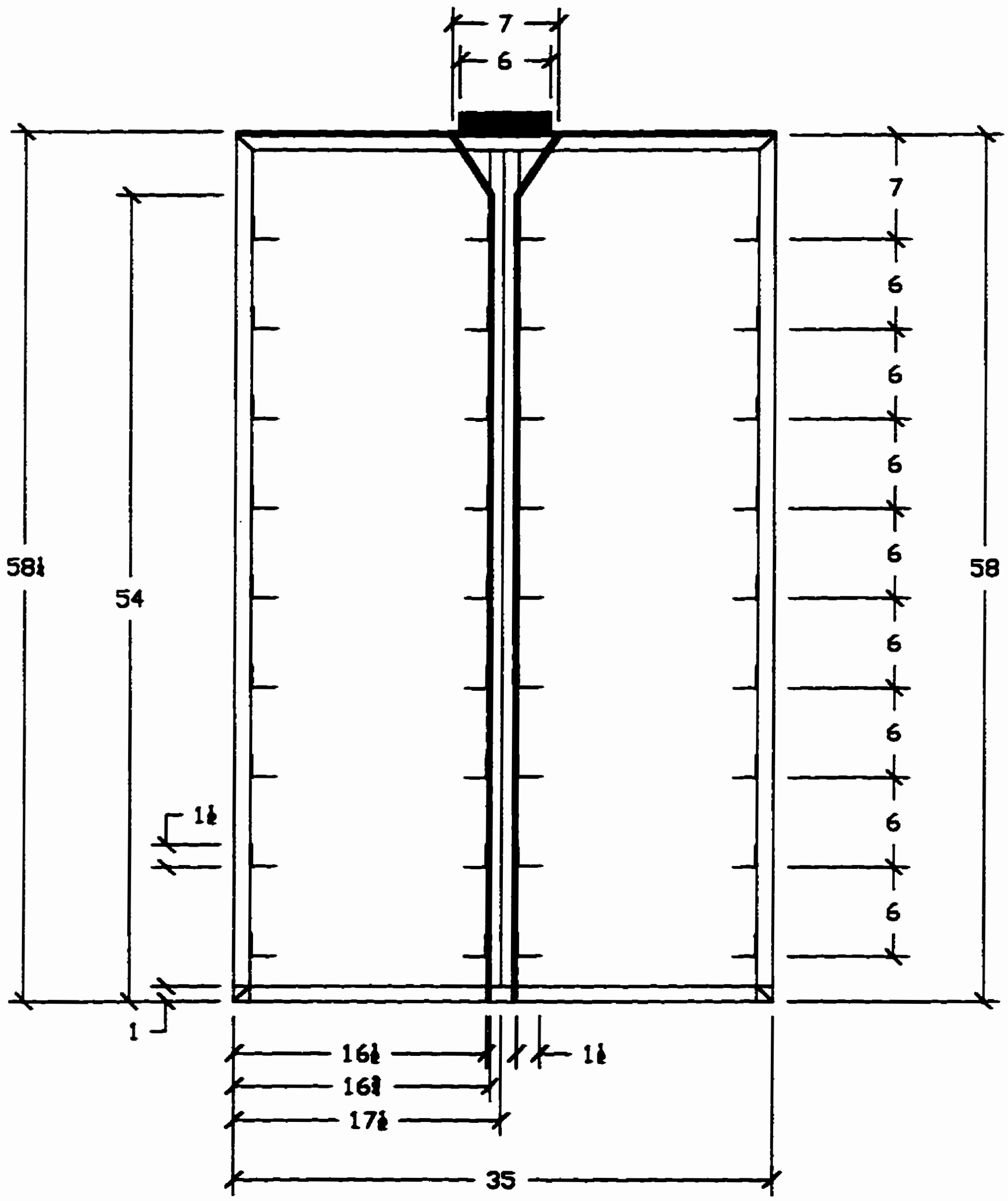

Figure A.1. Front view of the shelving unit with Imperial measurements (inches).

A.2. Side View

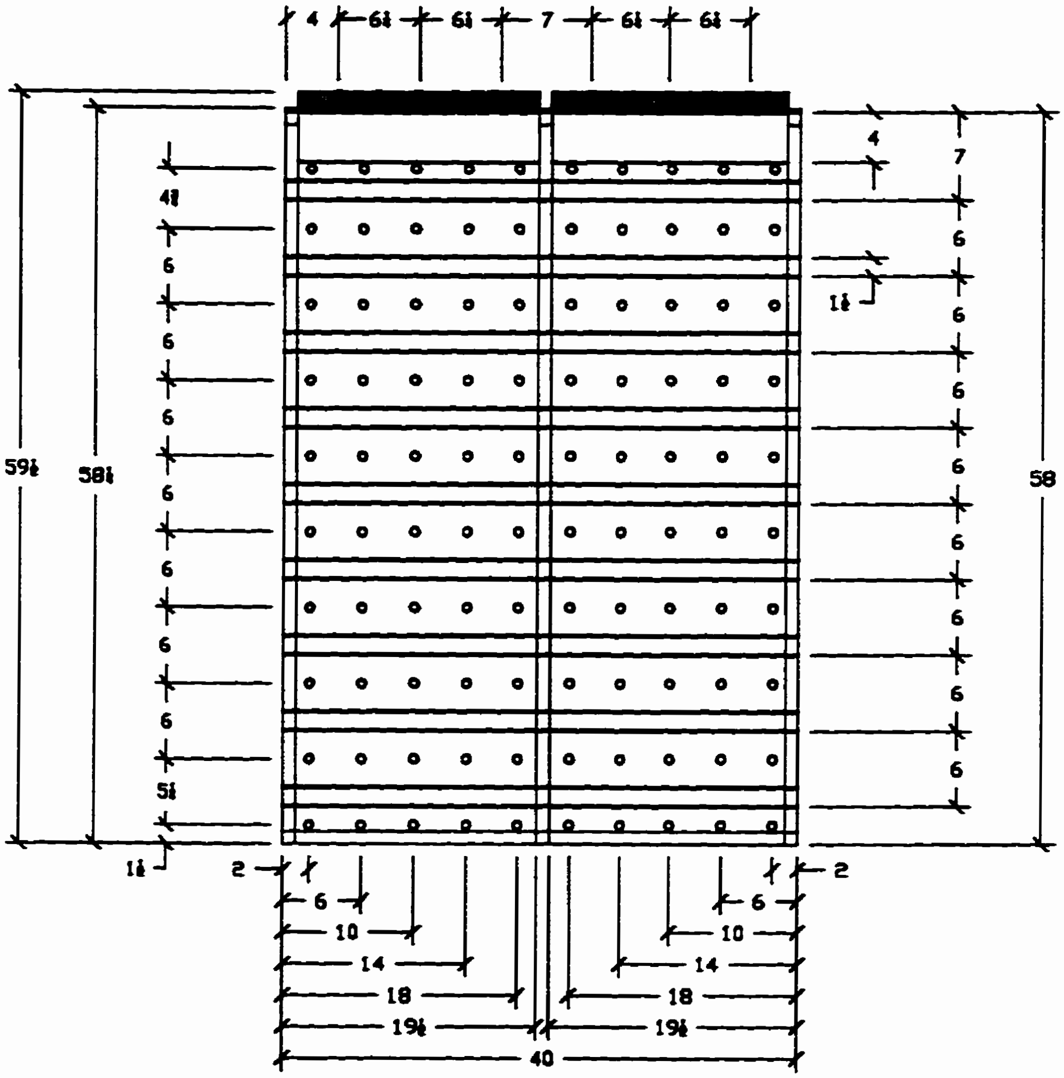

Figure A.2. Side view (from the inside of the container) of the shelving unit with Imperial dimensions (inches).

A.3. Top View

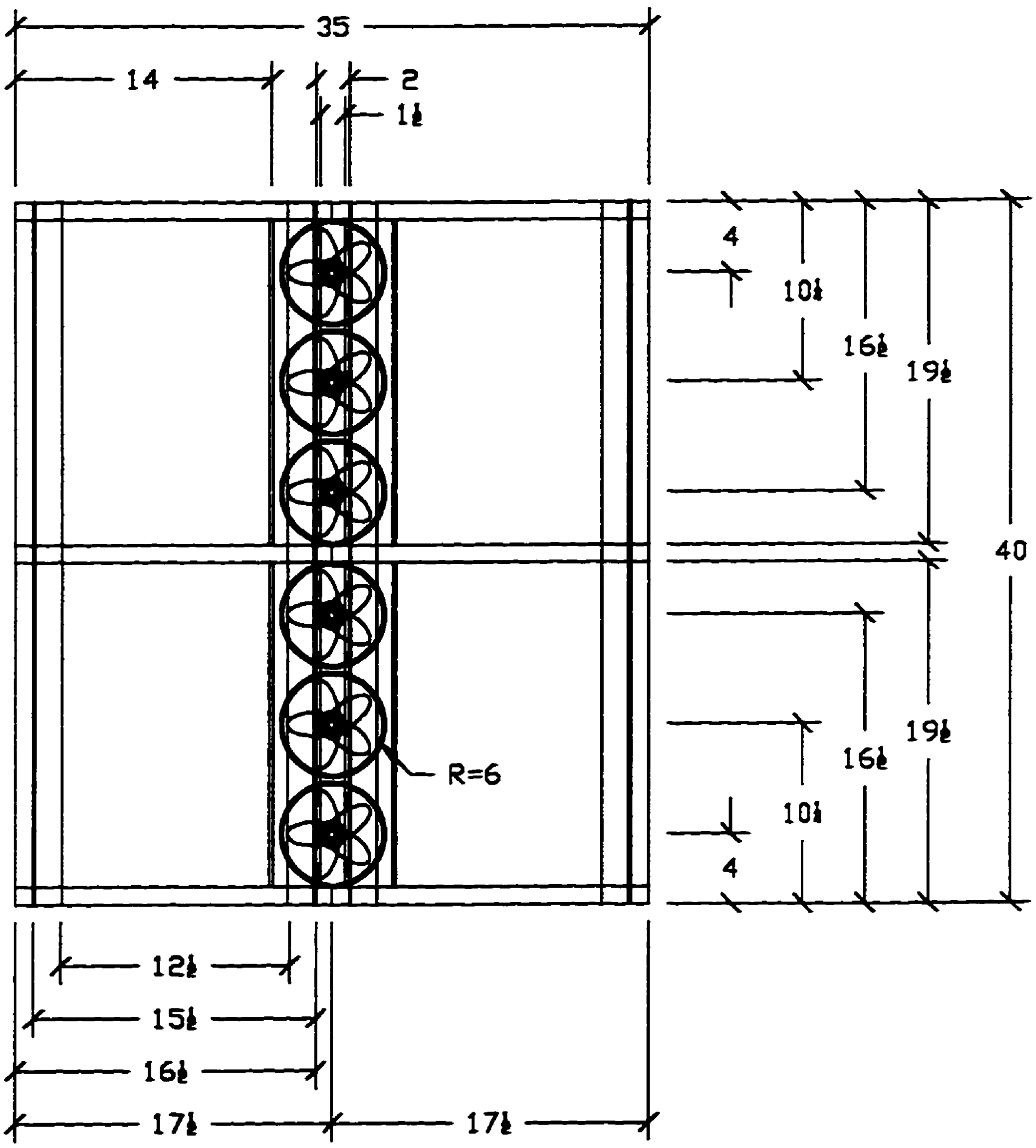

Figure A.3. Top view of the shelving unit with Imperial dimensions (inches).

APPENDIX B - CO_2 vs. N_2

It is desirable to store liquid N_2 and CO_2 at low pressures because they possess greater latent heats. Typically, the food industry uses liquid N_2 stored at tank pressures of either 152 or 241 kPa and liquid CO_2 at 2 MPA. At these pressures, liquid N_2 is approximately −185°C (Van Wylen and Sonntag 1985) and liquid CO_2 is approximately −20°C (ASHRAE 1994). The latent heats of liquid N_2 at 152 and 241 kPa are 186 and 181 kJ/kg, respectively (Van Wylen and Sonntag 1985). Liquid CO_2 has a much larger latent heat of 279 kJ/kg (Wood 1982). However, when liquid N_2 is superheated from a saturated vapour at 152 and 241 kPa to −1.5°C and 1 atm, its change in sensible heat is 199 and 197 kJ/kg, respectively. The change in sensible heat experienced by CO_2 when superheated from a saturated vapour at 2 MPA to the same final state as the N_2 is only 47 kJ/kg. Therefore, the total cooling potential of liquid N_2 injected into the container is 385 and 378 kJ/kg for storage at 152 and 241 kPa, respectively, while for liquid CO_2 it is 326 kJ/kg.

Although liquid N_2 has more cooling potential per unit mass, it is also more expensive. Two local suppliers were contacted and asked the list price (although one supplier remarked that N_2 and CO_2 were seldom sold at their list price and accurate pricing would depend on quantity purchased) of their products purchased in 160 L quantities (a common size of portable storage containers). One supplier (B. Hughes, Sales Representative with Liquid Air, Winnipeg, MB) quoted a price of \$2.40/kg of liquid N_2 and \$1.40/kg of liquid CO_2 (Note: price was quoted per cubic metre of N_2, but was converted to kilograms using the conversion factor provided). The other supplier

(S. Anderson, Manager of Winnipeg Welding Supply, Winnipeg, MB) quoted a price of \$2.45/kg of N_2 and \$1.00/kg of CO_2. Therefore, when the value of N_2 was calculated based on the cooling potential provided, it was between 6.4 and 6.5 ¢/kJ. For CO_2, this value was between 3.0 and 4.3 ¢/kJ. Therefore, CO_2 was considerably cheaper than N_2 for comparable amounts of refrigeration.

The cost of refrigerating meat (\$/kg of meat or \$/h) under various scenarios can be determined from information in Table 5.2. or Appendix H.

APPENDIX C - INSTRUMENTATION ERROR

C.1. Temperature Errors from the HP 34401A Multimeter

The multimeter was used to measure the resistance of the thermistors between temperature extremes of 55 and −10°C. This corresponds to a minimum resistance of approximately 1.5 kΩ for a single thermistor at 55°C and a maximum resistance of 55 kΩ for two thermistors in series at −10°C (OMEGA 1990). This encompasses two ranges of resistance measurement on the HP multimeter (the 10 and 100 kΩ resistance ranges). The 90 day accuracy for these ranges after a 1 h warm-up with the default 5 ½ digits of resolution (10 power line cycles) is ±(0.008% of reading + 0.002% of range) (HP Users Guide 1992).

Because of the nonlinear behaviour of thermistors, the maximum error in temperature measurement contributed by the multimeter was expected at 1.5 kΩ. This error was calculated as follows:

$$Error\ at\ 1.5k\Omega = \pm(0.008\% \times 1500\Omega + 0.002\% \times 10000\Omega) = \pm 0.32\Omega \quad \textbf{(C.1)}$$

At a thermistor resistance of 1.5 kΩ, a change in resistance of 55 Ω represents a change in temperature of 1°C. Using a linear approximation, the resistance error was converted to a temperature error:

$$Maximum\ Temperature\ Measurement\ Error = \frac{(1\,°C)(0.32\Omega)}{55\Omega} = 0.006\,°C \quad \textbf{(C.2)}$$

C.2. Thermistor Self Heating Error

Thermistors experience self heating because of the test current used by the multimeter to measure the resistance. The resulting errors in temperature measurement were a function of both the test current and the heat dissipation constant of the thermistor. Different test currents were used for different ranges of resistance measurement by the HP multimeter. The highest test current used was 100 μA in the 10 kΩ range. The greatest self heating occurs at the highest resistance measured in that range:

$$\begin{aligned} \text{Self Heating} &= (\text{Test Current})^2 \times (\text{Resistance}) \\ &= (100 \times 10^{-6})^2(10000) = 0.1\,mW \end{aligned} \tag{C.3}$$

The OMEGA engineering literature (1990) states that the minimum dissipation constant for a thermistors in still air is 1 mW/°C. Using this constant, the self heating error was determined:

$$\text{Self Heating Error} = \frac{0.1\,mW}{1\,\frac{mW}{°C}} = 0.1\,°C \tag{C.4}$$

Although this error was large, it must be remembered that it represents the worst self heating error for the data acquisition system. Certain assumptions in the preceding calculations do not necessarily hold true. For example, it was very seldom that the temperature of still air was measured because of the circulation fans present within the container. It was also difficult to conceive that self heating would be a major problem when the test current was supplied for only 10 power line cycles per reading of each channel. Although it was believed that the actual self heating error was much lower than calculated, it was difficult to prove. Therefore, thermistors were chosen in such a manner that a resistance of 10 kΩ would reflect a temperature outside the range of primary interest i.e. $-1.5 \pm 0.5°C$.

The OMEGA 44007 thermistor has an approximate resistance of 10 kΩ at 10°C. In the temperature range of $-1.5 \pm 0.5°C$, resistance varies from 17.19 to 18.10 kΩ. These resistance fall into the 100 kΩ range of the HP multimeter, which uses a test current of 10 μA for measurement in this range. The resulting self heating error was:

$$Self\ Heating = (Current)^2 \times (Resistance) = (10 \times 10^{-6})^2(18100) = 1.81\,\mu W$$

$$Self\ Heating\ Error = \frac{1.81\,\mu W}{1\,\frac{mW}{°C}} = 0.0018\,°C \qquad \text{(C.5)}$$

Evidently, all temperatures measured in the 100 kΩ range of the multimeter exhibit extremely small self heating errors. Because the multimeter was capable of overranging 20%, it was assumed that self heating was almost negligible for all resistance measured in excess of 12 kΩ. For the OMEGA 44007 thermistor, this corresponds to temperature measurements less than 6°C.

C.3. Error From the Calibration Equation

Calibration of the thermistors was accomplished using the following equation provided by OMEGA (1992):

$$\frac{1}{T} = A + B(LOG_eR) + C(LOG_eR)^3 \qquad \text{(C.6)}$$

where:

R = resistance (Ω) T = temperature (K) A, B, C = fitted constants

When the temperatures are chosen to span no more than 100°C, the above equation has an accuracy of ±0.02°C or better. Therefore, temperature measurement errors introduced by the calibration equation were negligible.

C.4. Error from Wire Lead and Instrumentation Resistance

Because thermistors measure temperatures based on their resistance, temperature measurement errors were caused by the resistance of wire leads connecting thermistors to the multiplexer and the resistance of the relays within the multiplexer.

The combined resistance of the wire leads and the resistance through the multiplexer was measured with the HP 34401A multimeter. The maximum resistance measured on any one channel was 2.0 Ω, which was insignificant compared to the 910 Ω change experienced when the thermistor changes temperatures from −1.0 to −2.0°C.

C.5. Calibration of Temperature Sensors

The three constants in Equation C.6. were solved by measuring the resistance of the thermistor at three known temperatures. To accomplish this, all twenty four thermistors were inserted into a stirred liquid bath and the temperature was measured with a calibration thermometer (see section C.6. for a description of the thermometers used). The bath was insulated to ensure temperature variations throughout the fluid were negligible. Table C.1. summarizes the resistance measured for the twenty-three channels at four different temperatures, and Table C.2. shows the calibration constants for Equation C.6.

Calibration of the thermistors was checked at the beginning and end of testing to ensure the precision of the instrumentation system. The check was performed by inserting all thermistors into a stirred ice bath and comparing the measured temperatures. Table C.3. shows the measured temperatures and testifies to the precision of the calibration. Temperatures measured by channels 3 and 4 deviate slightly from the norm, but these two sensors were calibrated for a different temperature range than the rest. Based on the accuracy of the calibration thermometers (see Section C.6.), a variation of this nature was expected.

C.6. Description of the Calibration Thermometers

Three different thermometers were used for the calibration. All three were products of Miller & Weber Inc., Ridgewood, NY and had graduations down to 0.1°C. A description follows:

1) ASTM 52C: Total immersion thermometer with a temperature range of -10 to 5°C.
2) ASTM 17C: Partial immersion thermometer with a temperature range of 19 to 27°C.
3) ASTM 19C: Partial immersion thermometer with a temperature range of 49 to 57°C.

According to the manufacturer's information, all three thermometers were accurate to plus or minus one graduation (±0.1°C).

C.7. Summary of Errors and Calibration

All temperature measurement errors (except for thermistor self heating errors) appear to be negligible. Self heating errors were also assumed minute for temperatures measured under 6°C. Therefore the precision of the thermistor was well within ±0.1°C, however they can not be considered more accurate than the calibration thermometers used i.e. ±0.1°C.

Table C.1. The resistance of thermistors attached to a data acquisition system at four different temperatures.

Channel number	Resistance (kΩ) at -9.7 °C	Resistance (kΩ) at 4.6 °C	Resistance (kΩ) at 19.8 °C	Resistance (kΩ) at 52.5 °C
2[a]	54.75	26.11	12.93	3.29
3	27.39	13.07	6.47	1.65
4	27.33	13.05	6.47	1.65
5	27.37	13.05	6.47	1.64
6	27.26	13.00	6.44	1.64
7	27.38	13.06	6.47	1.65
8	27.49	13.11	6.50	1.65
9	27.43	13.08	6.48	1.65
10	27.35	13.04	6.46	1.64
11	27.45	13.10	6.49	1.65
12	27.36	13.05	6.47	1.65
13	27.41	13.07	6.47	1.65
14	27.36	13.05	6.47	1.65
15	27.35	13.05	6.47	1.65
16	27.31	13.04	6.46	1.65
17	27.34	13.05	6.47	1.65
18	27.39	13.06	6.47	1.65
19	27.33	13.04	6.46	1.65
20	27.35	13.04	6.46	1.65
21	27.38	13.06	6.47	1.65
22	27.38	13.06	6.47	1.65
23	27.29	13.03	6.46	1.65
24	27.33	13.04	6.47	1.65

[a] channel two has two thermistors wired in series.

Table C.2. Constants of Equation C.6. for the thermistors attached to the data acquisition system.

Channel number	Constants for the OMEGA equation		
	A	B	C
2 [a]	8.129011×10^{-4}	2.786707×10^{-4}	-4.432981×10^{-8}
3 [b]	1.424941×10^{-3}	2.114309×10^{-4}	1.972551×10^{-7}
4 [b]	1.423625×10^{-3}	2.115216×10^{-4}	1.982444×10^{-7}
5 [a]	1.005379×10^{-3}	2.782671×10^{-4}	-4.940314×10^{-8}
6 [a]	1.005954×10^{-3}	2.783039×10^{-4}	-4.932356×10^{-8}
7 [a]	1.008979×10^{-3}	2.776356×10^{-4}	-4.682515×10^{-8}
8 [a]	1.002313×10^{-3}	2.784965×10^{-4}	-4.982788×10^{-8}
9 [a]	1.004475×10^{-3}	2.783707×10^{-4}	-5.008760×10^{-8}
10 [a]	1.004928×10^{-3}	2.783445×10^{-4}	-4.953817×10^{-8}
11 [a]	1.006819×10^{-3}	2.778559×10^{-4}	-4.756499×10^{-8}
12 [a]	1.013812×10^{-3}	2.768778×10^{-4}	-4.390037×10^{-8}
13 [a]	1.024967×10^{-3}	2.751767×10^{-4}	-3.854361×10^{-8}
14 [a]	1.005270×10^{-3}	2.782098×10^{-4}	-4.867273×10^{-8}
15 [a]	1.003086×10^{-3}	2.785421×10^{-4}	-4.968365×10^{-8}
16 [a]	1.007190×10^{-3}	2.779125×10^{-4}	-4.718487×10^{-8}
17 [a]	1.008695×10^{-3}	2.776646×10^{-4}	-4.649475×10^{-8}
18 [a]	1.004145×10^{-3}	2.784316×10^{-4}	-4.998201×10^{-8}
19 [a]	1.010949×10^{-3}	2.773096×10^{-4}	-4.510340×10^{-8}
20 [a]	1.004506×10^{-3}	2.783676×10^{-4}	-4.933511×10^{-8}
21 [a]	1.009393×10^{-3}	2.775713×10^{-4}	-4.659179×10^{-8}
22 [a]	1.001025×10^{-3}	2.788092×10^{-4}	-5.057847×10^{-8}
23 [a]	1.002555×10^{-3}	2.785624×10^{-4}	-4.883831×10^{-8}
24 [a]	1.005629×10^{-3}	2.781188×10^{-4}	-4.788411×10^{-8}

[a] these channels were calibrated for the temperatures of -9.7, 4.6, and 19.8°C.

[b] these channels were calibrated for the temperatures of -9.7, 19.8, and 52.5°C.

Table C.3. Measured temperatures of water baths demonstrating the precision of the instrumentation system at both the beginning and end of testing.

Channel number	Measured temperatures of two different water baths (°C)	
	Before starting experimentation	After completion of experimentation
2 [a]	0.54	0.15
3 [b]	0.79	0.40
4 [b]	0.78	0.41
5 [a]	0.53	0.16
6 [a]	0.53	0.15
7 [a]	0.52	0.14
8 [a]	0.53	0.14
9 [a]	0.54	0.15
10 [a]	0.51	0.15
11 [a]	0.51	0.15
12 [a]	0.54	0.15
13 [a]	0.53	0.16
14 [a]	0.51	0.16
15 [a]	0.50	0.14
16 [a]	0.51	0.14
17 [a]	0.51	0.14
18 [a]	0.51	0.14
19 [a]	0.51	0.14
20 [a]	0.49	0.13
21 [a]	0.51	0.14
22 [a]	0.49	0.13
23 [a]	0.50	0.13
24 [a]	0.50	0.13

[a] these channels were calibrated for the temperatures of -9.7, 4.6, and 19.8°C.

[b] these channels were calibrated for the temperatures of -9.7, 19.8, and 52.5°C.

APPENDIX D - TEMPERATURE CONTROL PROGRAM

D.1. Program1

Program1 is a QuickBASIC program which uses three temperature ratios to calculate the shut off temperature for the solenoid. For reasons of conserving space, it was not printed here. However, it was similar to Program2 in design and construction, which is included in the Section D.2. The major differences between both programs exist in the subroutine *SoleControl*. Lines changes from Program1 to Program2 were marked with asterisks (*), and the original code was stored in remark statements.

D.2. Program2

What follows is Quick Basic Program2:

```
DECLARE SUB PrintTime ()
DECLARE SUB Alarm ()
DECLARE SUB FileDating ()
DECLARE SUB TakeReadings ()
DECLARE SUB PrintGraph ()
DECLARE SUB SoleControl ()
DECLARE SUB GetTemp ()
DECLARE SUB Initialize ()
DECLARE SUB Reinitialize ()
DECLARE SUB ChannelCodes ()
DECLARE SUB ParallelPortOut ()

'**********************Variable Description*****************************
' IniData%      Reads value of the data bus (address &H378) at beginning
'               of program and restores it at the end of the program.
' IniStatus%    Reads value of the status bus (address &H379) at beginning
'               of program and restores it at the end of the program.
' IniControl%   Reads value of the control bus (address &H37A) at beginning
'               of program and restores it at the end of the program.
' DData%        Stores the code that is wrote to the data bus.
' Control%      Stores the code that is wrote to the control bus.
' FileName1$    Retains the name used for the data file containing
```

```
'               temperature and weight data.
' FileName2$    Retains the name used for the data file containing the
'               solenoid control information.
' Channel%      Stores the number of the active channel.
' Solenoid%     If = 1 then solenoid is energized. If 0 then not.
' Quit$         If Quit$ = "quit" then program will shut down.
' TempOff       Stores temperature at which solenoid is turned off.
' Optionn       Alerts main program as to which function key was pressed.
' Max()         An array with the 10 previous maximum temperatures.
' Avg()         An array with the 10 previous average temperatures.
' Min()         An array with the 10 previous minimum temperatures.
' T()           Stores the current temperatures of sensors 2 through 24.
' Ave2Temp      Stores the time-weighted average of temperature sensor 2.
' Sum2Temp      Stores the sum of control sensors for calculating Ave2Temp.
' number        Stores the number of numbers added to Sum2Temp.
' Weight        Stores the weight of the LGS
' Tare          Stores the tare weight of the LGS - stamped in side of LGS.
' BeginTime     Stores the start time of test.
' PrevTime      Stores the last time that the PrintTime subroutine was called.
' Time          Stores the total number of seconds ellapsed since test begun.
' RollTime      Stores the number of times the timer has rolled past midnight.
' LoopTime      Counts the number of seconds into each 60 s loop.
' Ratio()       Stores the ratios of average cycle temperatures to TempOff.
' BeginWeight   Stores the weight of the LGS at beginning of test.

COMMON SHARED IniData%, IniStatus%, IniControl%, DData%, Control%
COMMON SHARED FileName1$, FileName2$, BeginWeight
COMMON SHARED Channel%, Solenoid%, Quit$, TempOff, Optionn, Ratio()
COMMON SHARED Max(), Avg(), Min(), T(), Sum2Temp, Weight, Tare, number
COMMON SHARED BeginTime, PrevTime, Time, RollTime, LoopTime

DIM T(24), Max(10), Min(10), Avg(10), Ratio(6)

SCREEN 9
CALL Initialize: CALL FileDating
BeginTime = TIMER

DO:
 CALL PrintTime
 IF LoopTime = 0 THEN CALL TakeReadings: CALL PrintGraph
 IF LoopTime = 20 THEN CALL SoleControl
LOOP UNTIL Quit$ = "quit"
```

```
CALL Reinitialize

END   '*************************************************************

F1Key:
 IniRow = CSRLIN: IniColumn = POS(x)
 IF Optionn <> 0 THEN CALL PrintGraph
 Optionn = 1
 CLS 1: FOR I = 1 TO 23: LOCATE I, 1: PRINT "      ": NEXT I
 LOCATE 10,.20: PRINT "F1  ===> Displays help screen."
 LOCATE 11, 20: PRINT "F2  ===> Terminates the program."
 LOCATE 12, 20: PRINT "F3  ===> Displays temperature readings."
 LOCATE 13, 20: PRINT "F4  ===> DOS command shell."
 LOCATE 15, 20: PRINT "Press 'n' to clear."
 LOCATE IniRow, IniColumn: KEY(16) ON
RETURN

F2Key:
 IniRow = CSRLIN: IniColumn = POS(x)
 IF Optionn <> 0 THEN CALL PrintGraph
 Optionn = 2
 CLS 1: BEEP: LOCATE 12, 27: COLOR 12, 0: PRINT "Quit (y or n)?"
 LOCATE IniRow, IniColumn: KEY(15) ON: KEY(16) ON
RETURN

F3Key:
 IniRow = CSRLIN: IniColumn = POS(x)
 IF Optionn <> 0 THEN CALL PrintGraph
 Optionn = 3
 CLS 1: FOR I = 1 TO 23: LOCATE I, 1: PRINT "      ": NEXT I
 COLOR 12, 0
 FOR I = 2 TO 21
  LOCATE I, 20
  PRINT USING "Channel ## =====> ###.## °C"; I + 3; T(I + 3)
 NEXT I
 LOCATE 23, 20: PRINT "Press 'n' to clear."
 LOCATE IniRow, IniColumn: KEY(16) ON
RETURN

F4Key:
 IniRow = CSRLIN: IniColumn = POS(x)
 IF Optionn <> 0 THEN CALL PrintGraph
 Optionn = 4
```

```
 CLS 1: BEEP: LOCATE 13, 27: COLOR 12, 0: PRINT "DOS Shell (y or n)?"
 LOCATE IniRow, IniColumn: KEY(15) ON: KEY(16) ON
RETURN

NKey:
 CALL PrintGraph: Optionn = 0
RETURN

YKey:
 KEY(15) OFF: KEY(16) OFF
 IF Optionn = 2 THEN
  LOCATE 23, 55: COLOR 12, 0: PRINT "Shutting Down     ": Quit$ = "quit"
 ELSEIF Optionn = 4 AND Solenoid% = 1 THEN
  LOCATE 13, 27: PRINT "DOS Shell denied because solenoid is energized."
  CALL PrintGraph
 ELSEIF Optionn = 4 THEN
  CLS 1: CLS 2: COLOR 12, 0: PRINT "Type EXIT to return to program."
  SHELL: SCREEN 0: SCREEN 9: CALL PrintGraph
 ELSE
  CALL PrintGraph
 END IF
 Optionn = 0
RETURN

ErrorHandler:
RESUME NEXT
```

SUB Alarm

```
'If sensor 2 drops below -7 degrees, then entire control of program is
'passed over to this subroutine.  A warning message is flashed and the
'computer makes a bizar noise.  This subroutine is necessary in case the
'solenoid fails or some other disaster occurs.

CLS 1
LOCATE 10, 20: PRINT "Please find Chris Bailey immediately or"
LOCATE 11, 20: PRINT "alert Jack, Matt or Dale to the situation."
LOCATE 12, 20: PRINT "Chris' home phone number is 261-9513."

Channel% = 2: CALL ChannelCodes: CALL ParallelPortOut
DO
 SOUND RND * 1000 + 37, .2                'generates a bizar noise.
 CALL GetTemp
```

```
 LOCATE 16, 20: PRINT USING "Current Temp ###.## °C"; T(2)
LOOP UNTIL T(2) > -7

END SUB

SUB ChannelCodes        'Codes for multiplexer.

SELECT CASE Channel%   'Determines signal for data bus which activates
 CASE 9, 17            'the latches. Note: Channel 1 is the solenoid.
  DData% = &HFE
 CASE 2, 10, 18
  DData% = &HFD
 CASE 3, 11, 19
  DData% = &HFB
 CASE 4, 12, 20
  DData% = &HF7
 CASE 5, 13, 21
  DData% = &HEF
 CASE 6, 14, 22
  DData% = &HDF
 CASE 7, 15, 23
  DData% = &HBF
 CASE 8, 16, 24
  DData% = &H7F
END SELECT

SELECT CASE Channel%   'Determines signal for status bus which activates
 CASE 2 TO 8           'the 3 to 8 line decoder.
  Control% = &HB
 CASE 9 TO 16
  Control% = &H9
 CASE 17 TO 24
  Control% = &H3
END SELECT

END SUB

SUB FileDating

COLOR 12, 0: CLS
LOCATE 3, 1: PRINT "Please correct the current date and time or"
LOCATE 4, 1: PRINT "press <ENTER> to accept the default values."
LOCATE 5, 1: PRINT "Note:  Data files are labelled according to"
```

```
LOCATE 6, 1: PRINT "the  starting  date and time  of the  test.": PRINT

'Change date and time through the DOS shell and create a filename from date.
SHELL "Date": PRINT : SHELL "Time"
name1$ = LEFT$(DATE$, 2) + MID$(DATE$, 4, 2)
name2$ = LEFT$(TIME$, 2) + MID$(TIME$, 4, 2)
FileName1$ = name1$ + name2$ + ".txt"
FileName2$ = FileName1$: MID$(FileName2$, 8) = "s"
LOCATE 14, 1: PRINT "Based on the time and date given, the name"
LOCATE 15, 1: PRINT "of the temperature and solenoid data file are:"
COLOR 7, 0: LOCATE 17, 1: PRINT "C:\CHRIS\OUTPUT\"; FileName1$
LOCATE 18, 1: PRINT "C:\CHRIS\OUTPUT\"; FileName2$
CHDIR "C:\Chris\Output": OPEN FileName1$ FOR OUTPUT AS #1: CLOSE #1
OPEN FileName2$ FOR OUTPUT AS #2: CLOSE #2

'Allows user to use DOS editor to place a description of test in file.
COLOR 12, 0: LOCATE 20, 1: PRINT "Do you wish to attach a description of the"
LOCATE 21, 1: PRINT "test to your temperature file (y or n)?"
DO: Answer$ = INKEY$: LOOP UNTIL Answer$ <> ""
IF Answer$ = "y" OR Answer$ = "Y" THEN SHELL "edit " + FileName1$
SCREEN 0: SCREEN 9

END SUB

SUB GetTemp

'Read resistance of thermister from voltmeter.
PRINT #3, "meas:res?": INPUT #3, Rdg$: Resist = VAL(Rdg$)

'Constants for the OMEGA thermister equation.  Each thermistor was
'calibrated individually.
SELECT CASE Channel%
 CASE 2
  a = 8.129011E-04: b = 2.786707E-04: C = -4.432981E-08
 CASE 3
  a = 1.424941E-03: b = 2.114309E-04: C = 1.972551E-07
 CASE 4
  a = 1.423625E-03: b = 2.115216E-04: C = 1.982444E-07
 CASE 5
  a = 1.005379E-03: b = 2.782671E-04: C = -4.940314E-08
 CASE 6
  a = 1.005954E-03: b = 2.783039E-04: C = -4.932356E-08
 CASE 7
```

```
 a = 1.008979E-03: b = 2.776356E-04: C = -4.682515E-08
CASE 8
 a = 1.002313E-03: b = 2.784965E-04: C = -4.982788E-08
CASE 9
 a = 1.004475E-03: b = 2.783707E-04: C = -5.00876E-08
CASE 10
 a = 1.004928E-03: b = 2.783445E-04: C = -4.953817E-08
CASE 11
 a = 1.006819E-03: b = 2.778559E-04: C = -4.756499E-08
CASE 12
 a = 1.013812E-03: b = 2.768778E-04: C = -4.390037E-08
CASE 13
 a = 1.024967E-03: b = 2.751767E-04: C = -3.854361E-08
CASE 14
 a = 1.00527E-03: b = 2.782098E-04: C = -4.867273E-08
CASE 15
 a = 1.003086E-03: b = 2.785421E-04: C = -4.968365E-08
CASE 16
 a = 1.00719E-03: b = 2.779125E-04: C = -4.718487E-08
CASE 17
 a = 1.008695E-03: b = 2.776646E-04: C = -4.649475E-08
CASE 18
 a = 1.004145E-03: b = 2.784316E-04: C = -4.998201E-08
CASE 19
 a = 1.010949E-03: b = 2.773096E-04: C = -4.51034E-08
CASE 20
 a = 1.004506E-03: b = 2.783676E-04: C = -4.933511E-08
CASE 21
 a = 1.009393E-03: b = 2.775713E-04: C = -4.659179E-08
CASE 22
 a = 1.001025E-03: b = 2.788092E-04: C = -5.057847E-08
CASE 23
 a = 1.002555E-03: b = 2.785624E-04: C = -4.883831E-08
CASE 24
 a = 1.005629E-03: b = 2.781188E-04: C = -4.788411E-08
END SELECT

'Use the OMEGA thermister equation to convert resistance to temperature.
'Temperatures are rounded to the nearest second decimal place to conserve
'space during data storage.
LOGV = LOG(Resist) / LOG(2.71828)    'calculates log to base e
T(Channel%) = (1 / (a + b * LOGV + C * (LOGV ^ 3)) - 273.15)
```

```
END SUB

SUB Initialize

ON ERROR GOTO ErrorHandler

'Store initial values of data, control and status registers.
IniData% = INP(&H378): IniStatus% = INP(&H379): IniControl% = INP(&H37A)

'Open COM ports for input/output routines.
OPEN "COM1:9600,N,8,1,RS,CD,LF" FOR RANDOM AS #3     'COM 1 for voltmeter
OPEN "COM2:9600,E,7,2,RS,DS" FOR RANDOM AS #4        'COM 2 for scale
PRINT #3, "SYST:REM"

'Initialize variables
FOR I = 1 TO 6: Ratio(I) = 1: NEXT I: TempOff = -1.5
FOR I = 1 TO 10: Max(I) = -10: Min(I) = 99: Avg(I) = 99: NEXT I

'Display a description of the program and operative keys.
LOCATE 2, 1: COLOR 12, 0
PRINT "                    Chris Bailey's Program"
COLOR 7, 0: PRINT
PRINT "  MY GRADUATE THESIS OBJECTIVE:  research, design, fabricate, and test"
PRINT "  a refrigerated, controlled-atmosphere container for use in the"
PRINT "  distribution of fresh red meat to foreign and domestic markets."
PRINT
PRINT "  PROGRAM DESCRIPTION: this program monitors the amount of liquid"
PRINT "  nitrogen used to refrigerate the container. Twenty-four temperature"
PRINT "  sensors are also used to record temperatures within the container."
PRINT "  Based on these temperatures, an algorithm determines when to turn"
PRINT "  a solenoid on/off. This solenoid regulates the amount of liquid"
PRINT "  nitrogen delivered to the container or in other words, the amount"
PRINT "  of refrigeration."
PRINT
PRINT "  ACTIVE KEYSTROKES DURING PROGRAM OPERATION:"
PRINT "       F1 ===> Displays help screen."
PRINT "       F2 ===> Terminates the program."
PRINT "       F3 ===> Displays temperature readings."
PRINT "       F4 ===> DOS command shell."

DO: LOOP UNTIL INKEY$ <> ""

'Set up key trapping of F1, F2, F3, F4 keys.
```

```
ON KEY(1) GOSUB F1Key: ON KEY(2) GOSUB F2Key
ON KEY(3) GOSUB F3Key: ON KEY(4) GOSUB F4Key

'Define and set up key trappying of the 'y' and 'n' key.
KEY 15, CHR$(0) + CHR$(21): ON KEY(15) GOSUB YKey
KEY 16, CHR$(0) + CHR$(49): ON KEY(16) GOSUB NKey

'Read the initial weight of the container and ask for tare weight.
CLS
LOCATE 12, 10: INPUT "Please enter the tare weight of the LGS (kg)"; Tare
PRINT #4, "P": INPUT #4, Weight$
BeginWeight = VAL(MID$(Weight$, 4, 5))

END SUB

SUB ParallelPortOut         'Sends signals to the multiplexer.

IF Solenoid% = 0 THEN
 OUT &H378, &HFF            'Put clear signal on databus.
 OUT &H37A, &HB             'Enable latch 1.
 OUT &H37A, &H9             'Enable latch 2.
 OUT &H37A, &H3             'Enable latch 3.
 OUT &H37A, &H1             'Disable latches.
 OUT &H378, DData%          'Write data signal for channel selection.
 OUT &H37A, Control%        'Enable specified latch.
 OUT &H37A, &H1             'Disable latches
ELSEIF Solenoid% = 1 AND Control% = &HB THEN
 OUT &H378, &HFE EQV DData%     'the solenoid and channel and put on databus.
 OUT &H37A, Control%        'Enable latch 1.
 OUT &H37A, &H1             'Disable latches.
 OUT &H378, &HFF            'Place clear signal on databus.
 OUT &H37A, &H9             'Enable latch 2.
 OUT &H37A, &H3             'Enable latch 3.
 OUT &H37A, &H1             'Disable latches.
ELSEIF Solenoid% = 1 AND Control% <> &HB THEN
 OUT &H378, &HFE            'Place solenoid activation on databus.
 OUT &H37A, &HB             'Enable latch 1.
 OUT &H37A, &H1             'Disable latches.
 OUT &H378, &HFF            'Place clear signal on databus.
 OUT &H37A, &H9             'Enable latch 2.
 OUT &H37A, &H3             'Enable latch 3.
 OUT &H37A, &H1             'Disable latches.
 OUT &H378, DData%          'Write data signal for channel selection.
```

```
 OUT &H37A, Control%       'Enable specified latch.
 OUT &H37A, &H1            'Disable latches.
END IF

END SUB
```

SUB PrintGraph

```
'All key trapping is off during this subroutine.
KEY(1) OFF: KEY(2) OFF: KEY(3) OFF: KEY(4) OFF: KEY(15) OFF: KEY(16) OFF

'Define graphing window.
CLS 1: CLS 2: VIEW (55, 7)-(500, 298), , 9: COLOR 12, 0

LOCATE 1, 65: PRINT "TIME"
LOCATE 5, 65: PRINT "TEMPERATURES"
LOCATE 6, 65: PRINT USING "Max:  ###.## °C"; Max(1)
LOCATE 7, 65: PRINT USING "Avg:  ###.## °C"; Avg(1)
LOCATE 8, 65: PRINT USING "Min:  ###.## °C"; Min(1)
LOCATE 9, 65: PRINT USING "Rng:  ###.## °C"; Max(1) - Min(1)
LOCATE 10, 65: PRINT USING "Out:  ###.## °C"; T(3) / 2 + T(4) / 2

LOCATE 16, 65: PRINT "LGS STATUS"
LOCATE 17, 65: PRINT USING "Rmd:  ####.# kg"; Weight - Tare
LOCATE 18, 65: PRINT USING "Usd:  ####.# kg"; BeginWeight - Weight

LOCATE 20, 65: PRINT "SOLENOID STATUS"
IF Solenoid% = 0 THEN
 LOCATE 21, 65: PRINT "OFF            "
ELSE
 LOCATE 21, 65: PRINT USING "ON:  ###.## °C"; TempOff
END IF

'Finds limits of graph.
MaxL = -10000: MinL = 10000
FOR I = 1 TO 10
 IF Max(I) > MaxL THEN MaxL = Max(I)
 IF Min(I) < MinL THEN MinL = Min(I)
NEXT I

'Label temperature axis.
n = 0
FOR I = 22 TO 1 STEP -3
```

```
 value = (MaxL - MinL + 2) / 7 * n + MinL - 1
 n = n + 1
 LOCATE I, 1: COLOR 7, 0: PRINT USING "###.##"; value
NEXT I

'Assign coordinates to graph window and draw lines at -1 and -2°C.
WINDOW (1, MinL - 1)-(11, MaxL + 1): COLOR 7, 0
LINE (1, -1)-(11, -1): LINE (1, -2)-(11, -2)
LINE (1, -1.5)-(11, -1.5), , , &H8080

'Plot Max, Avg, and Min temperature in graph window.
n = 11
FOR I = 1 TO 10
 n = n - 1
 PSET (I, Max(n)), 14: PSET (I, Avg(n)), 14: PSET (I, Min(n)), 14
NEXT I

'Turn function keys on.
KEY(1) ON: KEY(2) ON: KEY(3) ON: KEY(4) ON

END SUB
```

SUB PrintTime

```
'Timer resets to zero every 86400 s (24 h) at midnight.
PrevTime = Time
Time = INT(TIMER + RollTime * 86400 - BeginTime)
IF Time = PrevTime THEN EXIT SUB
IF Time < PrevTime THEN RollTime = RollTime + 1

LoopTime = Time MOD 60
LOCATE 2, 65: COLOR 12, 0: PRINT USING "Elps: ######  s"; Time
LOCATE 3, 65: PRINT USING "Loop: ######  s"; LoopTime

'Calculates cyclic mean temperature every 5 s
IF LoopTime MOD 5 = 0 THEN
 IF Channel% = 2 THEN
  CALL GetTemp
 ELSE
  IniChannel% = Channel%
  Channel% = 2: CALL ChannelCodes: CALL ParallelPortOut: CALL GetTemp
  Channel% = IniChannel%
 END IF
```

```
 Sum2Temp = Sum2Temp + T(2): number = number + 1
 LOCATE 12, 65: COLOR 12, 0: PRINT "CONTROL SENSORS"
 LOCATE 13, 65: PRINT USING "Now:  ###.## °C"; T(2)
 LOCATE 14, 65: PRINT USING "Avg:  ###.## °C"; Sum2Temp / number
 IF Sum2Temp / number < -7 THEN CALL Alarm
END IF

END SUB
```

SUB Reinitialize

```
'Return registers to their initial value.
OUT &H378, IniData%: OUT &H379, IniStatus%: OUT &H37A, IniControl%

END SUB
```

SUB SoleControl

```
'Calculates the latest temperature ratio - that is the temperature of the
'air measured by the control sensors when the solenoid is de-energized
'divided by the cyclic mean temperature (time weighted average temperature
'of the control sensors between nitrogen injections e.g. Ave2Temp).
'Note:  Algorithm is based on Kelvin temperature scale.

FOR I = 1 TO 5: Ratio(I) = Ratio(I + 1): NEXT I '*
Ave2Temp = Sum2Temp / number
Ratio(6) = (TempOff + 273.15) / (Ave2Temp + 273.15) '**

CALL GetTemp

Sum2Temp = 0: number = 0

'Calculates the air temperature the solenoid should be turned off at so as
'to cause an average air temperature of -1.5°C during a 60 s cycle.
FOR I = 1 TO 6: Sum = Sum + Ratio(I): NEXT I '***
TempOff = 45.275 * Sum - 273.15 '****

'Decides if solenoid needs to be turned on and acts accordingly.
IF T(2) > -1.75 THEN

 Solenoid% = 1: CALL ParallelPortOut
 LOCATE 21, 65: PRINT USING "ON:   ###.## °C"; TempOff
 OPEN "C:\Chris\Output\" + FileName2$ FOR APPEND AS #2
```

```
WRITE #2, Time, Ratio(6), Ave2Temp, T(2), TempOff, Solenoid%: CLOSE #2

'Solenoid is forced on for minimum of 3 seconds.
DO: CALL PrintTime: LOOP UNTIL LoopTime = 23

DO
 CALL PrintTime: CALL GetTemp
 LOCATE 13, 65: PRINT USING "Now:  ###.## °C"; T(2)
LOOP UNTIL T(2) - TempOff < .01 OR LoopTime >= 59 OR Quit$ = "quit"

END IF

Solenoid% = 0: CALL ParallelPortOut
OPEN "C:\Chris\Output\" + FileName2$ FOR APPEND AS #2
WRITE #2, Time, Ratio(6), Ave2Temp, T(2), TempOff, Solenoid%: CLOSE #2
LOCATE 21, 65: PRINT "OFF           "
TempOff = T(2)

'Program 1 was different from this subroutine in that the above lines
'remarked with stars were replaced with these lines:
'*     For I = 1 to 2: Ratio(I) = Ratio(I+1): Next I
'**    Ratio(6) = (TempOff + 273.15) / (Ave2Temp + 273.15)
'***   FOR I = 1 TO 3: Sum = Sum + Ratio(I): NEXT I
'****  TempOff = 90.55 * Sum - 273.15

END SUB
```

SUB TakeReadings

```
'All key trapping is disabled during subroutine. If enabled, computer
'unexplainably loses cursor locations and values of variables.
KEY(1) OFF: KEY(2) OFF: KEY(3) OFF: KEY(4) OFF: KEY(15) OFF: KEY(16) OFF

'Stores ten previous max, min, and avg temperatures in arrays.
FOR I = 10 TO 2 STEP -1
 Max(I) = Max(I - 1): Avg(I) = Avg(I - 1): Min(I) = Min(I - 1)
NEXT I

FileTime = Time

'Read temperature sensors 5 through 24 and find max, min, and avg temperature.
Max(1) = -10000: Min(1) = 10000: Avg(1) = 0
FOR Channel% = 5 TO 24
```

```
 CALL PrintTime: CALL ChannelCodes: CALL ParallelPortOut: CALL GetTemp
 IF T(Channel%) > Max(1) THEN Max(1) = T(Channel%)
 IF T(Channel%) < Min(1) THEN Min(1) = T(Channel%)
 Avg(1) = Avg(1) + T(Channel%) / 20
NEXT Channel%

'Reads temperature sensors 3 and 4 which are located outside cabinet.
FOR Channel% = 3 TO 4
 CALL ChannelCodes: CALL ParallelPortOut: CALL GetTemp
NEXT Channel%

'Reads weight of nitrogen cylinder.
PRINT #4, "P": INPUT #4, Weight$
Weight = VAL(MID$(Weight$, 4, 5))

'Write temperature and weight readings to a file.
OPEN "C:\Chris\Output\" + FileName1$ FOR APPEND AS #1
WRITE #1, FileTime, Weight, T(2), T(3), T(4), T(5), T(6), T(7), T(8), T(9), T(10),
T(11), T(12), T(13), T(14), T(15), T(16), T(17), T(18), T(19), T(20), T(21), T(22),
T(23), T(24)
CLOSE #1

'All subsequent I/O during cycle is on channel 2. Set multiplexer
'for channel 2 and leave it there for remainder of loop.
Channel% = 2: CALL ChannelCodes: CALL ParallelPortOut

KEY(1) ON: KEY(2) ON: KEY(3) ON: KEY(4) ON
END SUB
```

APPENDIX E - FINITE ELEMENT MODELLING

E.1. Finite Element Program

The same finite element program was used to analyse both the saline water bag and master packaged meat problem. However, minor formatting changes were made to change the display of the output file for each of the problems and these changes are included in comment statements within the program.

This is the FORTRAN program:

```
      DIMENSION X(999),Y(999),E(1010,8),SKD(3,3),SK(999,999),FD(3),
     $F(999),CD(3,3),C(999,999),SKK(999,999),Q(999),TEMPIN(999),
     $U(999,999),SL(999,999),SLIJK(999)
C     ******************Variable List**********************
C     NP = # of nodal points
C     NE = # of elements
C     X() and Y() = X and Y coordinates of nodal points
C     E(I,1);E(I,2);E(I,3) = specify connectivity of nodes
C        i,j, and k of the element respectively.
C     E(I,4) = thermal conductivity of element (W/mK)
C     E(I,5) = density of the element (kg/m^3)
C     E(I,6) = specific heat of the element (J/kgK)
C     E(I,7) = convection side.  side ij = 1, etc; side 0 = none
C     E(I,8) = convection coefficient
C     TEMPI = intial temperature of all elements
C     TIMEINC = time increment between itinerations
C     TTIME = total time of simulation
C     SK() = stiffness matrix
C     F() = force matrix
C     C() = capacitance matrix
C     SKD() = elemental stiffness matrix
C     FD() = elemental force matrix
C     CD() = elemental capacitance matrix
C     SKK() and Q() = matrices assembled for transient portion
C     SLIJK() = stores the length of side ij, jk or ki
C     TEMPIN() = array which stores transient temperatures
C     TINFIN = temperature of convection fluid
C     *****************************************************
C
```

```
C     File 5 opened for input and file 6 and 7 are for output
      OPEN(5,file='input.txt')
      OPEN(6,file='output1.txt')
      OPEN(7,file='output2.txt')

C     Read and print number of elements and nodes
      READ(5,*) NP,NE
      WRITE(6,1) NP,NE
    1 FORMAT('NUMBER OF NODES:',I3,10X,'NUMBER OF ELEMENTS:',I4)
      WRITE(6,*)

C     Read and print X,Y coordinates of nodes
      WRITE(6,*) 'NODE   X COORDINATE      Y COORDINATE'
      WRITE(6,*) '*****************************************'
      DO I=1,NP
       READ(5,*) X(I),Y(I)
       WRITE(6,2) I,X(I),Y(I)
      END DO
    2 FORMAT(1X,I3,5X,F8.4,11X,F8.4)

C     Read and print element data
      WRITE(6,*)
      WRITE(6,*) 'ELEMEN  NODE  NODE  NODE  CONDUCT. DENSITY
SPECIFIC'
      WRITE(6,*) 'NUMBER  ONE  TWO  THREE  K           HEAT'
      WRITE(6,*) '**********************************************************'
      DO I=1,NE
       READ(5,*) E(I,1),E(I,2),E(I,3),E(I,4),E(I,5),E(I,6),E(I,7)
       E(I,8)=155.0
       WRITE(6,3) I,E(I,1),E(I,2),E(I,3),E(I,4),E(I,5),E(I,6)
      END DO
    3 FORMAT(1X,I4,5X,F4,2X,F4,3X,F4,2X,F5.3,4X,F6.1,3X,F6.1)
      WRITE(6,*)
      WRITE(6,*) 'ELEMENT   SIDE  CONVECTION  '
      WRITE(6,*) 'NUMBER          COEFFICIENT  '
      WRITE(6,*) '**********************************'
      DO I=1,NE
       IF (E(I,7).NE.0.0) THEN
        WRITE(6,4) I,E(I,7),E(I,8)
       ENDIF
      END DO
    4 FORMAT(1X,I4,7X,F3.1,6X,F6.2)
```

```
C    Read and print intial temp, time inc, and total time
     WRITE(6,*)
     WRITE(6,*) 'INITIAL TEMP.   TIME INC.   TOTAL TIME'
     WRITE(6,*) '****************************************'
     READ(5,*) TEMPI,TIMEINC,TTIME
     WRITE(6,5) TEMPI,TIMEINC,TTIME
   5 FORMAT(4X,F5.2,10X,F4,9X,F6)

C    Store initial temperatures in temperature array
     DO I=1,999
       TEMPIN(I)=TEMPI
     END DO
     ITIME=0.0

C    Initialize stiffness and capacitance matrix
     DO I=1,999
      DO J=1,999
       SK(I,J)=0.0
       C(I,J)=0.0
      END DO
     END DO

C    Formation of elemental stiffness and capacitance matrix
     DO I=1,NE
      FD(1)=0.0
      FD(2)=0.0
      FD(3)=0.0
      XI=X(E(I,1))
      YI=Y(E(I,1))
      XJ=X(E(I,2))
      YJ=Y(E(I,2))
      XK=X(E(I,3))
      YK=Y(E(I,3))
      AI=XJ*YK-XK*YJ
      AJ=XK*YI-XI*YK
      AK=XI*YJ-XJ*YI
      BI=YJ-YK
      BJ=YK-YI
      BK=YI-YJ
      CI=XK-XJ
      CJ=XI-XK
      CK=XJ-XI
      A=0.5*(XJ*YK+XI*YJ+XK*YI-XJ*YI-XI*YK-XK*YJ)
```

```
      SKD(1,1)=0.25*E(I,4)/A*(BI**2+CI**2)
      SKD(1,2)=0.25*E(I,4)/A*(BI*BJ+CI*CJ)
      SKD(1,3)=0.25*E(I,4)/A*(BI*BK+CI*CK)
      SKD(2,1)=0.25*E(I,4)/A*(BI*BJ+CI*CJ)
      SKD(2,2)=0.25*E(I,4)/A*(BJ**2+CJ**2)
      SKD(2,3)=0.25*E(I,4)/A*(BJ*BK+CJ*K)
      SKD(3,1)=0.25*E(I,4)/A*(BI*BK+CI*CK)
      SKD(3,2)=0.25*E(I,4)/A*(BJ*BK+CJ*CK)
      SKD(3,3)=0.25*E(I,4)/A*(BK**2+CK**2)
      CD(1,1)=A*E(I,5)*E(I,6)/6.0
      CD(1,2)=A*E(I,5)*E(I,6)/12.0
      CD(1,3)=CD(1,2)
      CD(2,1)=CD(1,2)
      CD(2,2)=CD(1,1)
      CD(2,3)=CD(1,2)
      CD(3,1)=CD(1,2)
      CD(3,2)=CD(1,2)
      CD(3,3)=CD(1,1)

C     Inclusion of convective terms to elemental stiffness matrix
      IF (E(I,7).EQ.1.0) THEN
       SLIJ=((XJ-XI)**2+(YJ-YI)**2)**0.5
       SLIJK(I)=SLIJ
       SKD(1,1)=SKD(1,1)+E(I,8)*SLIJ/3.0
       SKD(1,2)=SKD(1,2)+E(I,8)*SLIJ/6.0
       SKD(2,1)=SKD(2,1)+E(I,8)*SLIJ/6.0
       SKD(2,2)=SKD(2,2)+E(I,8)*SLIJ/3.0
      ENDIF
      IF (E(I,7).EQ.2.0) THEN
       SLJK=((XK-XJ)**2+(YK-YJ)**2)**0.5
       SLIJK(I)=SLJK
       SKD(2,2)=SKD(2,2)+E(I,8)*SLJK/3.0
       SKD(2,3)=SKD(2,3)+E(I,8)*SLJK/6.0
       SKD(3,2)=SKD(3,2)+E(I,8)*SLJK/6.0
       SKD(3,3)=SKD(3,3)+E(I,8)*SLJK/3.0
      ENDIF
      IF (E(I,7).EQ.3.0) THEN
       SLKI=((XI-XK)**2+(YI-YK)**2)**0.5
       SLIJK(I)=SLKI
       SKD(1,1)=SKD(1,1)+E(I,8)*SLKI/3.0
       SKD(1,3)=SKD(1,3)+E(I,8)*SLKI/6.0
       SKD(3,1)=SKD(3,1)+E(I,8)*SLKI/6.0
       SKD(3,3)=SKD(3,3)+E(I,8)*SLKI/3.0
```

```
      ENDIF

C     Assembles global stiffness and capacitance matrix
      DO J=1,3
       DO K=1,3
        SK(E(I,J),E(I,K))=SK(E(I,J),E(I,K))+SKD(J,K)
        C(E(I,J),E(I,K))=C(E(I,J),E(I,K))+CD(J,K)
       END DO
      END DO
     END DO

C     Converts problem into a transient problem
     DO I=1,NP
      DO J=1,NP
       SKK(I,J)=SK(I,J)+C(I,J)/TIMEINC
      END DO
     END DO

C     Transient loop part of program
   49 IF (ITIME.LE.TTIME) THEN

C     Initializes global force matrix
      DO I=1,NP
       F(I) = 0
      END DO

      READ (5,*) TINFIN

C     Assembles elemental force matrix from convective data
      DO I=1,NE
       FD(1)=0.0
       FD(2)=0.0
       FD(3)=0.0
       IF (E(I,7).EQ.1.0) THEN
        FD(1)=E(I,8)*TINFIN*SLIJK(I)/2.
          FD(2)=FD(1)
       ENDIF
       IF (E(I,7).EQ.2.0) THEN
        FD(2)=E(I,8)*TINFIN*SLIJK(I)/2.0
          FD(3)=FD(2)
         ENDIF
         IF (E(I,7).EQ.3.0) THEN
        FD(1)=E(I,8)*TINFIN*SLIJK(I)/2.0
```

```
       FD(3)=FD(1)
      ENDIF

C      Assembles global force matrix
     DO J = 1,3
      F(E(I,J))=F(E(I,J))+FD(J)
       END DO
    END DO

C     Prints temperature output from each transient loop
C     of the meat simulation.
    WRITE(6,*)
    WRITE(6,*)
    WRITE(6,*) 'TIME=====>',ITIME
    WRITE(6,15) TINFIN
 15 FORMAT('Tinfinite: ',F6.2)
    WRITE(6,*) '              Outer Profile    Center Profile'
    WRITE(6,6) TEMPIN(522),TEMPIN(537)
  6 FORMAT('N2/CO2         ',4X,F7.2,10X,F7.2)
    WRITE(6,7) TEMPIN(485),TEMPIN(500)
    WRITE(6,7) TEMPIN(448),TEMPIN(463)
    WRITE(6,7) TEMPIN(411),TEMPIN(426)
    WRITE(6,7) TEMPIN(374),TEMPIN(389)
    WRITE(6,7) TEMPIN(337),TEMPIN(352)
  7 FORMAT('CO2            ',4X,F7.2,10X,F7.2)
    WRITE(6,8) TEMPIN(300),TEMPIN(315)
  8 FORMAT('C02/MEAT       ',4X,F7.2,10X,F7.2)
    WRITE(6,9) TEMPIN(263),TEMPIN(278)
    WRITE(6,9) TEMPIN(226),TEMPIN(241)
    WRITE(7,*) TEMPIN(241)
    WRITE(6,9) TEMPIN(189),TEMPIN(204)
  9 FORMAT('MEAT           ',4X,F7.2,10X,F7.2)
    WRITE(6,10) TEMPIN(152),TEMPIN(167)
 10 FORMAT('MEAT/MEAT TRAY ',4X,F7.2,10X,F7.2)
    WRITE(6,11) TEMPIN(115),TEMPIN(130)
 11 FORMAT('MEAT TRAY      ',4X,F7.2,10X,F7.2)
    WRITE(6,12) TEMPIN(78),TEMPIN(93)
 12 FORMAT('MEAT TRAY/TRAY ',4X,F7.2,10X,F7.2)
    WRITE(6,13) TEMPIN(41),TEMPIN(56)
 13 FORMAT('TRAY           ',4X,F7.2,10X,F7.2)
    WRITE(6,14) TEMPIN(4),TEMPIN(19)
 14 FORMAT('TRAY/N2        ',4X,F7.2,10X,F7.2)
```

```
C    Prints temperature output from each transient loop
C    of the saline water bag simulation.
C     WRITE(6,*)
C     WRITE(6,*)
C     WRITE(6,*) 'TIME=====>',ITIME
C     WRITE(6,15) TINFIN
C     15 FORMAT('Tinfinite:  ',F6.2)
C     WRITE(6,*) '                Center Profile'
C     WRITE(6,) TEMPIN(222)
C     6 FORMAT('N2/AIR          ',4X,F7.2,10X,F7.2)
C     WRITE(6,7) TEMPIN(201)
C     7 FORMAT('AIR/H2O         ',4X,F7.2,10X,F7.2)
C     WRITE(6,8) TEMPIN(171)
C     WRITE(6,8) TEMPIN(140)
C     WRITE(6,8) TEMPIN(109)
C     WRITE(6,8) TEMPIN(78)
C     8 FORMAT('H2O             ',4X,F7.2,10X,F7.2)
C     WRITE(6,9) TEMPIN(47)
C     WRITE(7,9) TEMPIN(47)
C     9 FORMAT('H2O/WOOD        ',4X,F7.2,10X,F7.2)
C     WRITE(6,10) TEMPIN(16)
C    10 FORMAT('WOOD/N2         ',4X,F7.2,10X,F7.2)

C    More time increment garbage
    ITIME=ITIME+TIMEINC

    SUM=0.0
    DO I=1,NP
     DO J=1,NP
      SUM=SUM+C(I,J)*TEMPIN(J)/TIMEINC
     END DO
     Q(I)=SUM+F(I)
     SUM=0.0
    END DO

C    LDU DECOMPOSITION SOLVER
C    Sets diagonals in U to 1
    DO I=1,NP
     TEMPIN(I)=0.0
     U(I,I)=1.0
    END DO
C    Forming U and L from stiffness Matrix
    INCREM=0
```

```
      DO J=1,NP
       INCREM=INCREM+1
       I=INCREM
  50   IF (I.LE.NP) THEN
        IF (J.GT.1) THEN
         DO K=1,(J-1)
          SUML=SUML+SL(I,K)*U(K,J)
          SUMU=SUMU+SL(J,K)*U(K,I)
         END DO
        ENDIF
        SL(I,J)=SKK(I,J)-SUML
        U(J,I)=(SKK(J,I)-SUMU)/SL(J,J)
        SUML=0.0
        SUMU=0.0
        I=I+1
        GOTO 50
       ENDIF
      END DO
C     Modifying force matrix
      Q(1)=Q(1)/SL(1,1)
      DO I=2,NP
       SUMF=0.0
       DO K=1,(I-1)
        SUMF=SUMF+SL(I,K)*Q(K)
       END DO
       Q(I)=(Q(I)-SUMF)/SL(I,I)
      END DO

C     Back substitution
      TEMPIN(NP)=Q(NP)
      J=NP
 51   IF (J.NE.1) THEN
       J=J-1
       DO K=J,NP
        SUMX=SUMX+U(J,K)*TEMPIN(K)
       END DO
       TEMPIN(J)=Q(J)-SUMX
       SUMX=0.0
       GO TO 51
      ENDIF
      GO TO 49
     ENDIF
     END
```

E.2. Finite Element Mesh

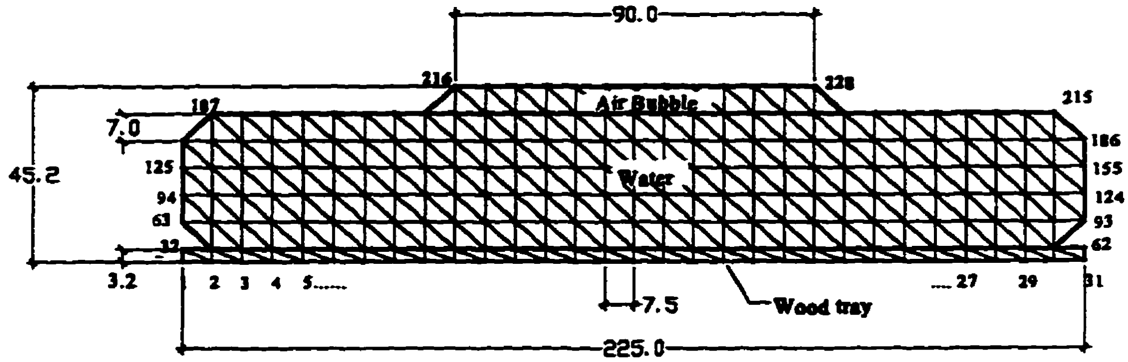

Figure E.1. Finite element mesh for the saline water bag with dimensions (mm) and node numbering.

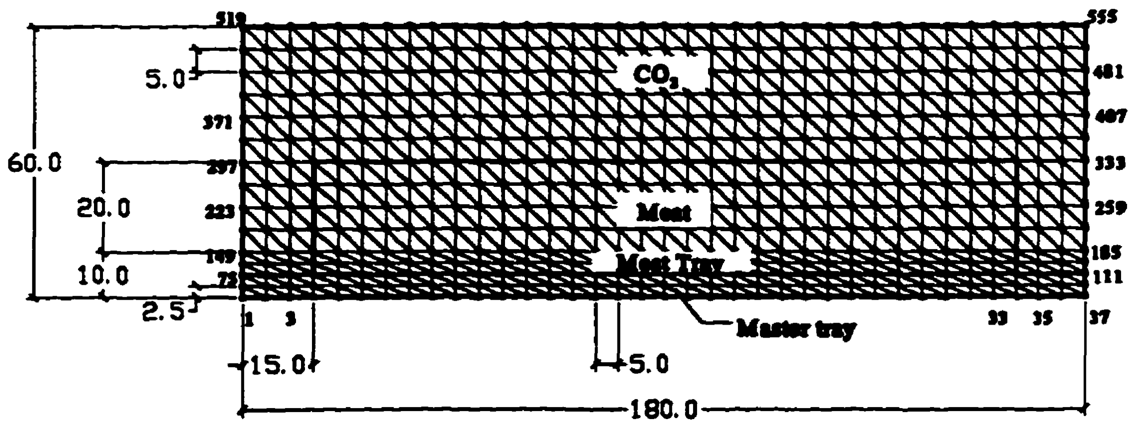

Figure E.2. Finite element mesh for the master package with dimensions (mm) and node numbering.

E.3. Determination of Convective Heat Transfer Coefficient

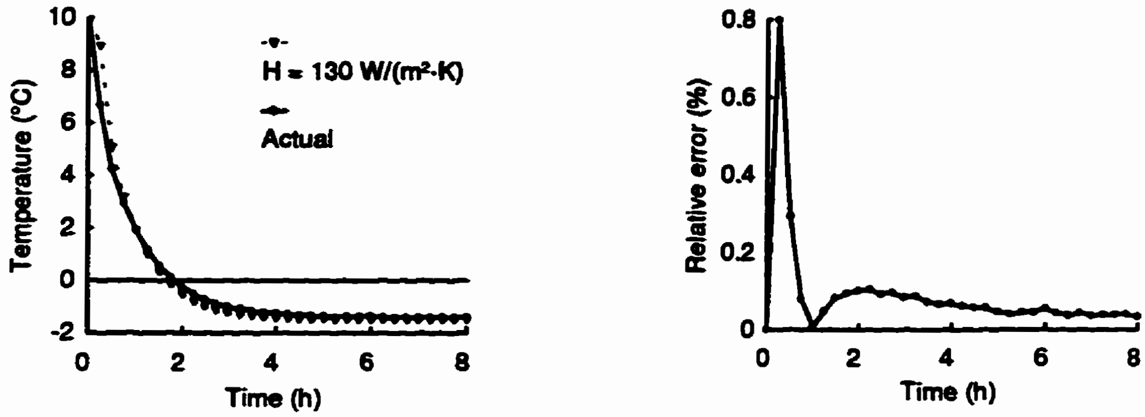

Figure E.3. Comparison of measured and predicted temperatures of water bags when H = 130 W/(m^2•K); and the relative error of prediction (temperature outside container was 30°C and four fans were used).

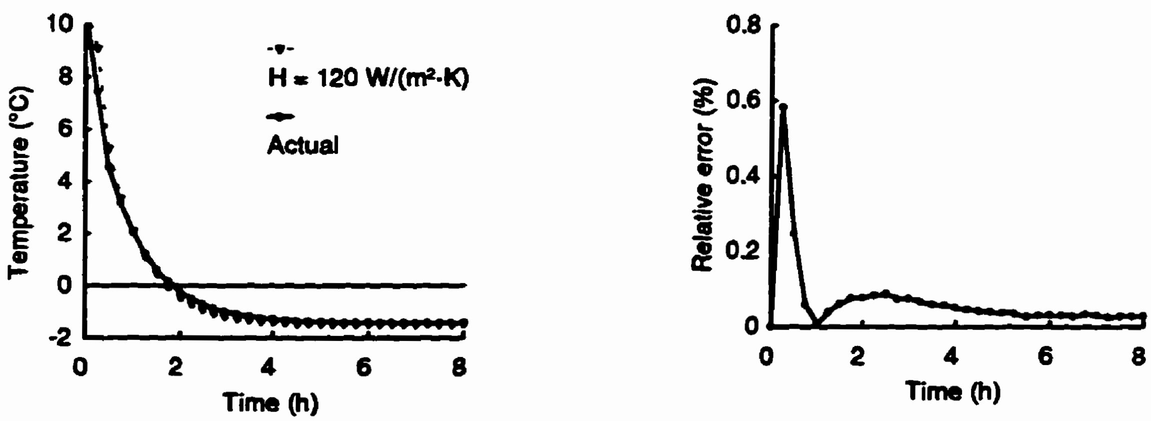

Figure E.4. Comparison of measured and predicted temperatures of water bags when H = 120 W/(m^2•K); and the relative error of prediction (temperature outside container was 15°C and four fans were used).

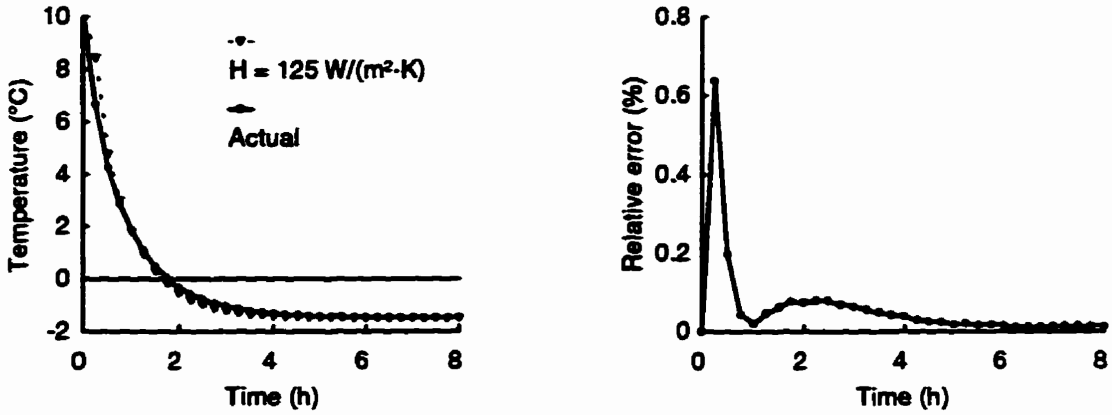

Figure E.5. Comparison of measured and predicted temperatures of water bags when H = 125 W/(m²•K); and the relative error of prediction (temperature outside container was 0°C and four fans were used).

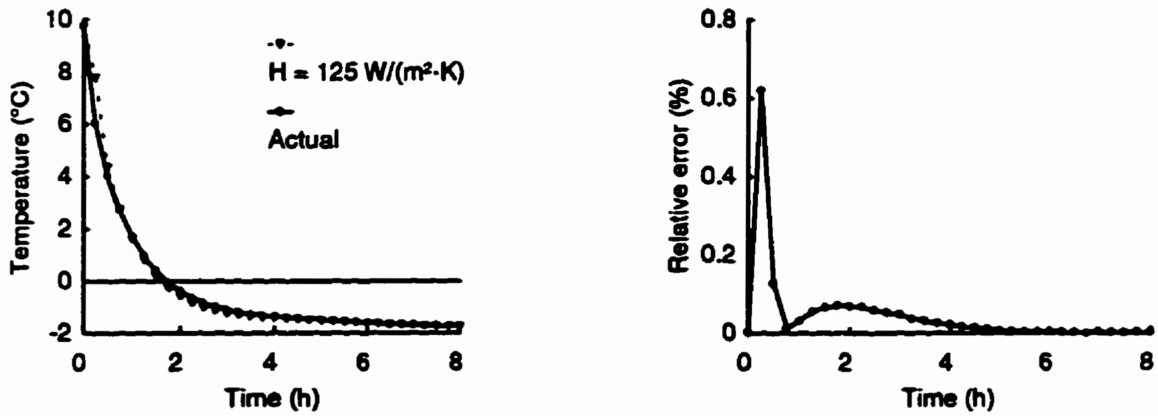

Figure E.6. Comparison of measured and predicted temperatures of water bags when H = 125 W/(m²•K); and the relative error of prediction (temperature outside container was -15°C and four fans were used).

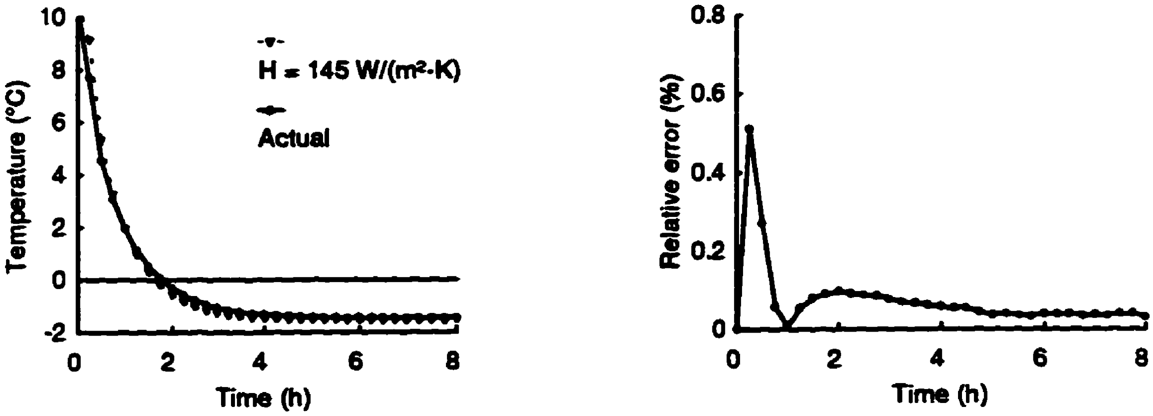

Figure E.7. **Comparison of measured and predicted temperatures of water bags when H = 145 W/(m^2•K); and the relative error of prediction (temperature outside container was 30°C and six fans were used in trial 1).**

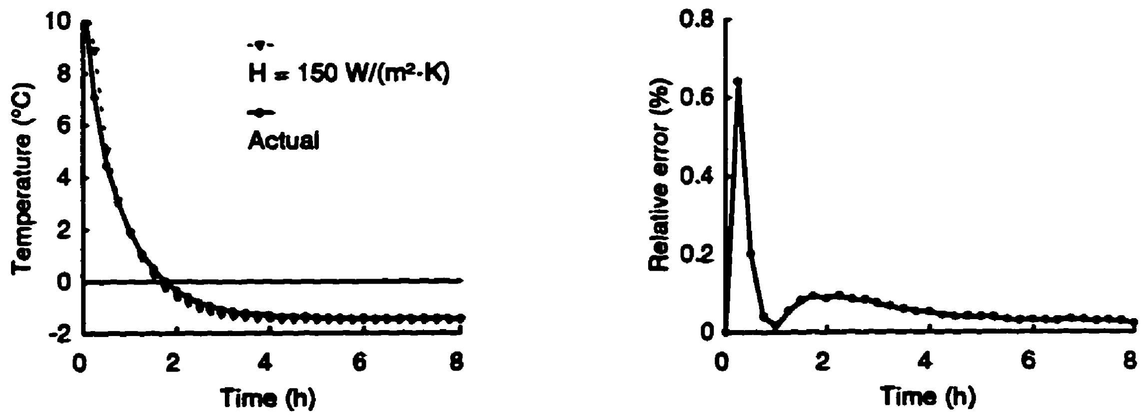

Figure E.8. **Comparison of measured and predicted temperatures of water bags when H = 150 W/(m^2•K); and the relative error of prediction (temperature outside container was 30°C and six fans were used in trial 2).**

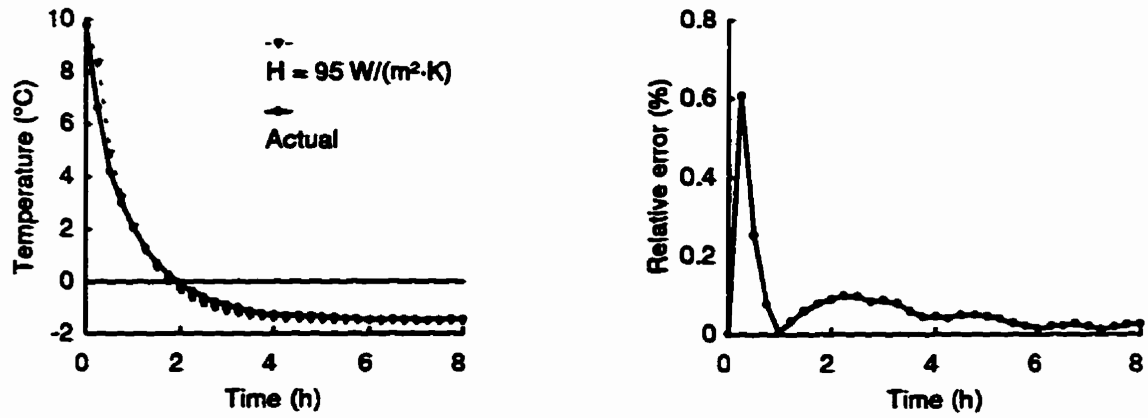

Figure E.9. Comparison of measured and predicted temperatures of water bags when H = 95 W/(m²•K); and the relative error of prediction (temperature outside container was 30°C and two fans were used in trial 1).

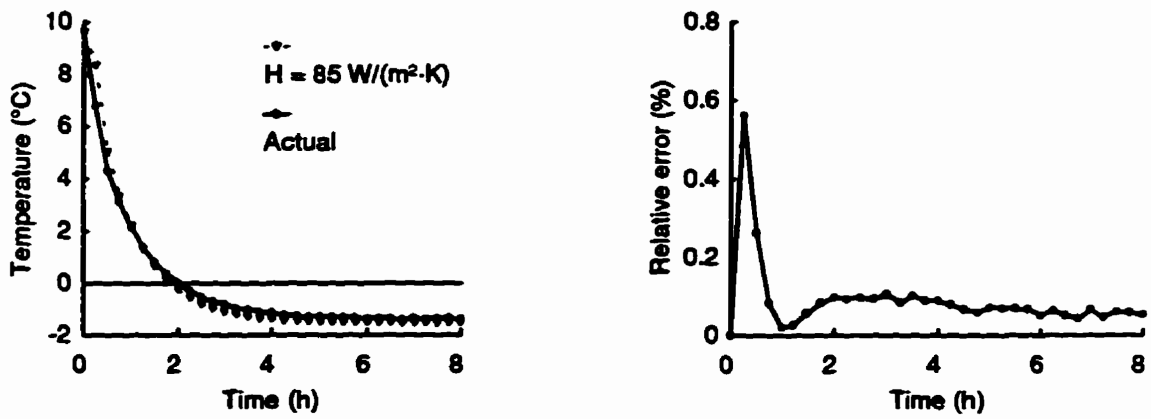

Figure E.10. Comparison of measured and predicted temperatures of water bags when H = 85 W/(m²•K); and the relative error of prediction (temperature outside container was 30°C and two fans were used in trial 2).

APPENDIX F - THERMAL RESISTANCE OF THE CONTAINER

For the purposes of determining the thermal resistance of the container, the effects of convection and radiation were ignored. Small temperature gradients make the contribution to heat flow by radiation negligible. As well, the thermal resistance due to convection was ignored because of the assumption of a minute boundary layer. This reflects a worst case scenario, predicting a lower thermal resistance for the container than might be experienced. However, ignoring the thermal resistance contributed by convection may be justified by recognizing that the thermal boundary layer will be small because of high quantities of air flow in and outside the container.

Theoretically, the equation for predicting the thermal resistance of a wall due to conduction is as follows:

$$R = \frac{L}{KA} \qquad \text{(F.1)}$$

where:

R	= thermal resistance (K/W)	L	= length of heat flow path (m)
K	= thermal conductivity (W/(m•K))	A	= cross sectional area (m^2)

The walls of the container vary in thickness from front to back (by approximately 12 mm). It is suspected that this is a requirement of the moulding process for the polyethylene exterior of the container. An average wall thickness of 51 mm was assumed. Two layers of the polyethylene shell (each 6.4 mm thick) sandwich the polyurethane insulation (38.2 mm thick). The thermal conductivity of the polyurethane was assumed to be 0.026 W/(m•K) (Incropera and De Witt 1990) and that of the polyethylene was

0.41 W/(m•K) (ASHRAE 1993). Using a value of 8.3 m^2 for the interior surface area of the container, and substituting the above values into Equation F.1., the following values were obtained:

$$R_{polyurethane} = \frac{L}{KA} = \frac{0.0382}{0.026(8.3)} = 177 \times 10^{-3}\ K/W$$

$$R_{polyethylene} = \frac{L}{KA} = \frac{2(0.0064)}{0.410(8.3)} = 3.76 \times 10^{-3}\ K/W \qquad (F.2)$$

$$R_{total} = R_{polyurethane} + R_{polyethylene} = 181 \times 10^{-3}\ K/W$$

APPENDIX G - TEMPERATURE AND N_2 USE DATA

G.1. Two Fans with an Outside Temperature of 30°C

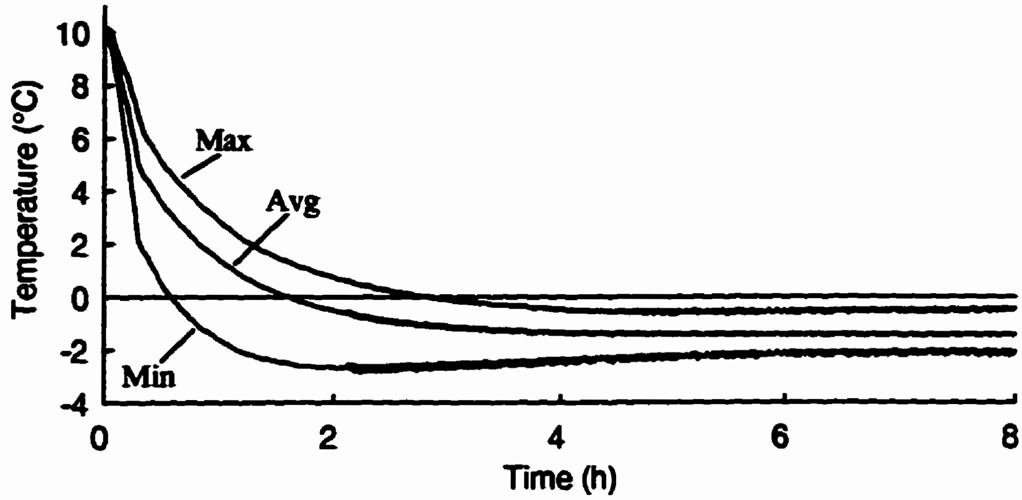

Figure G.1. **Minimum, maximum, and average temperatures (Trial 1) of saline water bags when the container was equipped with two ventilation fans, a N_2 tank pressure of 241 kPa, and was exposed to an outside temperature of 30°C.**

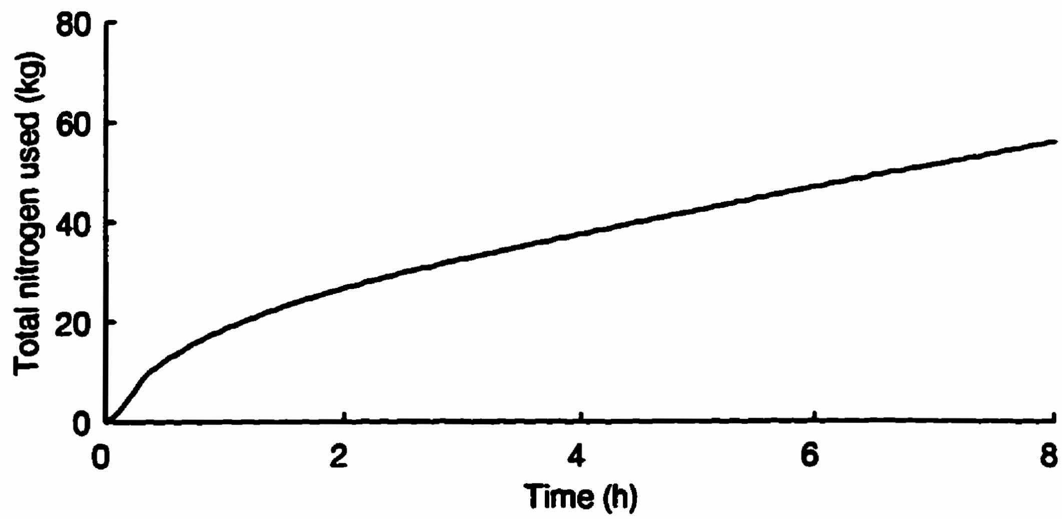

Figure G.2. **Nitrogen consumption during Trial 1 (Figure G.1.).**

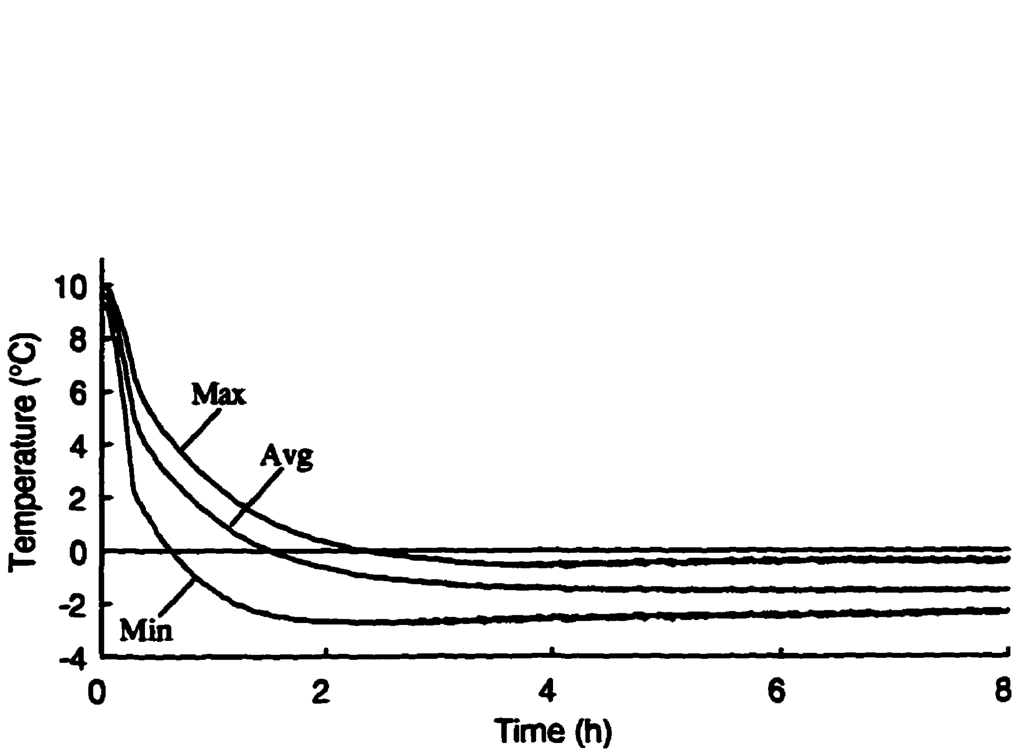

Figure G.3. Minimum, maximum, and average temperatures (Trial 2) of saline water bags when the container was equipped with two ventilation fans, a N_2 tank pressure of 241 kPa, and was exposed to an outside temperature of 30°C.

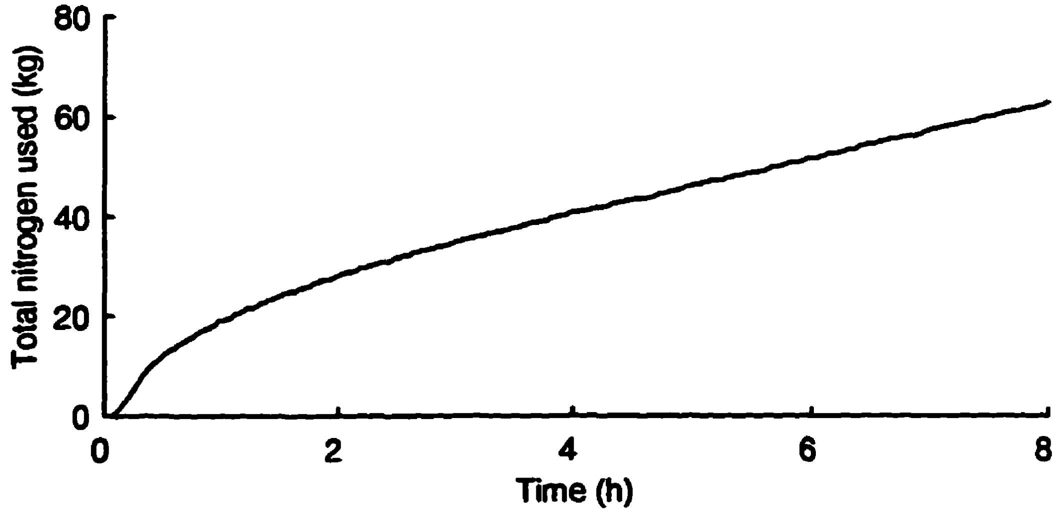

Figure G.4. Nitrogen consumption during Trial 2 (Figure G.3.).

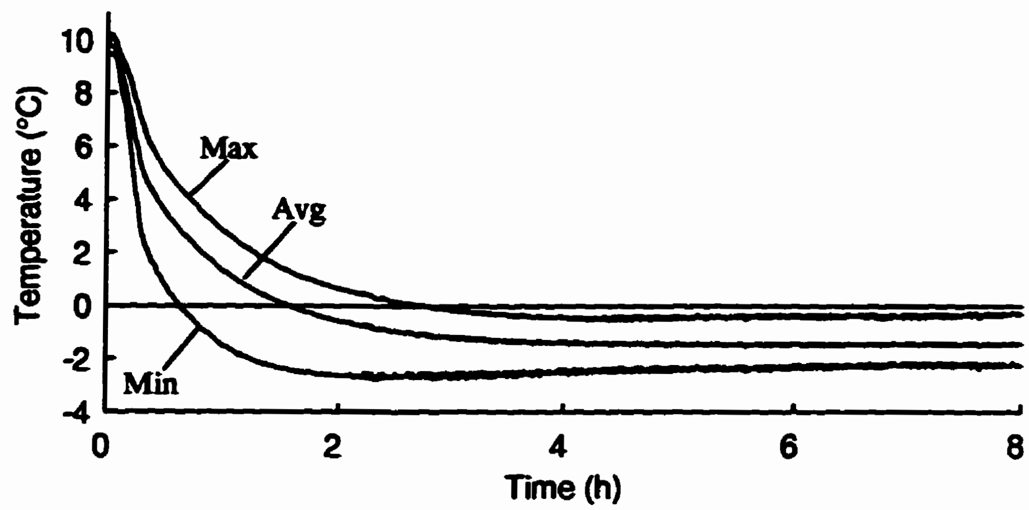

Figure G.5. Minimum, maximum, and average temperatures (Trial 3) of saline water bags when the container was equipped with two ventilation fans, a N_2 tank pressure of 152 kPa, and was exposed to an outside temperature of 30°C.

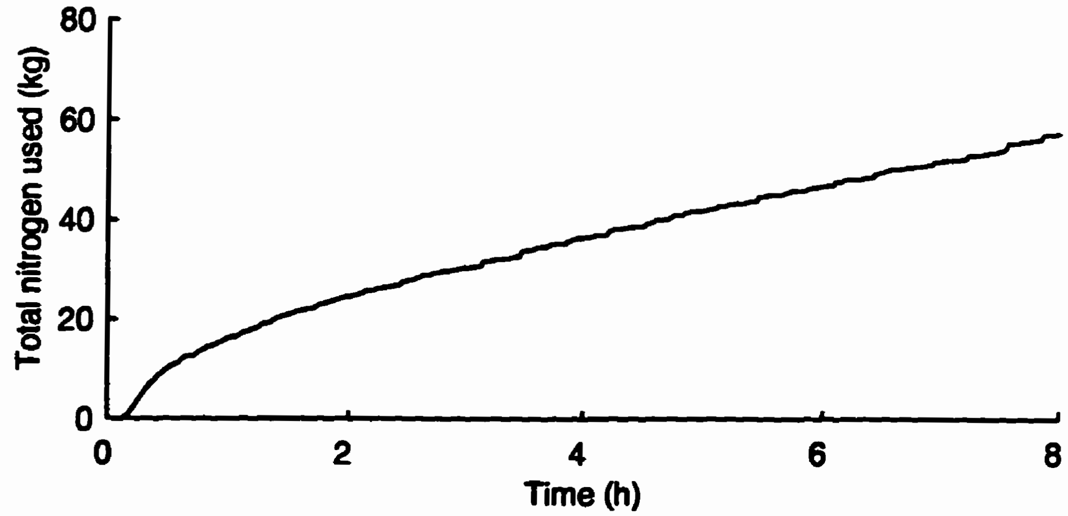

Figure G.6. Nitrogen consumption during Trial 3 (Figure G.5.).

G.2. Four Fans with an Outside Temperature of 30°C

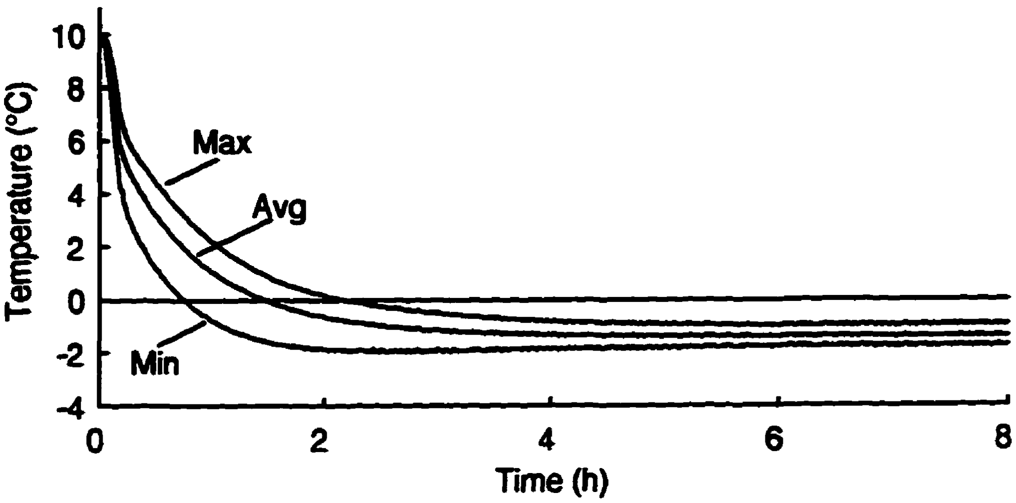

Figure G.7. **Minimum, maximum, and average temperatures (Trial 1) of saline water bags when the container was equipped with four ventilation fans, a N_2 tank pressure of 241 kPa, and was exposed to an outside temperature of 30°C.**

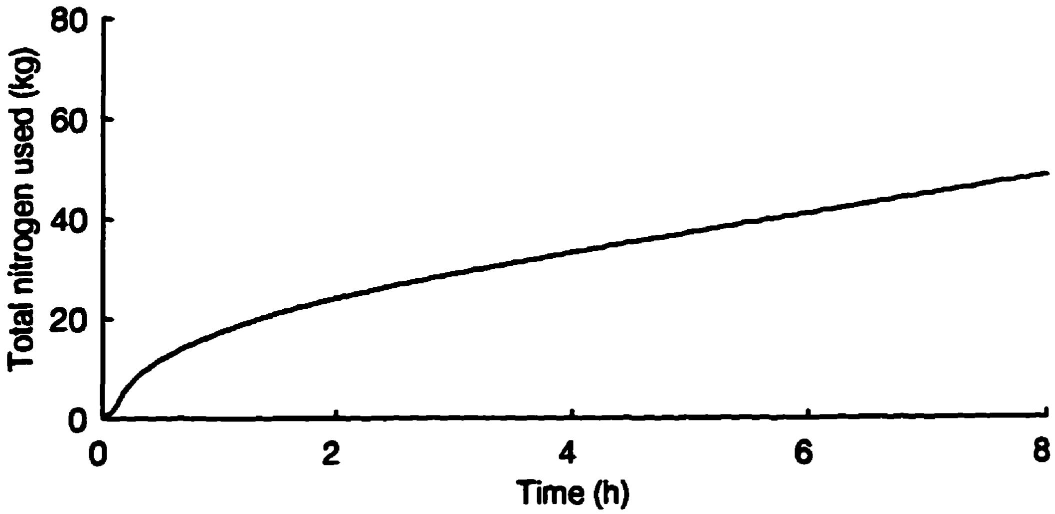

Figure G.8. **Nitrogen consumption during Trial 1 (Figure G.7.).**

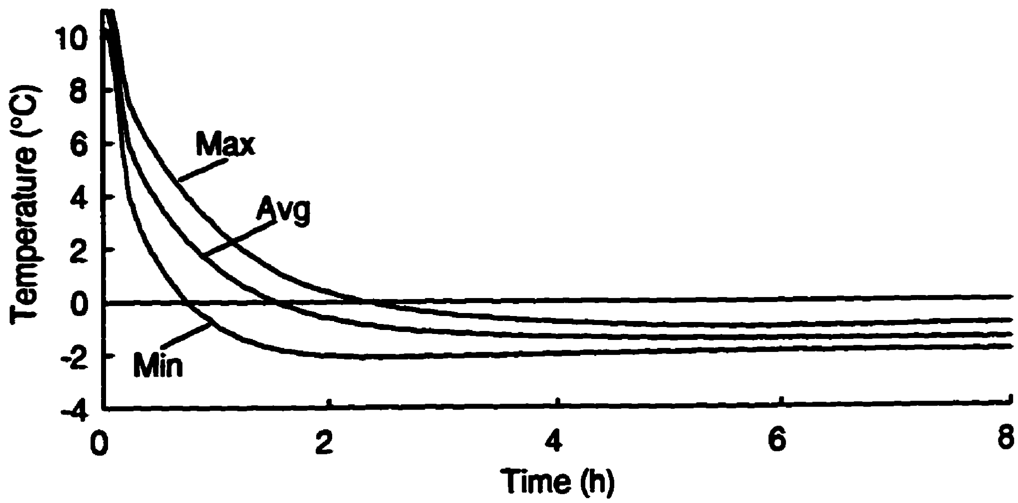

Figure G.9. **Minimum, maximum, and average temperatures (Trial 2) of saline water bags when the container was equipped with four ventilation fans, a N_2 tank pressure of 241 kPa, and was exposed to an outside temperature of 30°C.**

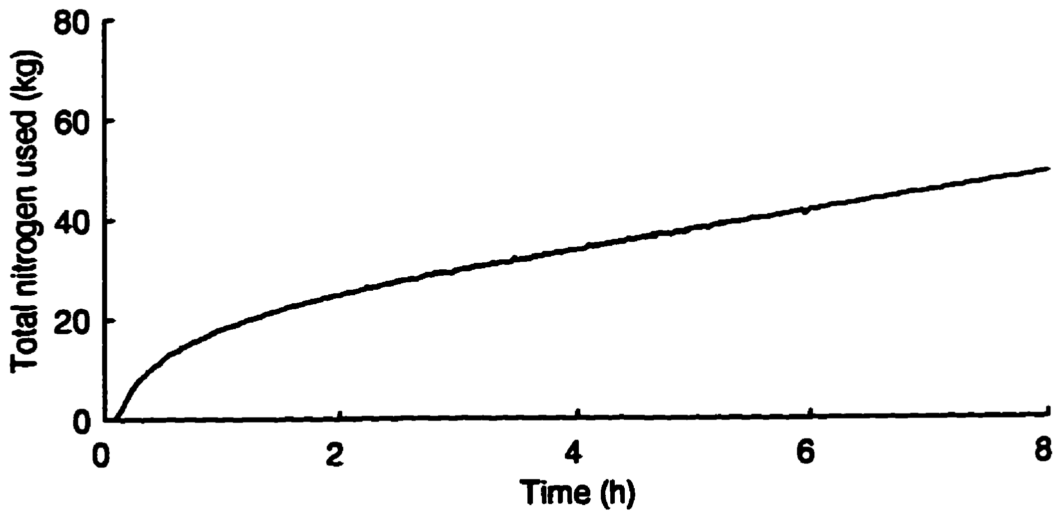

Figure G.10. Nitrogen consumption during Trial 2 (Figure G.9.).

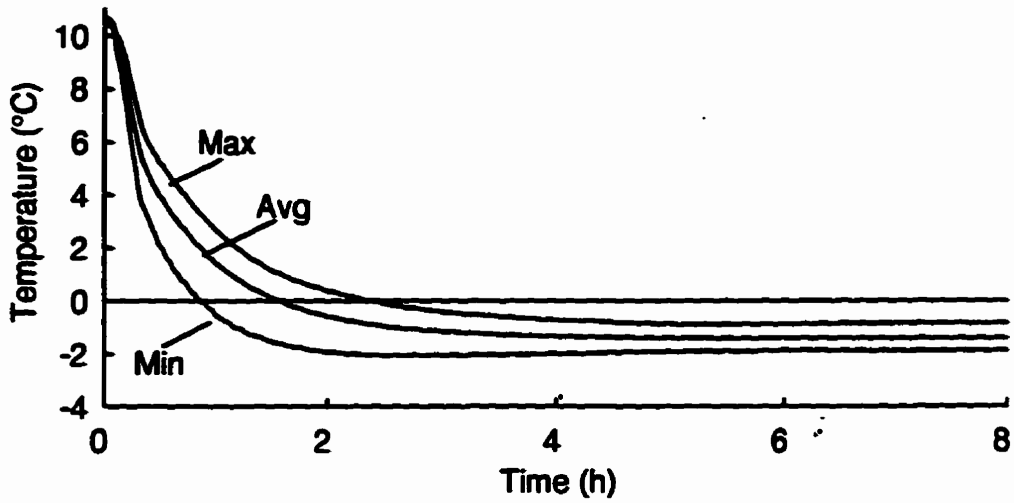

Figure G.11. Minimum, maximum, and average temperatures (Trial 3) of saline water bags when the container was equipped with four ventilation fans, a N_2 tank pressure of 241 kPa, and was exposed to an outside temperature of 30°C.

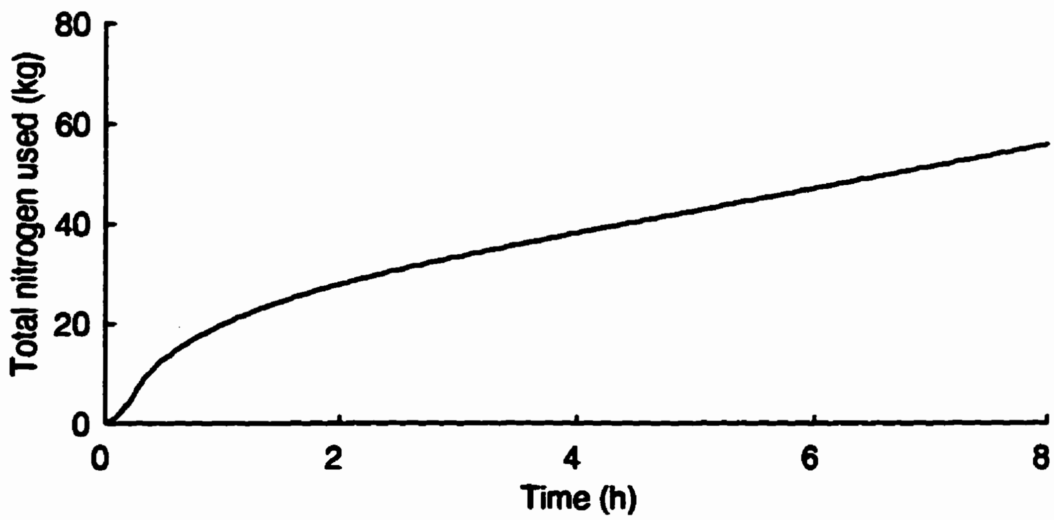

Figure G.12. Nitrogen consumption during Trial 3 (Figure G.11.).

G.3. Six Fans with an Outside Temperature of 30°C

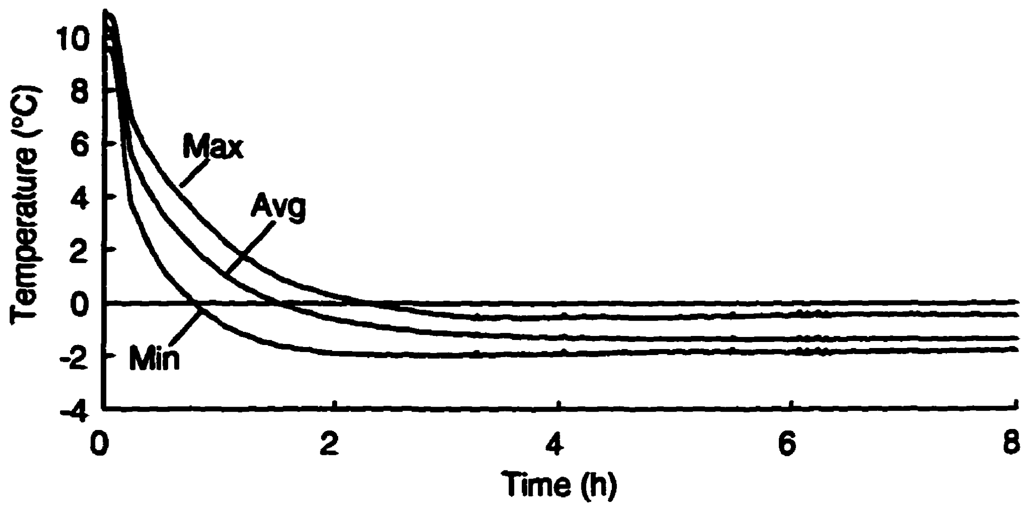

Figure G.13. Minimum, maximum, and average temperatures (Trial 1) of saline water bags when the container was equipped with six ventilation fans, a N_2 tank pressure of 241 kPa, and was exposed to an outside temperature of 30°C.

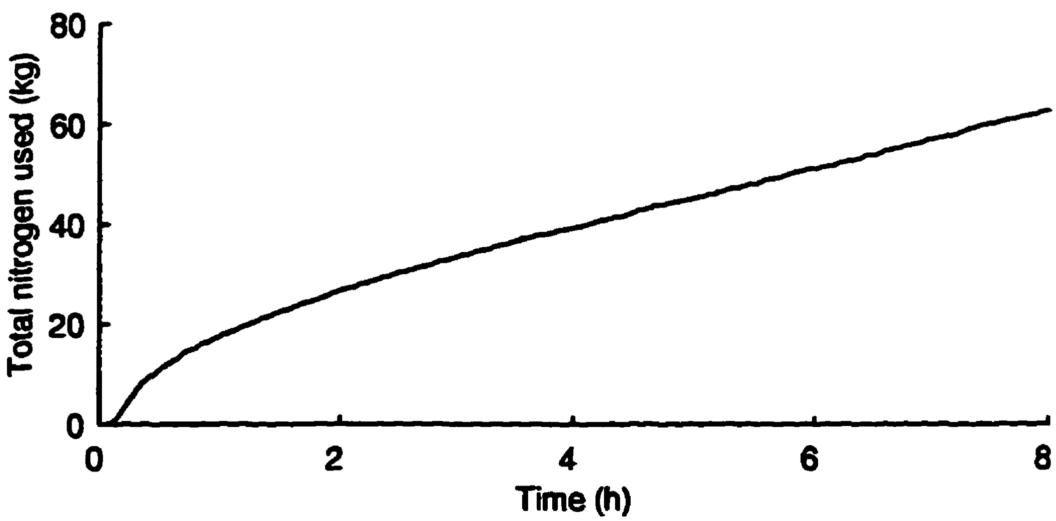

Figure G.14. Nitrogen consumption during Trial 1 (Figure G.13.).

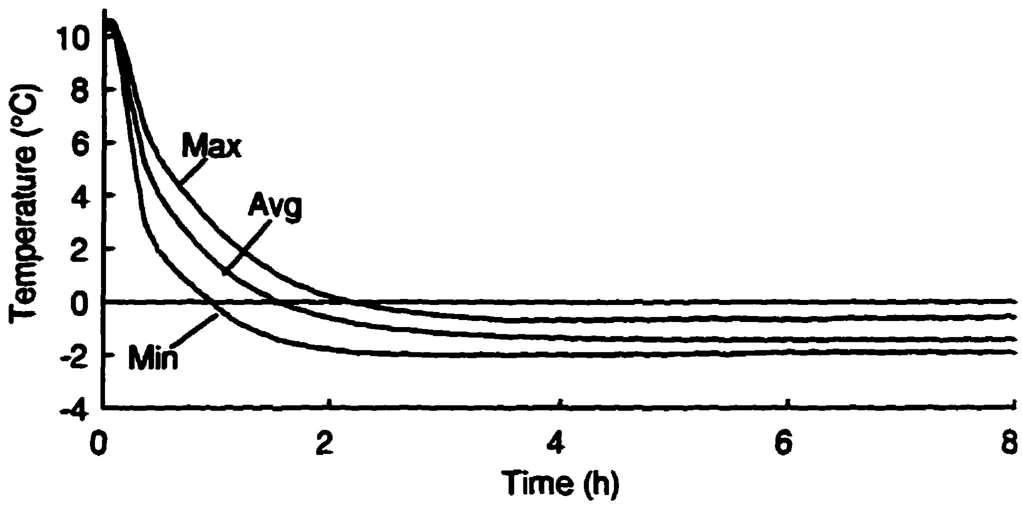

Figure G.15. Minimum, maximum, and average temperatures (Trial 2) of saline water bags when the container was equipped with six ventilation fans, a N_2 tank pressure of 152 kPa, and was exposed to an outside temperature of 30°C.

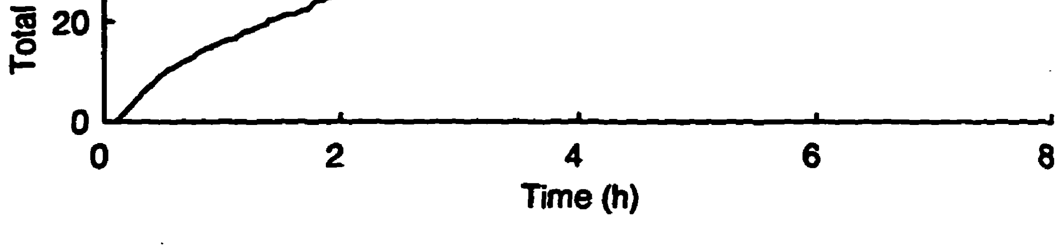

Figure G.16. Nitrogen consumption during Trial 2 (Figure G.15.).

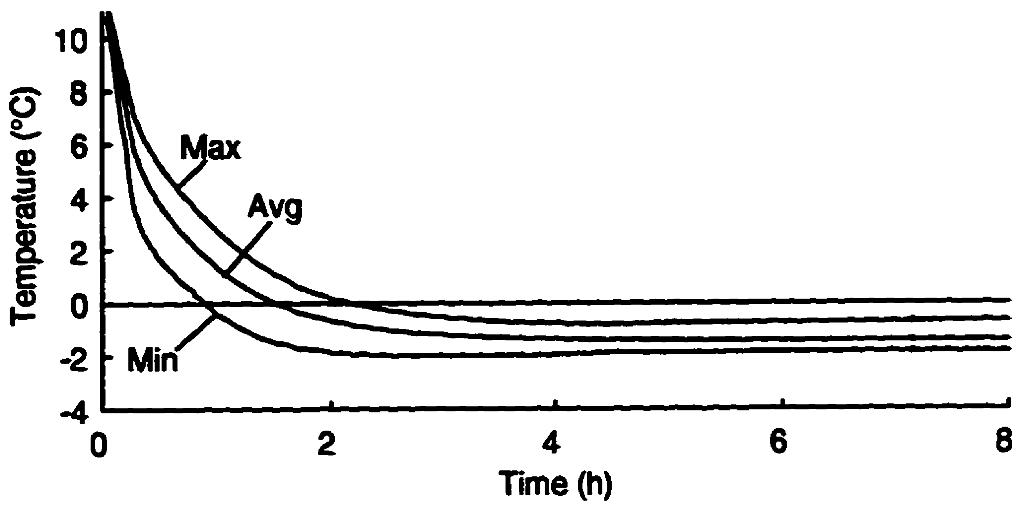

Figure G.17. Minimum, maximum, and average temperatures (Trial 3) of saline water bags when the container was equipped with six ventilation fans, a N_2 tank pressure of 241 kPa, and was exposed to an outside temperature of 30°C.

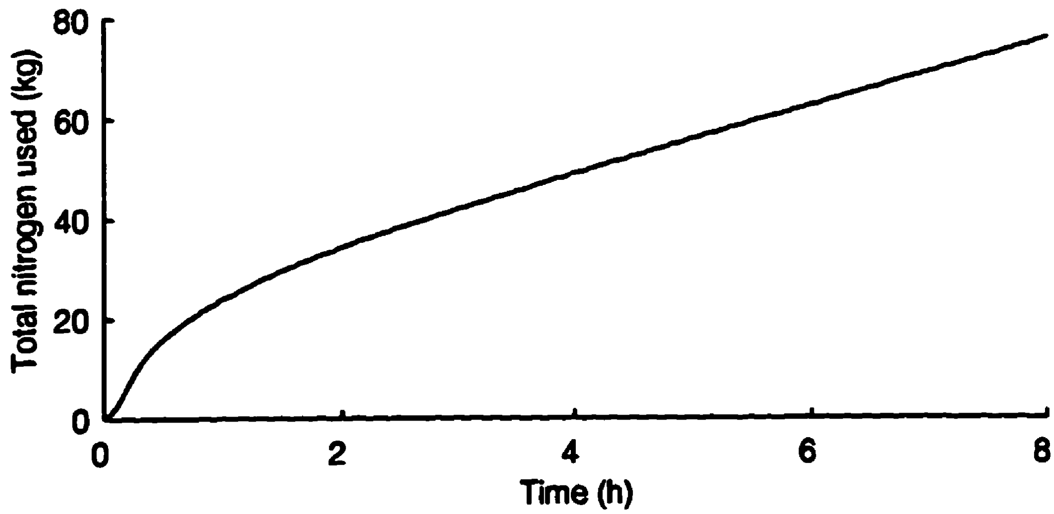

Figure G.18. Nitrogen consumption during Trial 3 (Figure G.17.).

G.4. Four Fans with an Outside Temperature of 15°C

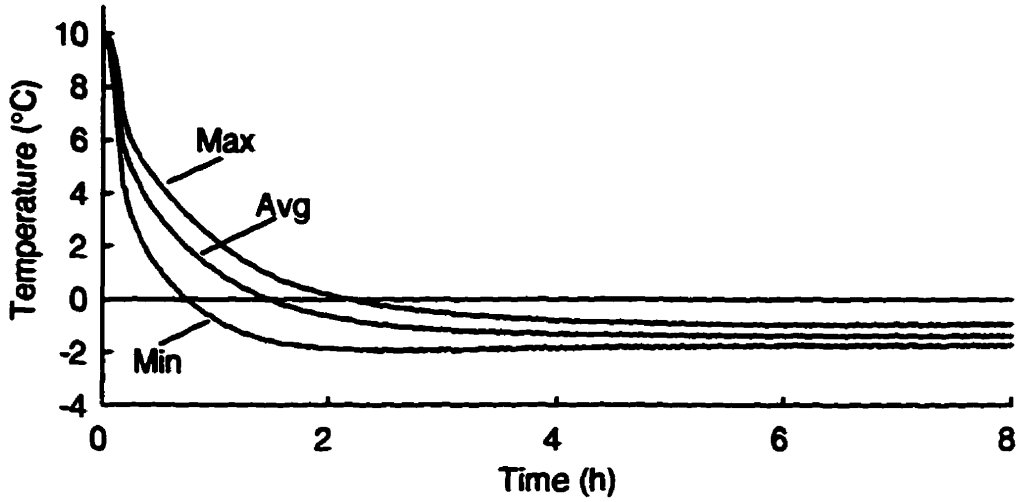

Figure G.19. Minimum, maximum, and average temperatures (Trial 1) of saline water bags when the container was equipped with four ventilation fans, a N_2 tank pressure of 241 kPa, and was exposed to an outside temperature of 15°C.

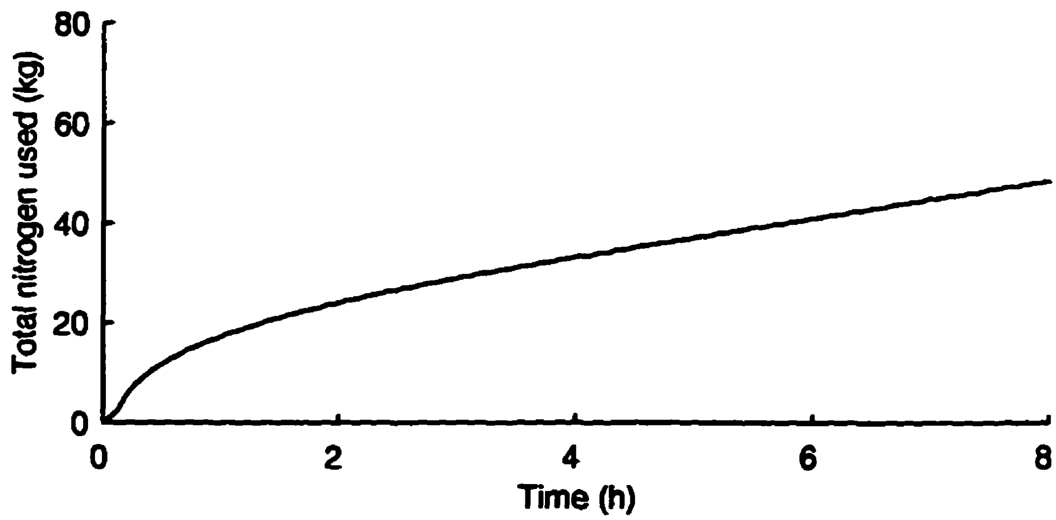

Figure G.20. Nitrogen consumption during Trial 1 (Figure G.19.).

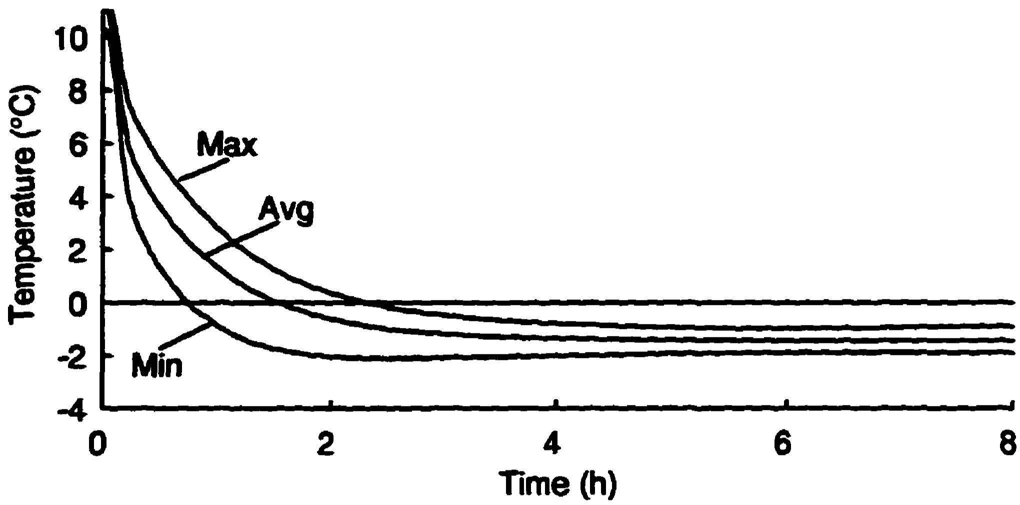

Figure G.21. Minimum, maximum, and average temperatures (Trial 2) of saline water bags when the container was equipped with four ventilation fans, a N_2 tank pressure of 241 kPa, and was exposed to an outside temperature of 15°C.

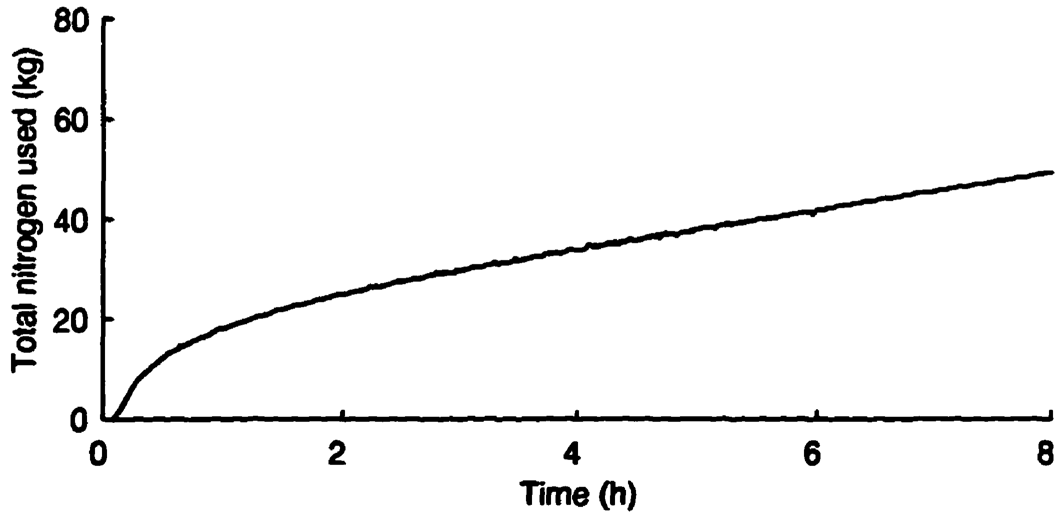

Figure G.22. Nitrogen consumption during Trial 2 (Figure G.21.).

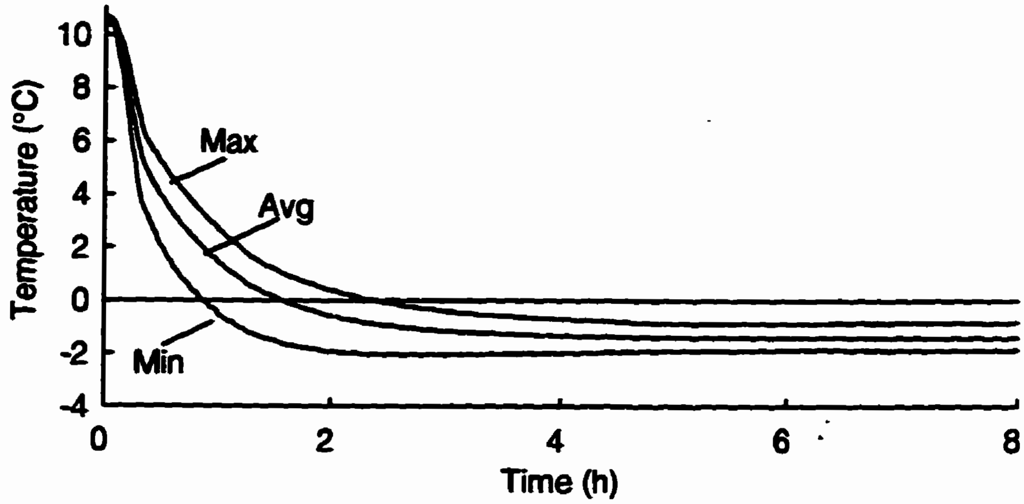

Figure G.23. **Minimum, maximum, and average temperatures (Trial 3) of saline water bags when the container was equipped with four ventilation fans, a N_2 tank pressure of 152 kPa, and was exposed to an outside temperature of 15°C.**

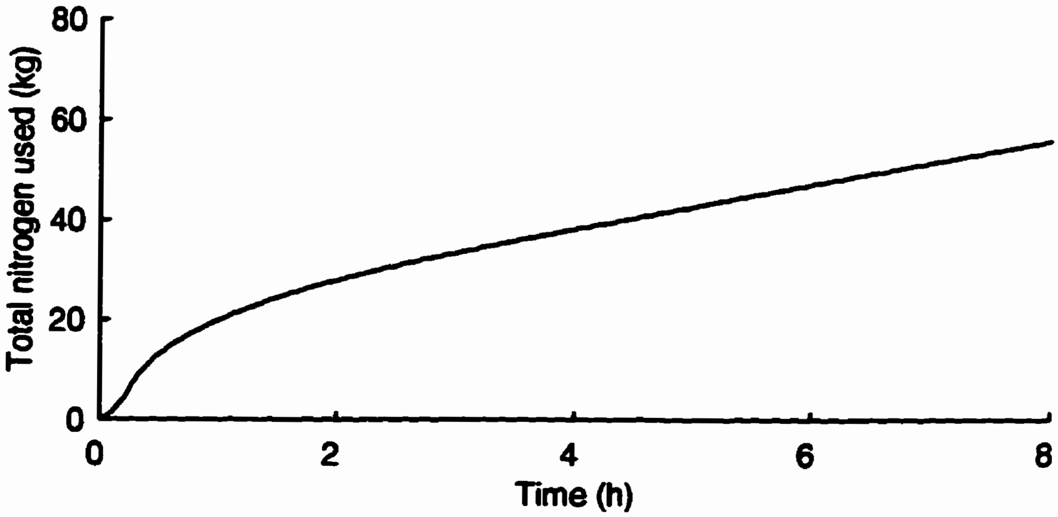

Figure G.24. **Nitrogen consumption during Trial 3 (Figure G.23).**

G.5. Four Fans with an Outside Temperature of 0°C

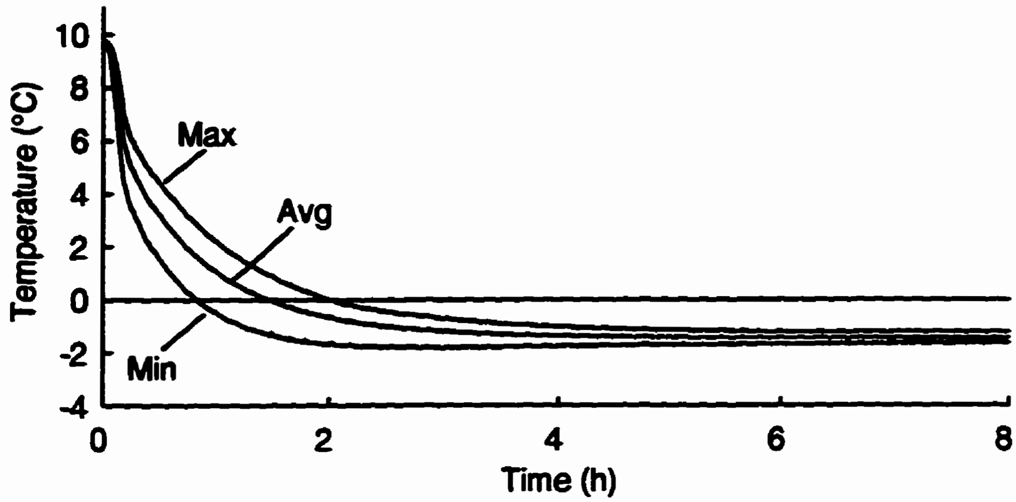

Figure G.25. Minimum, maximum, and average temperatures (Trial 1) of saline water bags when the container was equipped with four ventilation fans, a N_2 tank pressure of 241 kPa, and was exposed to an outside temperature of 0°C.

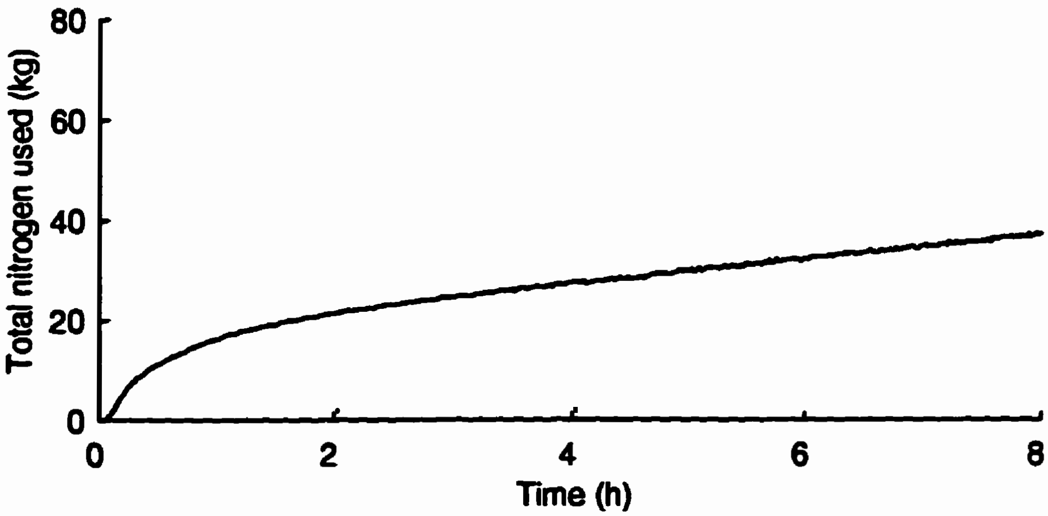

Figure G.26. Nitrogen consumption during Trial 1 (Figure G.25.).

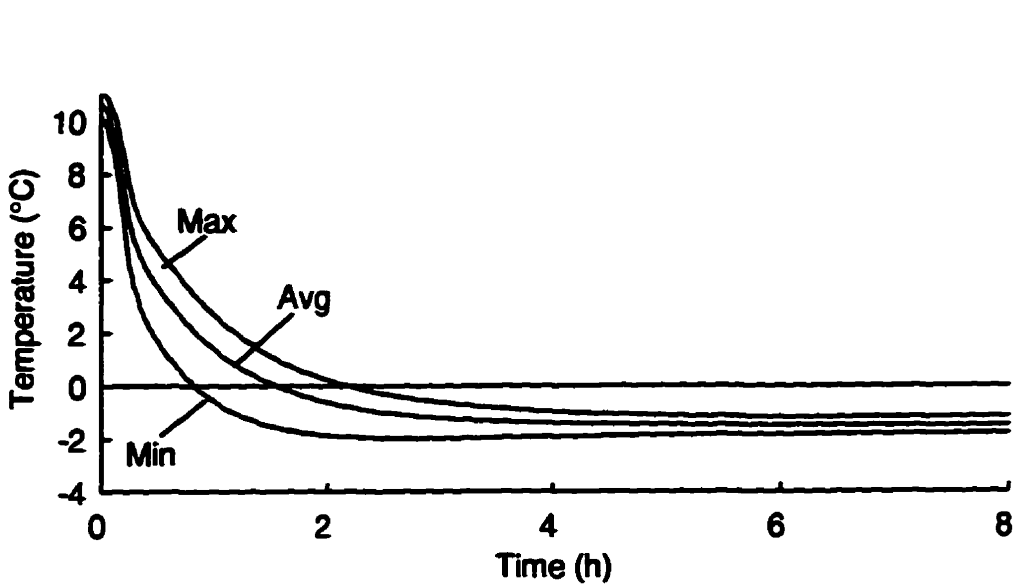

Figure G.27. Minimum, maximum, and average temperatures (Trial 2) of saline water bags when the container was equipped with four ventilation fans, a N_2 tank pressure of 241 kPa, and was exposed to an outside temperature of 0°C.

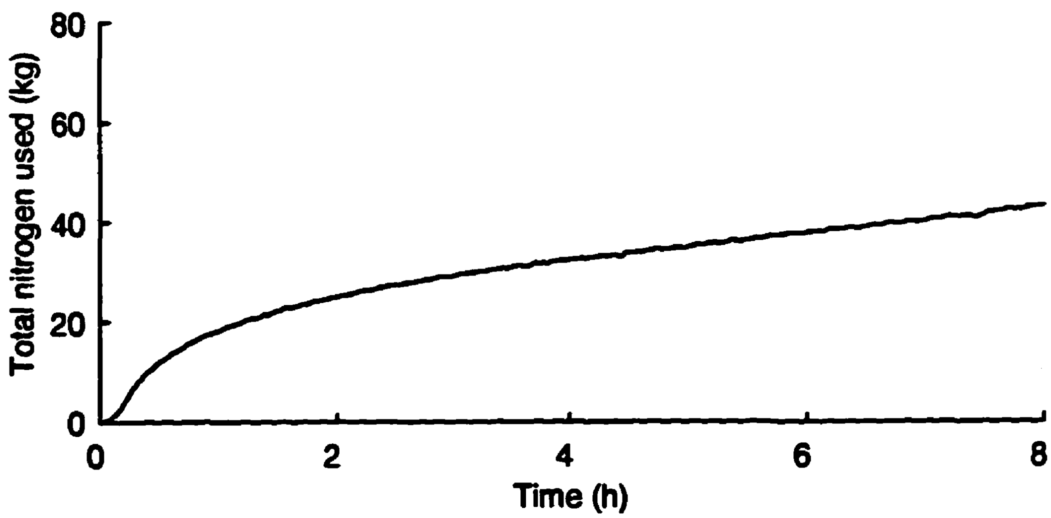

Figure G.28. Nitrogen consumption during Trial 2 (Figure G.27.).

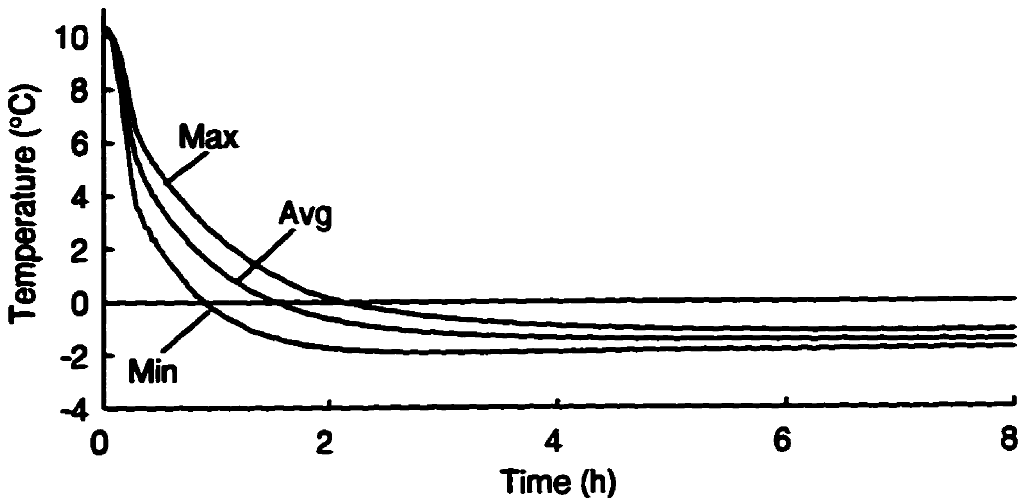

Figure G.29. Minimum, maximum, and average temperatures (Trial 3) of saline water bags when the container was equipped with four ventilation fans, a N_2 tank pressure of 152 kPa, and was exposed to an outside temperature of 0°C.

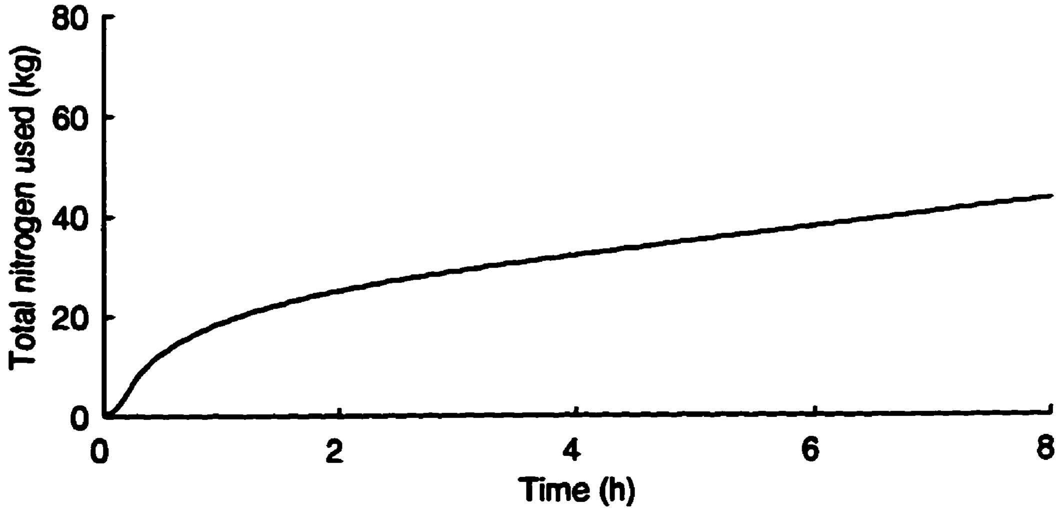

Figure G.30. Nitrogen consumption during Trial 3 (Figure G.29.).

G.6. Four Fans with an Outside Temperature of -15°C

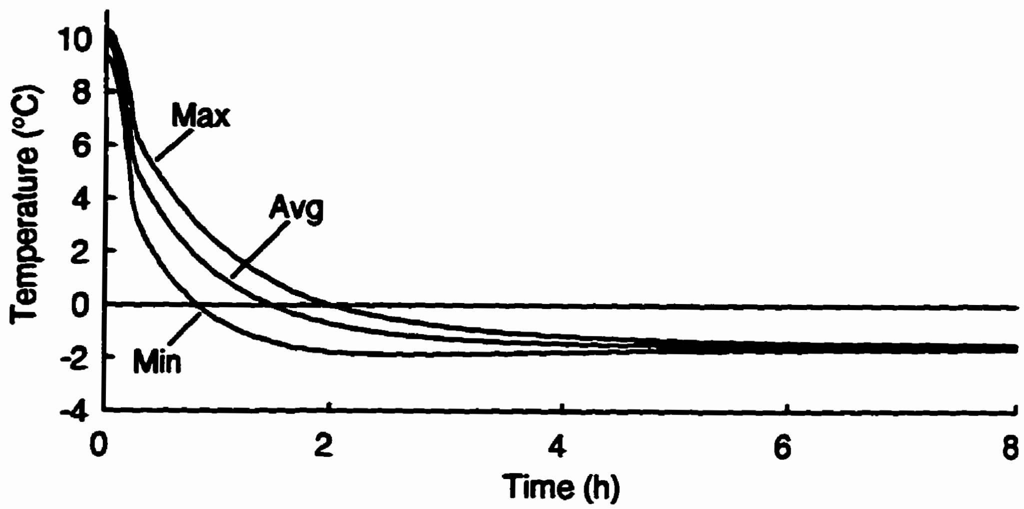

Figure G.31. Minimum, maximum, and average temperatures (Trial 1) of saline water bags when the container was equipped with four ventilation fans, a N_2 tank pressure of 241 kPa, and was exposed to an outside temperature of −15°C.

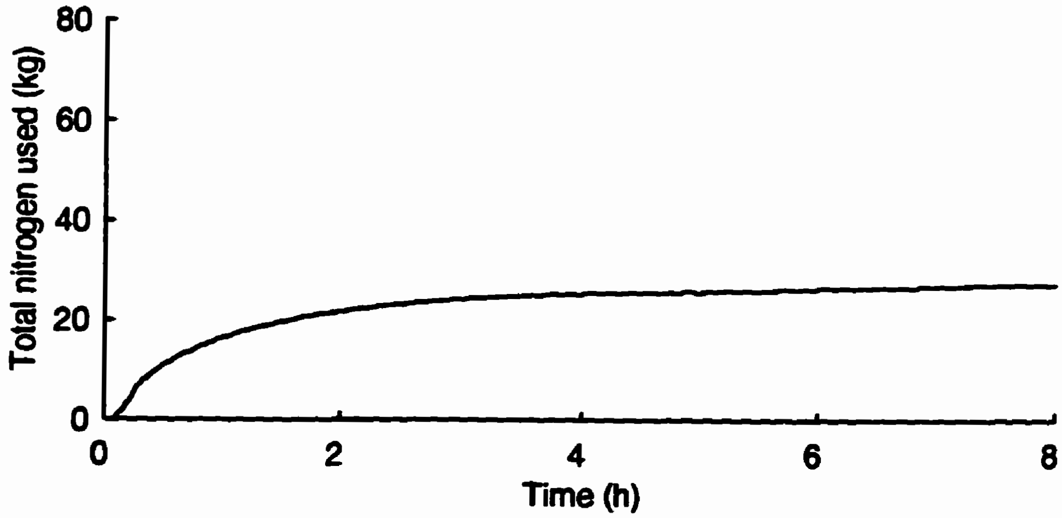

Figure G.32. Nitrogen consumption during Trial 1 (Figure G.31.).

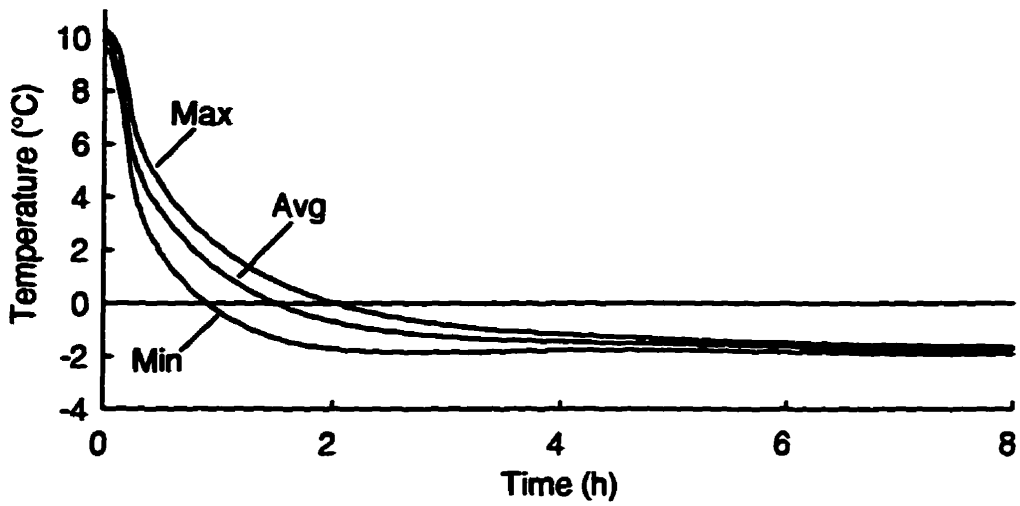

Figure G.33. **Minimum, maximum, and average temperatures (Trial 2) of saline water bags when the container was equipped with four ventilation fans, a N_2 tank pressure of 241 kPa, and was exposed to an outside temperature of −15°C.**

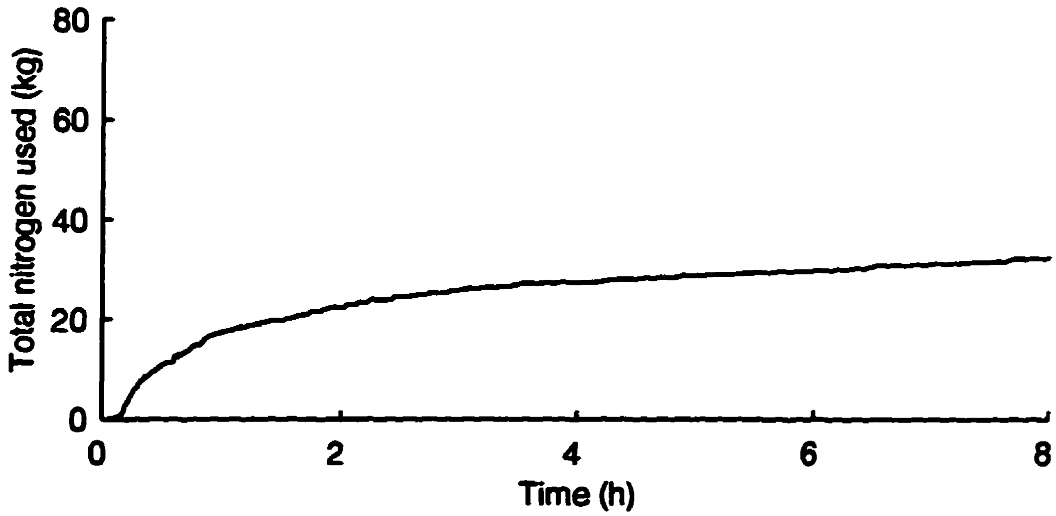

Figure G.34. **Nitrogen consumption during Trial 2 (Figure G.33.).**

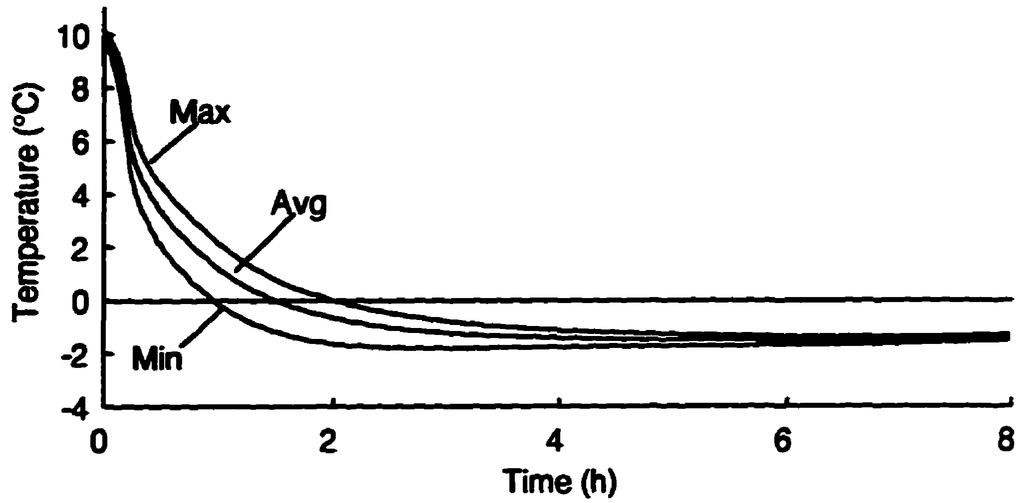

Figure G.35. Minimum, maximum, and average temperatures (Trial 3) of saline water bags when the container was equipped with four ventilation fans, a N_2 tank pressure of 241 kPa, and was exposed to an outside temperature of −15°C.

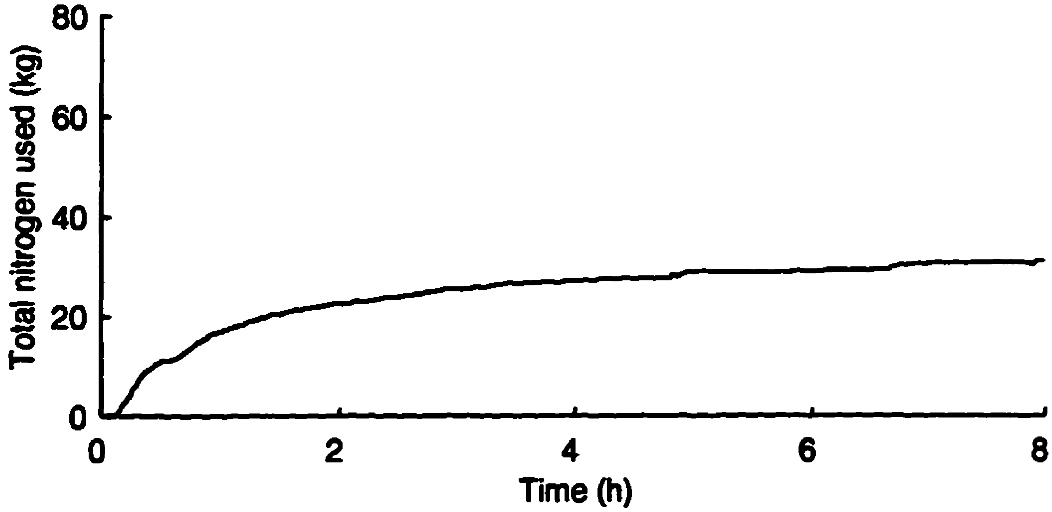

Figure G.36. Nitrogen consumption during Trial 3 (Figure G.35.).

G.7. Nitrogen Failure

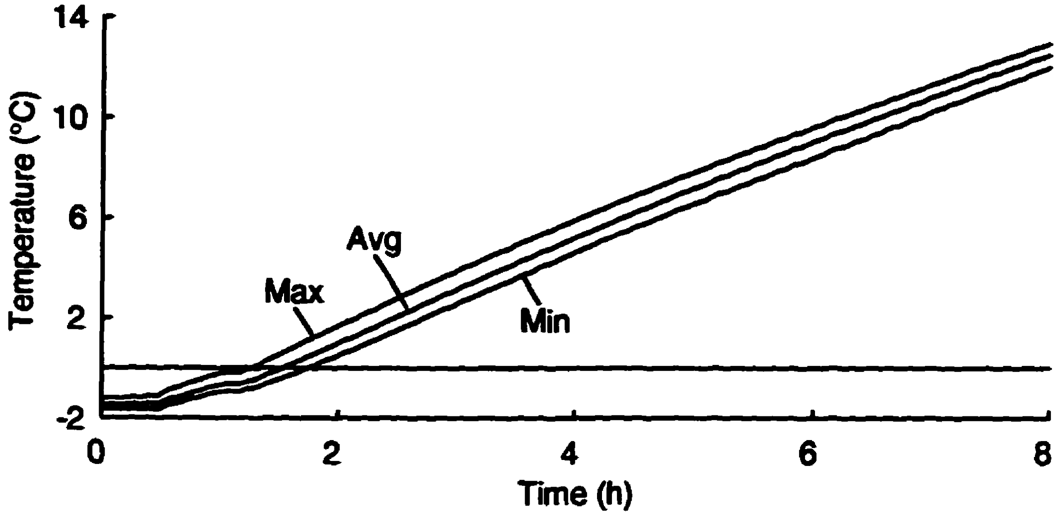

Figure G.37. Minimum, maximum, and average temperatures (Trial 1) of saline water bags when the container, which was equipped with four ventilation fans and exposed to an outside temperature of 30°C, runs out of N_2 at Time = 0.5 h.

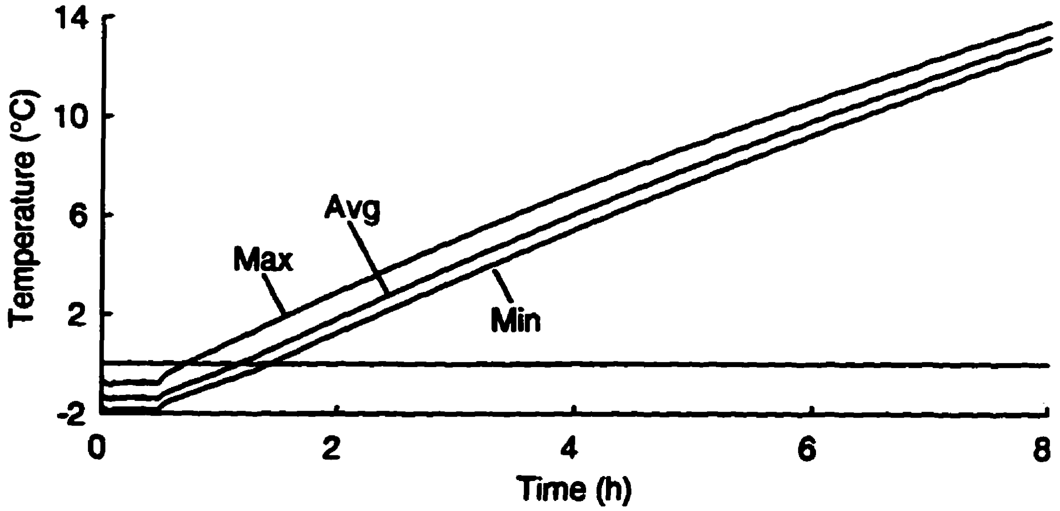

Figure G.38. Minimum, maximum, and average temperatures (Trial 2) of saline water bags when the container, which was equipped with four ventilation fans and exposed to an outside temperature of 30°C, runs out of N_2 at Time = 0.5 h.

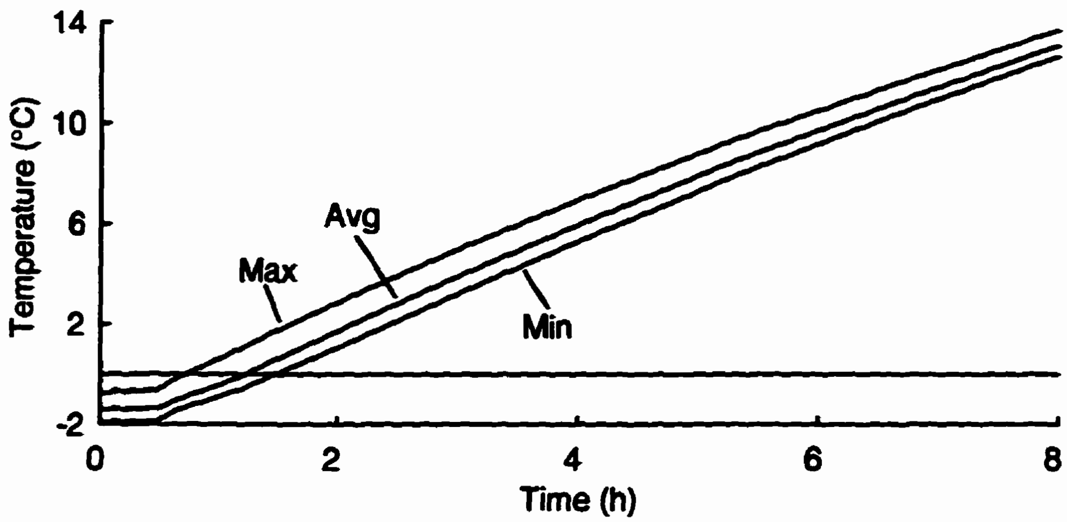

Figure G.39. Minimum, maximum, and average temperatures (Trial 3) of saline water bags when the container, which was equipped with four ventilation fans and exposed to an outside temperature of 30°C, runs out of N_2 at Time = 0.5 h.

G.8. Power Failure

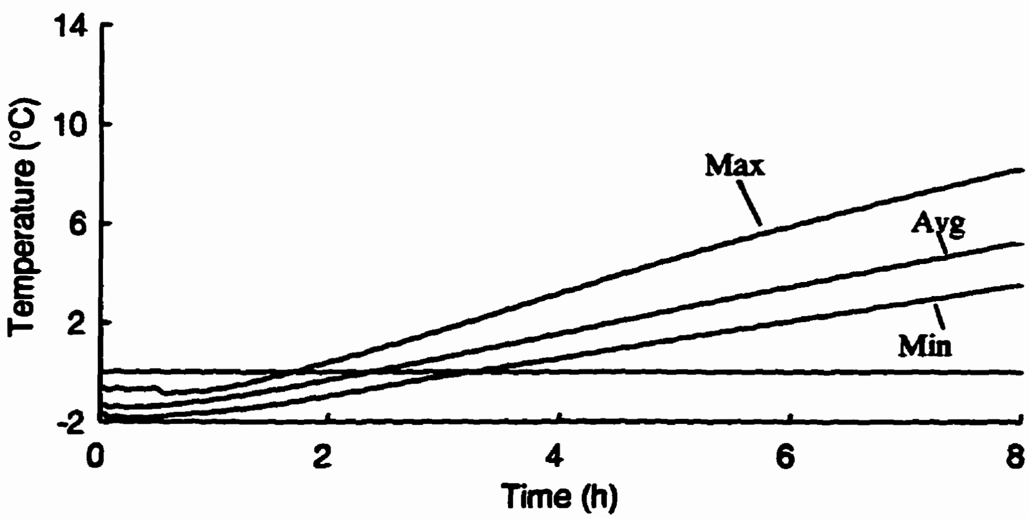

Figure G.40. Minimum, maximum, and average temperatures (Trial 1) of saline water bags when the container, which was exposed to an outside temperature of 30°C, experiences a power outage causing the solenoid valve and fans to cease operation at Time = 0.5 h.

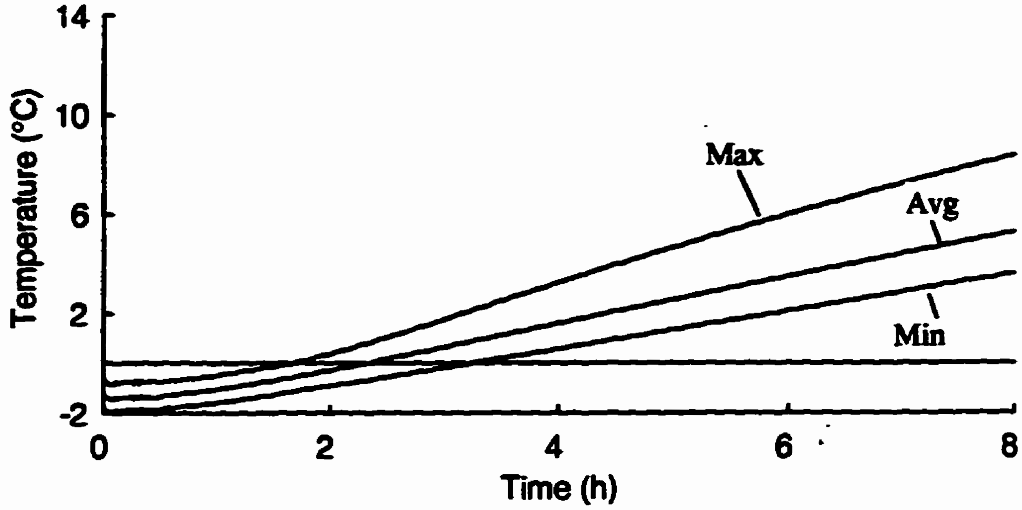

Figure G.41. **Minimum, maximum, and average temperatures (Trial 2) of saline water bags when the container, which was exposed to an outside temperature of 30°C, experiences a power outage causing the solenoid valve and fans to cease operation at Time = 0.5 h.**

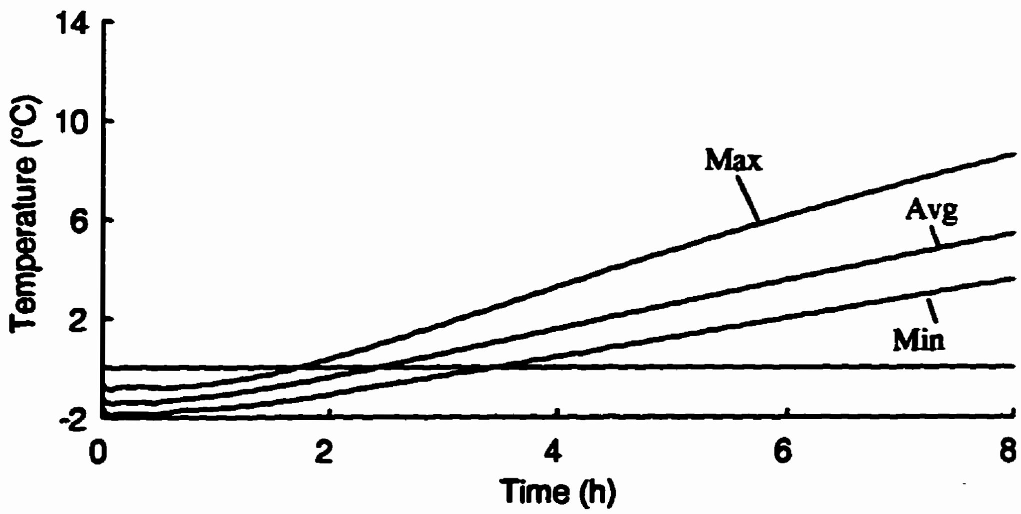

Figure G.42. **Minimum, maximum, and average temperatures (Trial 3) of saline water bags when the container, which was exposed to an outside temperature of 30°C, experiences a power outage causing the solenoid valve and fans to cease operation at Time = 0.5 h.**

G.9. Door Opening

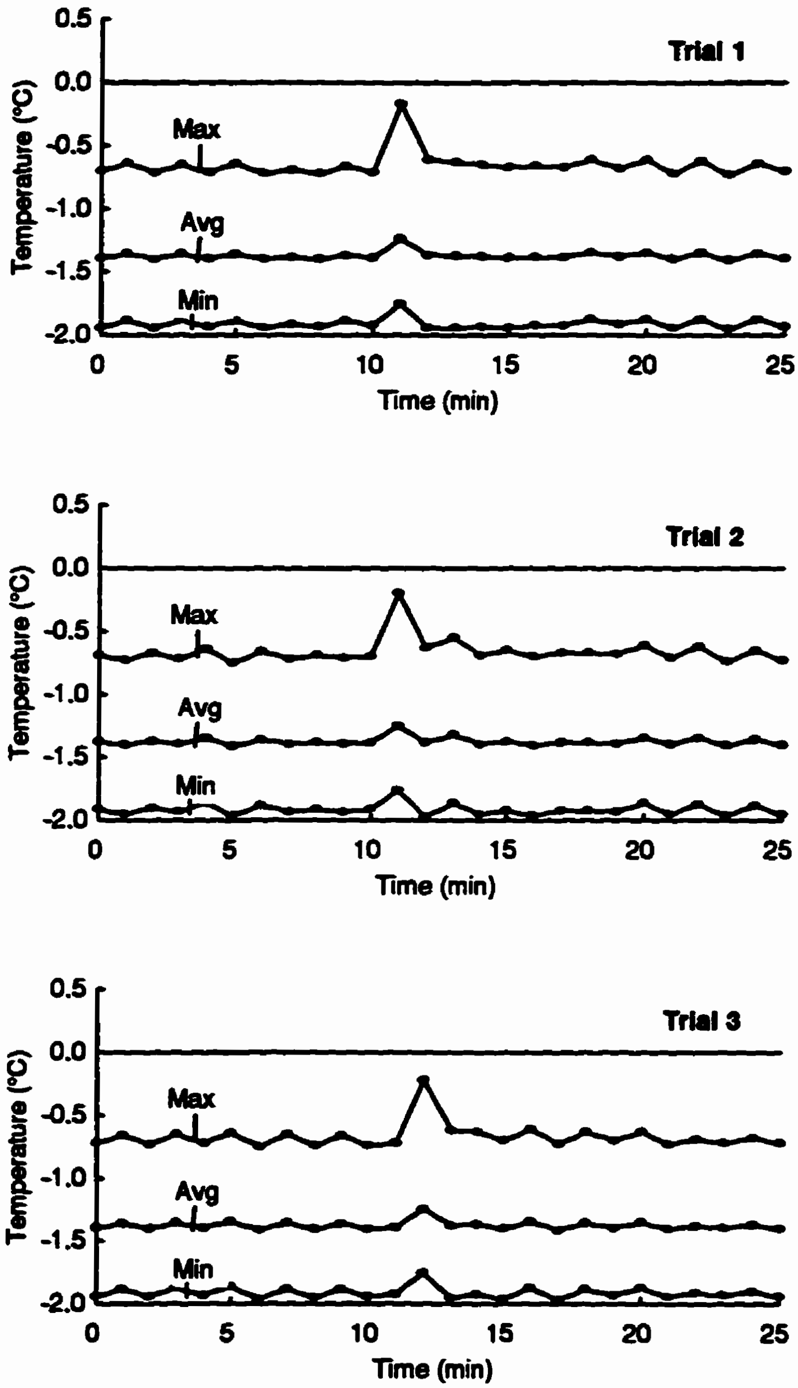

Figure G.43 Minimum, maximum, and average temperatures of saline water bags resulting from the container door being opened for 15 s and exposed to an outside temperature of 21°C at Time = 10 min and 40 s.

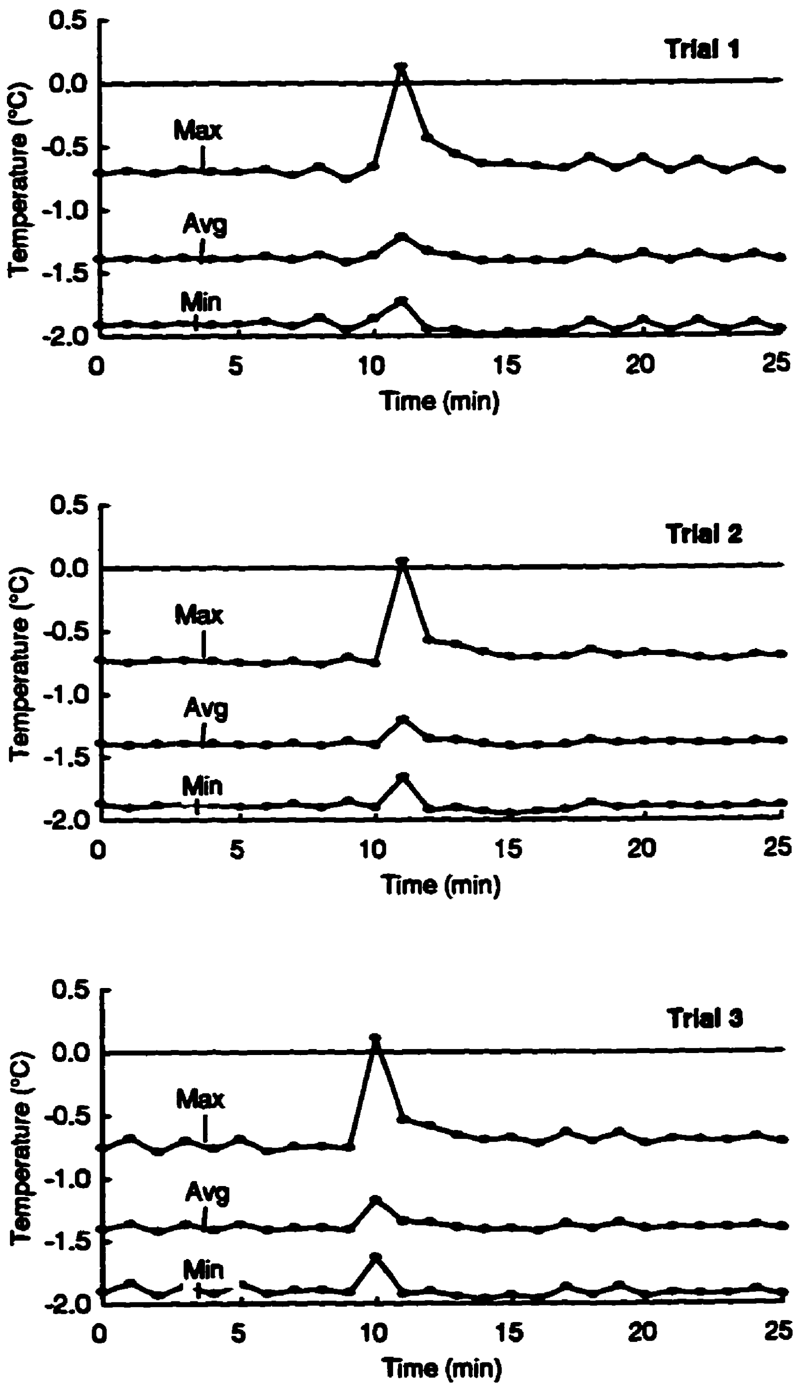

Figure G.44. Minimum, maximum, and average temperatures of saline water bags resulting from the container door being opened for 25 s and exposed to an outside temperature of 21°C at Time = 10 min and 40 s.

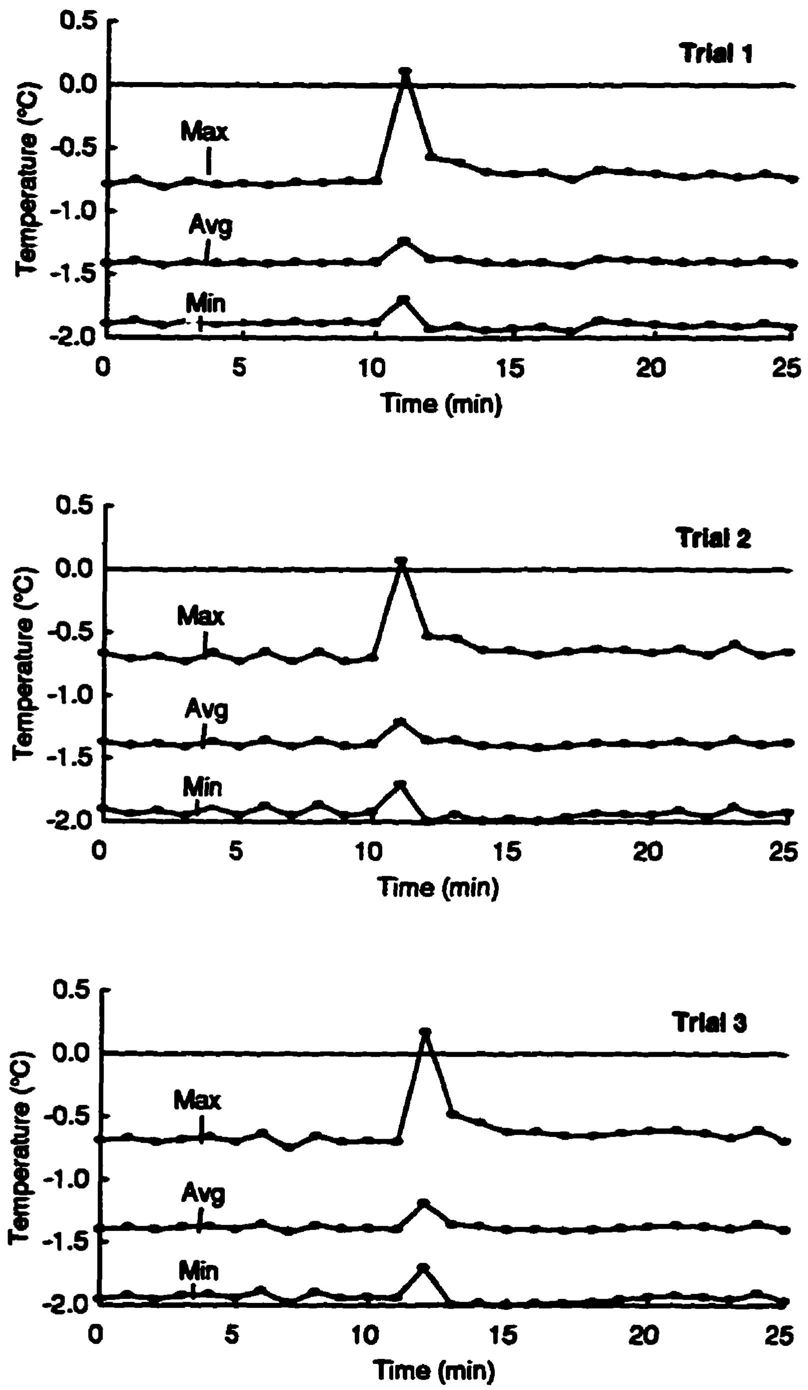

Figure G.45. Minimum, maximum, and average temperatures of saline water bags resulting from the container door being opened for 35 s and exposed to an outside temperature of 21°C at Time = 10 min and 40 s.

G.10. Oxygen Concentrations

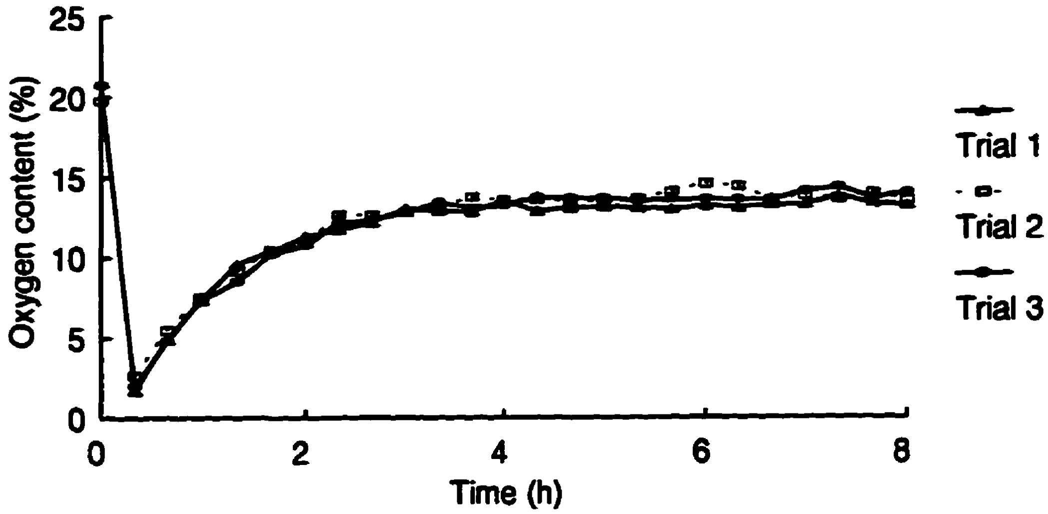

Figure G.46. Oxygen concentrations within the container (which has four circulation fans and was exposed to an outside temperature of 21°C) resulting from cooling the stored product from a temperature of 10°C to −1.5°C and maintaining it at this temperature.

APPENDIX H - NITROGEN USAGE

H.1. Fixed Heat Sources

Nitrogen required to cool the interior of the container (walls, shelving unit, and meat or water bags) was classified as fixed usage. The following equation shows the total contribution to fixed heat:

$$Q_{fixed} = Q_{plastic} + Q_{steel} + Q_{H_2O} \quad \textbf{(H.1)}$$

where:

Q_{fixed}	= total fixed heat removed (J)	$Q_{plastic}$	= fixed heat from plastic (J)
Q_{steel}	= fixed heat from shelving unit (J)	Q_{H2O}	= fixed heat from salt solution (J)

Fixed heat removal was calculated as the product of the mass of the item, its specific heat, and the difference in temperature between its initial and final state:

$$\begin{aligned} Q_{plastic} &= m_{plastic}(Cp_{plastic})\left[T_{initial} - T_{final}\right] \\ Q_{steel} &= m_{steel}(Cp_{steel})\left[T_{initial} - T_{final}\right] \\ Q_{H_2O} &= m_{H_2O}(Cp_{H_2O})\left[T_{initial} - T_{final}\right] \end{aligned} \quad \textbf{(H.2)}$$

where:

m_{steel}	= mass of shelving unit (kg)	m_{H20}	= mass of saline solution (kg)
$m_{plastic}$	= mass of polyethylene (kg)	Cp_{steel}	= specific heat of steel (J/(kg• K))
Cp_{H20}	= specific heat of water (J/(kg•K))	$Cp_{plastic}$	= specific heat of plastic (J/(kg• K))
$T_{initial}$	= initial temperature (K)	T_{final}	= final temperature (K)

The approximate masses of the polyethylene walls and air distribution duct, steel shelving unit, and the saline water bags within the container were 70, 60, and 108 kg, respectively. Their corresponding specific heats are 2300, 475, and 3720 J/(kg•K)

(ASHRAE 1993; Incropera and De Witt 1990). Substituting these values into Equation H.2 yields:

$$\begin{aligned} Q_{plastic} &= 70(2300)\left[T_{initial} - T_{final}\right] \\ Q_{steel} &= 60(475)\left[T_{initial} - T_{final}\right] \\ Q_{H_2O} &= 108(3720)\left[T_{initial} - T_{final}\right] \end{aligned} \qquad \textbf{(H.3)}$$

H.2. Variable Heat Sources

Nitrogen required on a continual basis during refrigeration was referred to as variable usage of N_2. Variable heat sources include heat produced by the circulation fans, heat conducted through the walls of the container, and heat conducted into the transfer line.

$$\dot{q}_{variable} = \dot{q}_{fans} + \dot{q}_{container} + \dot{q}_{transfer} \qquad \textbf{(H.4)}$$

where:

$\dot{q}_{variable}$ = variable heat (W)
$\dot{q}_{fans}$ = heat produced by fans (W)
$\dot{q}_{container}$ = conduction into container (W)
$\dot{q}_{transfer}$ = conduction into transfer line (W)

Heat production of the fans was measured and found to be approximately 30.8 W/fan (Section 5.1.1.). Therefore, the following equation can be derived:

$$\dot{q}_{fans} = Number\ of\ Fans \times 30.8 \qquad \textbf{(H.5)}$$

The heat flow into the container was determined from the following equation:

$$\dot{q}_{cabinet} = \frac{T_{outside} - T_{inside}}{R_{container}} \quad \textbf{(H.6)}$$

where:

$\dot{q}_{container}$ = conduction into container (W) $R_{container}$ = thermal container resistance (K/W)
$T_{outside}$ = temperature outside container (K) T_{inside} = temperature inside container (K)

The thermal resistance of the container was measured experimentally (Section 5.1.1.) and found to be 137×10^{-3} K/W. Substituting this value into Equation H.6 gives:

$$\dot{q}_{container} = \frac{T_{outside} - T_{inside}}{137 \times 10^{-3}} \quad \textbf{(H.7)}$$

Transfer lines are typically insulated with pipe insulation in a cylindrical fashion and heat flow equations have been derived to solve these types of problems. However, the transfer line designed in this thesis was insulated with a 200 mm square block of insulation. In order to analyse the heat flow into the transfer line and to use existing equations, it was necessary to approximate the square block of insulation as a cylinder with inscribed diameter equal to 200 mm. The following equation was used (Incropera and De Witt 1990):

$$\dot{q}_{transfer} = \frac{2\pi Lk(T_{outside} - T_{line})}{\ln\left[\frac{r_{outer}}{r_{inner}}\right]} \quad \textbf{(H.8)}$$

where:

$\dot{q}_{transfer}$ = conduction into transfer line (W)
r_{outer} = outer insulation radius (m)
r_{inner} = inner insulation radius (m)
L = length of transfer line (m)
$T_{outside}$ = temperature outside insulation (K)
T_{line} = temperature inside pipe (K)
k = thermal conductivity (W/(m•K))

The length of the copper transfer line is 1.68 m and its outer radius is 6 mm. Extruded polystyrene was used as insulation. Its outer radius is 100 mm and its thermal conductivity is 0.027 W/(m•K) (Incropera and De Witt 1990). Substituting these values into Equation H.7 yields the following:

$$\dot{q}_{transfer} = \frac{2(3.14)(1.68)(0.027)(T_{outside} - T_{line})}{\ln\left[\frac{0.1}{0.006}\right]} \qquad \text{(H.9)}$$

$$= 101 \times 10^{-3}\ (T_{outside} - T_{line})$$

H.3. Usage of Nitrogen

To calculate the fixed usage of N_2 the following formula was used:

$$m_{N_2} = \frac{Q_{fixed}}{\Delta h_{fluid-gas}} \qquad \text{(H.10)}$$

where:

m_{N2} = fixed mass of N_2 used (kg)
$\Delta h_{fluid-gas}$ = enthalpy change of N_2 (kJ/kg)
Q_{fixed} = amount of fixed heat removed (kJ)

Variable usage of N_2 is similarly calculated:

$$\dot{m}_{N_2} = \frac{\dot{q}_{variable}}{\Delta h_{fluid-gas}} \qquad \text{(H.11)}$$

where:

$\dot{m}_{N2}$ = variable mass of N_2 used (kg/s) $\dot{q}_{variable}$ = variable heat removed (W)
$\Delta h_{fluid-gas}$ = enthalpy change of N_2 (J/kg)

Liquid N_2 was stored at 152 kPa or 241 kPa, resulting in $\Delta h_{fluid-gas}$ equalling 385 and 378 kJ/kg, respectively (Appendix B).

Table H.1. Predicted sources of fixed heat and the N_2 required to remove it based on experimental observations of initial and final container temperatures.

Number of fans	Trial number	$T_{initial}$ (°C)	T_{final} (°C)	Fixed heat removal (kJ) $Q_{plastic}$ [a]	Q_{shelf} [a]	Q_{H2O} [a]	Q_{fixed} [b]	N_2 use [c] (kg)
				Temperature outside the container = 30°C				
2	1	9.9	-1.4	1820	320	4540	6680	17.7
	2	9.7	-1.5	1800	320	4500	6620	17.5
	3[d]	9.8	-1.5	1820	320	4540	6680	17.4
4	1	10.3	-1.4	1880	330	4700	6910	18.3
	2	9.3	-1.4	1720	300	4300	6320	16.7
	3	10.2	-1.4	1870	330	4660	6860	18.1
6	1	10.4	-1.4	1900	340	4740	6980	18.5
	2[d]	10.5	-1.4	1920	340	4780	7040	18.3
	3	11.2	-1.4	2030	360	5060	7450	19.7
				Temperature outside the container = 15°C				
4	1	9.9	-1.4	1820	320	4540	6680	17.7
	2	11.1	-1.4	2010	360	5020	7390	19.6
	3[d]	10.6	-1.4	1930	340	4820	7090	18.4
				Temperature outside the container = 0°C				
4	1	9.7	-1.5	1800	320	4500	6620	17.5
	2	10.6	-1.5	1950	340	4860	7150	18.9
	3[d]	10.2	-1.5	1880	330	4700	6910	17.9
				Temperature outside the container = -15°C				
4	1	9.9	-1.5	1840	320	4580	6740	17.8
	2	9.9	-1.7	1870	330	4660	6860	18.1
	3	9.8	-1.5	1820	320	4540	6680	17.7

[a] calculated from Equation H.3.

[b] calculated from Equation H.1.

[c] calculated from Equation H.10.

[d] liquid N_2 storage tank at 152 kPa instead of the usual 241 kPa.

Table H.2. Predicted sources of variable heat and the N_2 required to remove it based on experimental observations of inner and outer container temperatures and the number of fans operating.

Number of fans	Trial number	$T_{outside}$ (°C)	T_{inside} (°C)	Variable heat removal (W) $\dot{q}_{fans}$ [a]	$\dot{q}_{container}$ [b]	$\dot{q}_{transfer}$ [c]	$\dot{q}_{variable}$ [d]	N_2 use [e] (kg/h)
				Temperature outside the container = 30°C				
2	1	30.0	-1.4	61.6	229	21.7	313	3.0
	2	29.9	-1.5	61.6	229	21.7	313	3.0
	3[f]	29.8	-1.5	61.6	228	21.7	312	2.9
4	1	30.1	-1.4	123	330	21.7	375	3.6
	2	29.9	-1.4	123	228	21.7	373	3.6
	3	29.9	-1.4	123	228	21.7	373	3.6
6	1	30.0	-1.4	185	229	21.7	436	4.1
	2[f]	30.0	-1.4	185	229	21.7	436	4.1
	3	30.0	-1.4	185	229	21.7	436	4.1
				Temperature outside the container = 15°C				
4	1	14.9	-1.4	123	119	20.2	262	2.5
	2	15.0	-1.4	123	120	20.2	263	2.5
	3[f]	15.1	-1.4	123	120	20.2	264	2.5
				Temperature outside the container = 0°C				
4	1	-0.3	-1.5	123	8.8	18.7	151	1.4
	2	-0.8	-1.5	123	5.1	18.6	147	1.4
	3[f]	-0.6	-1.5	123	6.6	18.6	148	1.4
				Temperature outside the container = -15°C				
4	1	-15.8	-1.5	123	-104	17.1	35.9	0.09
	2	-15.4	-1.7	123	-100	17.1	40.3	0.11
	3	-15.3	-1.5	123	-101	17.1	39.6	0.10

[a] calculated from Equation H.5.
[b] calculated from Equation H.7.
[c] calculated from Equation H.9.
[d] calculated from Equation H.4.
[e] calculated from Equation H.11.
[f] liquid N_2 storage tank at 152 kPa instead of the usual 241 kPa.

APPENDIX I - AIR VELOCITY DISTRIBUTIONS

The air duct consisted of an array of holes through which air was distributed at different shelf levels throughout the container. The air velocity was measured at each of these 16 mm diameter holes with a hot wire anemometer (Model TA3000, AirFlow Developments Canada Ltd., Georgetown, ON). The frontal area of the container was covered in cardboard while measurements were taken to simulate the effect of the door being closed. Small holes were cut in the card board to facilitate movement of the hot wire anemometer to each hole in the array.

Measurements were recorded as an array in Tables I.1. through I.4. A letter-number combination refers to the row and column of each hole. The letters "a" and "j" refer to the top and bottom rows of holes, respectively. As well, the numbers "1" and "10" correspond to holes located at the front and back of the container, respectively. The velocities were also recorded for both the left and right side of the air distribution conduit.

Table I.1. Array of air velocities (m/s), which were measured at the holes to the distribution duct, resulting from four circulation fans.

Array rows[a]	Array columns (left hand side)[a]										Array columns (right hand side)[a]									
	1	2	3	4	5	6	7	8	9	10	1	2	3	4	5	6	7	8	9	10
a	11	5	5	5	4	6	4	5	4	4	7	8	9	7	10	7	8	7	7	10
b	2	6	5	5	5	4	4	3	4	4	6	7	6	7	6	6	5	4	5	3
c	5	5	4	4	5	5	6	4	4	6	7	7	7	7	6	6	7	7	7	8
d	8	7	8	7	7	8	8	8	8	8	8	8	8	7	8	7	7	7	7	8
e	8	8	8	8	8	8	7	8	8	7	8	8	8	8	8	8	7	7	7	8
f	8	8	8	8	8	8	7	7	8	7	9	8	8	8	8	8	7	7	7	7
g	8	8	8	8	8	7	7	7	8	7	8	8	8	8	8	7	7	8	7	7
h	8	8	8	8	8	7	7	7	7	7	8	8	8	8	8	7	7	7	7	7
i	8	8	8	8	8	7	7	8	7	7	8	8	8	7	8	7	7	7	7	7
j	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	7	7	7	7	7

[a] rows are lettered from the top of the conduit to the bottom (with "a" being the top row of holes), whereas columns are numbered from the back to the front (with "1" being the back column of holes). See Appendix A for exact hole locations.

Table I.2. Array of air velocities (m/s), which were measured at the holes to the distribution duct, resulting from four circulation fans. [a]

Array rows[b]	Array columns (left hand side)[b]										Array columns (right hand side)[b]									
	1	2	3	4	5	6	7	8	9	10	1	2	3	4	5	6	7	8	9	10
a	11	5	6	5	4	8	5	7	9	3	7	8	9	7	10	7	7	7	6	9
b	2	5	3	5	6	2	6	5	5	5	5	5	5	6	5	5	5	6	6	3
c	5	7	6	7	7	7	7	7	7	7	6	6	6	8	7	7	7	7	7	7
d	6	6	8	6	6	8	7	7	8	8	7	7	7	7	8	7	7	8	7	8
e	9	8	8	8	8	8	7	8	8	8	8	7	8	7	7	7	7	8	7	8
f	8	8	8	8	8	8	8	7	8	8	8	7	8	7	8	7	7	7	7	7
g	8	8	8	8	8	8	8	7	8	8	7	7	8	7	7	7	7	7	7	7
h	8	8	7	8	8	8	7	8	8	8	7	7	7	7	7	7	7	7	7	7
i	8	8	8	8	8	7	7	8	8	8	8	8	8	7	8	8	7	8	8	8
j	8	8	8	8	8	8	8	8	8	8	8	8	8	8	9	8	8	8	8	8

[a] the positions of the four fans were shuffled from the test conducted in Table I.1.
[b] rows are lettered from the top of the conduit to the bottom (with "a" being the top row of holes), whereas columns are numbered from the back to the front (with "1" being the back column of holes). See Appendix A for exact hole locations.

Table I.3. Array of air velocities (m/s), which were measured at the holes to the distribution duct, resulting from six circulation fans.

Array rows[a]	Array columns (left hand side)[a]										Array columns (right hand side)[a]									
	1	2	3	4	5	6	7	8	9	10	1	2	3	4	5	6	7	8	9	10
a	7	5	3	5	3	11	7	3	6	5	10	8	5	4	11	5	4	3	6	12
b	4	7	6	4	4	7	5	5	5	7	7	5	7	8	4	5	6	3	5	3
c	6	7	7	7	7	7	7	8	8	8	8	6	7	8	8	7	8	7	5	5
d	9	9	8	8	8	9	9	9	9	9	7	8	8	8	8	8	9	8	8	9
e	9	9	8	8	8	8	8	8	9	9	8	8	8	8	8	9	8	8	9	9
f	9	8	8	8	9	9	9	8	9	9	9	8	8	8	8	8	8	8	8	8
g	9	8	8	8	9	8	8	8	9	9	8	8	8	8	8	8	8	8	9	8
h	8	8	8	8	8	9	9	8	9	9	8	8	8	8	8	8	8	8	8	9
i	8	8	8	8	8	9	9	9	9	9	8	8	8	8	8	8	8	8	8	8
j	8	8	8	8	8	8	8	8	9	9	8	8	8	8	8	8	8	8	8	8

[a] rows are lettered from the top of the conduit to the bottom (with "a" being the top row of holes), whereas columns are numbered from the back to the front (with "1" being the back column of holes). See Appendix A for exact hole locations.

Table I.4. Array of air velocities (m/s), which were measured at the holes to the distribution duct, resulting from two circulation fans.

Array rows[a]	Array columns (left hand side)[a]										Array columns (right hand side)[a]									
	1	2	3	4	5	6	7	8	9	10	1	2	3	4	5	6	7	8	9	10
a	6	5	2	2	6	7	5	1	4	7	7	5	1	5	7	7	2	1	6	8
b	7	1	4	4	7	8	6	6	6	8	7	3	5	2	7	7	3	5	5	7
c	7	7	6	6	7	8	6	7	7	8	6	4	5	5	4	6	6	6	5	7
d	6	4	5	5	5	7	7	6	7	7	7	6	5	5	4	7	6	6	7	7
e	7	6	5	6	6	8	7	6	7	8	7	6	5	6	5	7	7	5	6	7
f	7	6	6	6	7	8	7	7	7	7	7	6	6	6	6	7	7	6	7	7
g	6	6	6	6	7	8	7	7	7	7	7	6	6	6	7	7	7	7	7	7
h	7	6	6	6	7	7	7	7	7	7	7	6	6	6	6	8	7	7	7	7
i	7	7	7	7	7	8	7	7	8	7	7	7	6	6	6	8	8	8	8	8
j	7	6	7	7	7	7	8	8	8	8	7	7	7	7	6	7	7	7	7	6

[a] rows are lettered from the top of the conduit to the bottom (with "a" being the top row of holes), whereas columns are numbered from the back to the front (with "1" being the back column of holes). See Appendix A for exact hole locations.