THE ANALYSIS OF AN EXPERIMENTAL SYSTEM TO TEMPER VENTILATION AIR USING SOIL HEAT

by

Tracy Vivian Murray, P.Eng.

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MASTER OF SCIENCE

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ABSTRACT

A full-scale field model, earth-to-air heat exchanger at the University of Manitoba Research Farm at Glenlea, Manitoba was monitored to evaluate its performance under Manitoba conditions. The heat exchanger consisted of four, 30 m long PVC pipes buried at an average depth of 3.0 m below grade in Red River Gumbo Clay, (two 250 mm diameter pipes with volumetric flow rates of 0.05 m^3/s and 0.10 m^3/s and two 150 mm diameter pipes with volumetric flow rates of 0.15 m^3/s and 0.05 m^3/s). A drainage problem with the 150 mm, 0.15 m^3/s pipe eliminated it from consideration.

Ambient air temperatures, outlet air temperatures, air temperatures along each pipe, soil temperatures up to 1.0 m from the surface of each pipe, and background soil temperatures were sensed for continous operation over three years.

All three pipes demonstrated the ability to temper the extreme temperatures experienced during Manitoban winters and summers. Temperature changes between inlet and outlet air as high as 29°C under winter conditions and 18°C under summer conditions were observed. Little tempering occurred when ambient air was in the range of -10°C to 15°C. The system was found to regenerate from one season to the next. Classical heat transfer methods adequately described the heat transfer between the soil and the air.

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TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

FIGURE page

Chapter I

INTRODUCTION

Agricultural structures are necessary in order to provide a modified environment for animal shelters, plant production facilities and food storage. A controlled thermal environment in livestock shelters can increase production, improve feed efficiency, and reduce mortality (Scott et. al. (1965)). Weisbecker and Jacobson (1980) predict that an increase of one pig per litter is possible in farrowing houses where thermal stress has been eliminated. In western Canada, where extreme temperatures occur in both summer and winter, supplementary energy is often required to achieve and maintain an optimum environment; yet this is normally not economically feasible using conventional methods.

The high cost/benefit ratio in livestock and plant structures has limited the use of refrigeration systems to produce storage (Scott, et al., 1965). Scott, Deshazer and Roller reported in Hellikson and Walker (1983), "... except for the very young (new-born) animal where some environmental modification is necessary, the provision of a thermally controlled environment for adult animals is not always economically justified." Diener et al. (1986) suggested that nearly one-quarter of the total energy in livestock production is used for space heating and cooling and to replace ventilation losses. Britton (1981) reported that in the western Canadian prairies most controlled environment livestock barns would function without the need for supplemental heat until ambient air temperatures reach the -20°C range. Below this temperature, warming the incoming ventilation air would account for 75% to 85% of the total heat loss. He further stated that although economic loss is realized when animals are heat stressed, costly conventional cooling systems can not be justified. Therefore, the need to provide an economical means of achieving temperature and humidity control in these structures is obvious.

One way to achieve this is to temper ambient air by drawing it through buried pipes as in an earth-to-air heat exchanger. The air will gain heat from the soil during the winter and loose heat to the soil during the summer, thus minimizing the extremes of incoming air temperature throughout the year. Ventilation air will then require only a minimal amount of heating/cooling energy. As the temperature of the earth 3.0 m below grade varies only slightly during the year, using the earth at this depth as a heat source/sink provides a promising solution.

Performance of such a system will depend mainly on the temperature difference between the soil surrounding the pipe and the ambient air. Other factors affecting efficiency might include soil characteristics such as soil composition, soil moisture content, ground water level; and design parameters such as pipe thermal resistance, pipe length, pipe diameter, pipe spacing, depth of burial, and airflow rate. Therefore the design of an earth-to-air heat exchanger becomes a very complex task which must be studied carefully. To date, research has been carried out in the mid-western United States, Canada, Japan and Europe but of the limited data available, most is too site specific to be directly applicable to Manitoba conditions.

Therefore the objectives of this thesis are:

1. to evaluate the performance characteristics of a buried pipe ventilation air tempering system at Glenlea, Manitoba, and
2. to determine which factors among pipe diameter, pipe length, and airflow rate, have the greatest effect on system performance.

Chapter II

LITERATURE REVIEW

2.1 EXPERIMENTAL INSTALLATIONS

In a brief summary of the development of modern day earth-to-air heat exchangers, Scott et al. (1965) recorded several examples where such systems have been used in the past. The first case, dated from 1878, was an Iowa barn which was air conditioned during the summer months with a 152 m (500 ft) underground passage serving as a heat exchanger duct. Then, during a hot summer around 1940, a New Jersey poultry house was cooled by drawing ambient air through the collecting main of an existing herring bone tile drain. In 1962 Mr. Paul Sturger of Sturger Heat Recovery Inc., connected 91.4 m (300 ft) of 600 mm (24 in.) pipe to his home heat pump system. Finally, in 1964, a heat exchanger consisting of 27 m (90 ft) of 200 mm (8 in.) stovepipe was used to condition air for a small broiler house in Maryland. Although each of these systems had proven their ability to temper the incoming air, little or no reliable data were available on which future designs could be based.

As a result of this lack of data Scott et al. (1965) initiated a project at Cornell University to supply engineering data on earth-to-air heat exchangers. They monitored the heat transfer through the soil near a buried pipe to determine the effect of intermittent and continuous operation and they studied the economic feasibility of using these systems for conditioning ventilation air for agricultural shelters. The paper presented a simplified steady state approximation of earth-air heat exchange as well as limited field data based on 134 m (440 ft) of 450 mm (18 in.) diameter Armco HEL-COR pipe buried at a depth of 2.4 m (8 ft) below grade in clay soil. Thermocouples were used to measure soil and air temperatures at 30.5 m (100 ft) intervals along the pipe. At each measuring station soil temperatures were measured to 1.5 m from the pipe surface in the 3, 6, 9, and 12 o'clock positions and air temperatures were measured at the center of the pipe. Undisturbed soil temperatures were also monitored. Airflow rates ranging from 533 L/s to 1147 L/s (1130 cfm to 2430 cfm) were studied.

The results presented were based on data collected between March 1964 and December 1965. The conclusion drawn in 1965 was that favorable heat transfer was achieved under both continuous and intermittent operation. An average temperature difference between inlet and outlet air of 15°C (27°F) was possible under summer conditions with continuous operation, 17°C (30°F) with intermittent operation. A steady state approximation yielded conservative, but reasonably accurate, predictions of outlet air temperatures when the thermal properties of the soil were known. Despite the proven technical feasibility of this system, it was felt to be of doubtful economic value for use in animal shelters given 1965 energy prices.

Research in this area slowed until the late 1970's when fuel costs increased and concern grew over the future of our energy supply. In 1977, Walker and Buxton analysed the possibility of using an earth-to-air heat exchanger to heat and cool a small greenhouse. They studied a theoretical model based upon 30.5 m of 450 mm diameter pipe buried 3.0 m below grade. The results of their analysis showed adequate heating could be achieved only in the very southern parts of the United States if a greenhouse temperature of 10°C was acceptable. At the same time, they concluded that the system, when used for cooling, could effectively limit the occurrence of high temperatures in greenhouses throughout the United States.

In 1981, Francis reported the results of three experimental installations for summer space conditioning. Only two of the systems will be discussed here since the third system, consisting of two concrete tanks filled with dolomite and installed 2.0 m below grade, is not relevant to this thesis. The first system consisted of two, 120 m long plastic field tile tubes, 200 mm in diameter, buried to an average depth of 3.0 m below grade. A velocity of 3.48 m/s was chosen to give an airflow rate of 0.1128 m^3/s. He found that 35°C ambient air could be cooled to 18.3°C in 120 m of pipe. Most of the cooling was found to occur in the first 60 m of pipe. The extreme length and relatively small diameter of the tubes dictated the use of a large fan and high air flow rates. Therefore, he recommended increasing the number of tubes used while decreasing the length of each tube in order to increase efficiency. The second system consisted of a single 17 m long, 150 mm diameter PVC smooth surface tube, 3.0 m below grade. Both sensible and latent cooling were observed. For this configuration ambient air was cooled from 32.2°C to 20.5°C at an airflow rate of 6.86 m^3/s. Long term cooling performance was greater during rainy periods.

Sinha et al. (1981) presented the results of a theoretical and experimental analysis at North Carolina A&T State University. The experimental design included four pipes buried 1.8 m to 2.4 m deep in a laboratory soil bin. Two pipes were corrugated, galvanized steel 450 mm and 300 mm in diameter, 18.3 m and 24.4 m long respectively, and the other two were plastic drainage tubes 200 mm and 300 mm in diameter, both 24.4 m long. Soil and air temperatures were measured along the pipe, and various airflow rates were tested.

The authors concluded that pipe material had little effect on the heat transfer performance probably due to their small thickness (1.6 mm to 3.2 mm). The 300 mm diameter pipes were more effective than the 450 mm pipe but there was insufficient data to confirm whether this was the optimum diameter. Finally, they concluded that increasing the airflow rate decreased the temperature difference achieved and very little tempering occurred beyond a length of 15.2 m (50 ft). Their recommendations were that more than one pipe of 15.2 m lengths be used rather than one pipe of equivalent length and that adjacent pipes be placed 4 to 5 pipe diameters apart.

Goetsch, Peterson and Muehling (1981) presented the results of field observation of earth-tube heat exchangers used as air tempering systems for swine ventilation systems in Illinois. The four systems monitored incorporated nonperforated corrugated plastic drainage tubing ranging in size from 130 mm diameter to 300 mm diameter. The tubes were buried between 2.4 m and 3.5 m below grade in soils of varying composition and moisture contents. Two of the installations had the ducts radiating from a sump. In the other two systems the ducts were laid parallel to each other at spacings of 1.2 m and 3.4 m. Air velocities ranged from 1.11 m/s to 2.67 m/s. Each system was attached to a swine farrowing operation. Beginning in the summer of 1980 air temperatures and relative humidity levels from the two systems with radial configurations were measured at the inlet and outlet end of each pipe. The results were then used in the design of the next two systems which were operational by the summer of 1981. The use of plastic pipes made these heat exchangers more economical than the metal pipe system studied by Scott et al. in 1965.

Results of the testing revealed that appreciable winter heating and summer cooling could be achieved due to the minimal variation in outlet temperatures. The most effective systems operated at airflow velocities near 2.54 m/s with a tubing surface area to flow rate ratio (assuming smooth pipe) of 295 to 393 $m^2/(m^3/s)$. Sandy soil was blamed for the poor performance of one system while small diameter tubes (130 mm) were believed to account for the inadequate behavior of another. Outlet temperatures from efficient systems varied only a few degrees in any 24 hour period. The authors also noted that system performance would be enhanced if the earth-air system was bypassed when the ambient air was suitable for direct use.

Design recommendations made by Goetsch et al. at that time included:

1. choose silt and clay soils over sandy soils,
2. use zone cooling rates for evaporative cooled air for design purposes,

3. use either radial or lateral systems but realize that total cost for lateral systems may be higher,
4. use a design velocity of 2.5 m/s and a ratio of surface area to system flow rate of 393 $m^2/(m^3/s)$,
5. bury tubes at depths of 2.1 m to 3.7 m below grade and space them 2.4 m to 3.0 m apart, and
6. allow at least a 0.25% slope to a free drain or a sump with a capacity of 18.9 L/h or more.

Goetsch and Muehling (1983) reported on three systems in central Illinois which used nonperforated, corrugated plastic drainage tubing buried at an average depth of 3.05 m to 3.40 m, to temper ventilation air for swine farrow-nursery housing. All ducts were 300 mm in diameter and ranged in length from 61.0 m to 79.2 m. These systems were monitored for a period of 3 years. Results showed substantial tempering could be realized for both heating and cooling at a 1981 cost of $1.18 to $1.77 per m^3/s ($2 to $3 per cfm). A simple economic analysis of the heat exchanger, excluding interest, yielded a payback period of 1.9 to 2.9 years. Changes to the design recommendations presented in their earlier paper allowed for the installation of systems into sandy soil providing that tube length and/or depth of burial were increased. Suggested tube lengths and airflow rates for tube diameters ranging from 76 mm to 610 mm were presented. Recommended lengths were based on a surface area (assuming smooth pipe) to system airflow rate ratio of 314 to 393 $m^2/(m^3/s)$. Finally, a recommendation was made to insulate the collection sump with moisture-proof insulation to at least 1.8 m below grade to a level of at least 3.35 $m^2 \cdot {}^\circ C/W$, (R-19). `.MYP A 25 stall farrowing house equipped with solar wall panels for heating ventilation air and underground ducts for heating or cooling incoming air was described by Cramer and Kammel (1980) and Kammel and Cramer (1981). The underground installation consisted of two branches of four, 150 mm diameter plastic drainage tubes. The tubes were buried 1.2 m below grade and sloped toward a drain. One branch, 21.3 m long, was located along the north side of the building. The second branch was located along the south side of the building and continued along the east side giving it a total length of 38.7 m. Monitoring included the recording of inlet and outlet air temperatures every three hours, as well as intermittent recording of airflow rates and static pressures. Results of the first summer's observations were reported in Cramer and Kammel (1980) while the first winter's performance was detailed in Kammel and Cramer (1981). The uneven duct lengths and small airflow rates reduced system efficiency. Summer data indicated a peak outdoor temperature of 29.4°C (85°F) in mid July and a peak ground temperature of 22°C (72°F) in mid August. The authors found that during a typical summer period, a modest cooling of 700 W to 1900 W (2400 Btu/h to 6600 Btu/h) was achieved. A similar time lag was evident under winter conditions with a minimum outside temperature of -22°C (-7°F) in December and a minimum outlet temperature of -6°C (22°F) in January. At an outside temperature of approximately 3°C (38°F), heat was neither lost nor gained by the ground. Linear relationships were developed for each duct to relate temperature change through the system to outlet temperature for December, January and February.

Baxter (1986) gave a comprehensive report on the performance of an earth-tube heat exchanger at the University of Tennessee campus in Knoxville. Initially, a vertical soil temperature profile was studied to determine the magnitude and seasonal variation of soil temperatures in order to establish the potential for an earth-tube system. This profile consisted of thermocouples placed every 0.3 m from 0.305 m to a 2.1 m depth. The author found distinct and predictable temperature gradients at each depth for each season. The soil temperature at 1.8 m below grade varied from a minimum of 7.5°C in mid February to a maximum of 20°C during mid August. This temperature range and depth was found to be favorable for the operation of an earth-tube system for winter heating and summer cooling. Use during spring and fall would provide little air temperature change.

After determining that a suitable temperature gradient existed, Baxter placed 64 m of 152 mm diameter corrugated metal pipe 1.8 m below grade in red clay soil. Temperature monitoring stations were located at the beginning and the end of the pipe, and at 15.2 m, 30.5 m, 48.8 m and 64.0 m along the pipe length. Thermocouples were located at each station to measure the temperature of the air at the center of the pipe, the temperature of the pipe wall, and the temperature of the soil at 51 mm, 102 mm, and 152 mm radially outward from the pipe in both horizontal directions. A vertical soil temperature profile, similar to the one just described, was located 1.8 m laterally from the center of the pipe length. Data were collected using a 24 point strip chart recorder. This restricted the amount of data which could be recorded at any time. Therefore, regular monitoring concentrated on measuring ambient air, outlet air and vertical soil temperature gradients. Occasionally lead wires were disconnected and connected to one of the other stations for monitoring. Air was blown through the system from the inlet end. Initially a fan rated at 8.21 m^3/min at 0.249 kPa gage static water head was installed. In fall, 1983, this fan was replaced with a fan rated at 15.01 m^3/min at the same static water head. The system was operated under both continuous, (24 hours a day), and intermittent, (8 hours a day), conditions with the exhaust air wasted to the environment.

The results reported were limited to those collected during the heating mode for the winter of 1985. The earth-tube system was on a 24-hour/day continuous operation. Baxter reported an inverse relationship between ambient air temperature and temperature gain. Predictablly the greatest temperature gain occurred during periods of cold weather with the gain decreasing as the weather became warmer. A maximum gain of 22.8°C was achieved with an ambient air temperature of -31°C. Baxter defined the exchanger efficiency as the ratio of the temperature difference between inlet and outlet air to the temperature difference between inlet air and the undisturbed soil temperature at pipe depth.

The effect of the phase change as the soil moisture froze was discussed in great detail. During the freezing process, the latent heat of fusion, 335 kJ/kg, was released into the airstream. This heat, when combined with the sensible heat already present, increased the performance of the system and caused the outlet temperature to stabilize around 1°C. As the reverse phase change occurred, that is as the soil moisture thawed, the heat of fusion was withdrawn from the air stream.

While ambient air temperatures rose quickly during this period, outlet temperatures rose slowly until the soil was thawed. The cold period experienced during testing was too short to determine the net effect of this phenomenon on the system performance, however, it was evident that the phase change began at the inlet end and slowly moved radially outward from the pipe and longitudinally along the pipe. The author hypothesized that the temperature plateau would have a relatively long duration if the soil moisture were allowed to freeze from the inlet to the outlet end.

Finally, Baxter presented a regression equation which related air temperature to distance along the pipe. This equation was very specific to his data and therefore would be of little use to other installations. Using this equation, he determined that 72.7% of the total temperature gain occurred in the first 15.2 m of pipe, 85.9% in the first 30.5 m, and 94.4% in the first 48.8 m of pipe.

Actual earth-air installations in Canada are not well documented. Lawand et al. (1983) summarized the results of the first stage of a study which investigated soil as an underground heat storage medium. The test facility included two test beds, 6.0 m long, 1.0 m wide, and 1.0 m deep, installed in an experimental greenhouse. The air ducts were 102 mm diameter, corrugated nonperforated drain pipe placed under 0.5 m of cover. One pipe was installed in sand while the second was installed in sandy loam. Warm air was blown through both pipes. Airflow rates were monitored using a hot wire anemometer and balanced at 5.1 m/s for most tests. Thermocouples were used to measure soil and air temperatures at 1.5 m, 3.0 m, and 4.5 m from the inlet end of the pipe.

At each sampling point air temperature at the center of the pipe, soil temperature at the pipe wall, and soil temperature radially outward from the pipe in three directions to a distance of 500 mm were measured. Temperatures were monitored, averaged and recorded at 30 minute intervals. Six tests, which studied heat input rates from the air to the soil, were conducted on each soil type. Airflow rates from 3.2 m/s to 5.1 m/s were observed. Results showed that 156 W·h/day·m of pipe of heat energy was transfered from the air to the soil for storage. This gave daily heat transfer rates from 26 W/m to 106 W/m, while the rate of heat extraction from the soil varied from 5 W/m to 10 W/m. Inlet temperatures ranged from 4.3°C to 40.1°C in tests conducted on the sandy soil. From these, outlet temperatures of 5.6°C to 29.9°C were achieved. Likewise, for the sandy loam soil with inlet temperatures ranging from 9.9°C to 51.2°C, outlet air temperatures of 11.2°C to 32.9°C were recorded. Initial soil temperatures near the ducts ranged from 3°C to 16°C. The authors concluded that heat transfer rates were mainly dependent on the temperature gradient between the incoming air and the soil and the airflow rate through the duct. No difference in heat transfer rates and heat storage capacity were observed between the two soils tested. Standard equations for flow in a pipe were used to calculate theoretical heat transfer rates. These rates compared favorably with measured values if the total ribbed surface of the corrugated pipe was used in the calculations. The soil storage system behaved like a pipe buried in an infinite medium with unsteady state conduction in the soil during charging and discharging. The authors recommended a center to center duct spacing of 500 mm and depths of 300 mm.

A more recent Canadian study was reported by Jackson et al. in 1986. This study investigated the use of 40 m of 150 mm diameter, nonperforated corrugated plastic drainage tube buried 2 m below grade. Airflow was maintained at approximately 60 L/s. Despite the 1% slope from the inlet end to a sump at the outlet end, the high groundwater level (1.5 m to 1.0 m depth) caused occasional blockage of airflow due to water. Data collected showed a 7°C temperature drop in June and an 11.5°C temperature rise in January. Further evaluation of the results found that outlet temperatures were only slightly influenced by changes in daily ambient temperatures but varied with ambient temperature on a monthly basis. They concluded that earth tubes could be effectively used for heating and cooling ventilation air.

2.2 THEORETICAL ANALYSIS

Analytical analysis of earth-to-air heat exchangers is useful for determining design specifications for a new system or for modelling the performance of an existing system. Much of the early work on the theory of earth coupled heat exchangers was carried out by Ingersol and Plass (1948) and Ingersol et al. (1951). This work was an elaboration of Kelvin heat source theory based on the assumption of an infinitely long permanent line heat source or sink in an infinite medium at an initial uniform temperature.

Problems encountered in developing a theoretical model were:

1. the lack of knowledge of thermal conductivity and diffusivity of soil in a given location,

2. moisture migration from warmer to cooler regions,
3. underground water movement,
4. ice formation around the pipe,
5. periodic variation of soil temperature,
6. cyclic rather than continuous operation, and
7. boundary layer effect due to lack of intimate contact between the pipe and the soil.

The authors found that appreciable error may occur using line source theory with large pipe diameters or short time intervals. As a general criterion, $\alpha t/R^2>20$, (where α is the soil thermal diffusivity, t is the time from start of operation and R is the pipe radius), will produce small error. A more general solution involving Bessel functions was given by Carslaw and Jaeger (1959) but it is more difficult to evaluate.

Ingersol et al. (1951) concluded that relatively isolated pipes gave superior performance to planar exchangers or roughly spherical cavities on the basis of heat exchange rate per unit area. Small pipes are more efficient than larger ones on this same basis. A hairpin loop, two pipes well separated in the same trench, were only slightly less efficient than two isolated pipes. This efficiency would be further reduced if two such loops were placed in the same trench. They also determined that lack of ground temperature recovery in the summer will not be a problem unless winter operation cools a block of ground deeper than 6 m. Efficiency will be enhanced due to moisture migration to the cold pipe surface during winter operation; however, efficiency will be only slightly improved due to soil moisture freezing. Underground water movement will influence performance only when the velocity exceeds 0.05 mm/s (0.01 ft/h).

Guernsey et al. (1949) also used line source theory in numerous calculations of heat flow in earth heat sources under several specific conditions. Results of this study suggested that the ground coil systems required would be so large that installation costs made general use impractical. However, this analysis has been criticized for neglecting important factors such as the freezing of soil moisture, which resulted in seriously oversized designs.

Periodic fluctuations in soil temperature due to diurnal and annual changes in ambient conditions must also be accounted for. Prediction of these fluctuations was possible using Carslaw and Jaeger's (1959) solution to the partial differential equation for soil temperature at a depth x below the surface. Scott et al. (1965) expanded on this approach to include the temperature distributions due to the earth-air complex. Assumptions made in the development of this theory included:

1. soil conductivity is neither temperature nor location dependent,
2. air temperature is constant in the plane perpendicular to flow direction,
3. there is no heat conduction in the air in the direction of flow,
4. pipe of infinite conductivity perpendicular to airflow and of zero conductivity in direction of flow, and
5. there was finite soil conductivity perpendicular to flow and zero conductivity in the direction of flow.

This analysis was undertaken to provide a simplified steady state approach to the solutions present by Ingersol et al. (1951) and Carslaw and Jaeger (1959). Equations were given to predict outlet air temperatures and the overall heat transfer coefficient from soil conductivity and soil temperature. This method provided a conservative (1 to 8% error) prediction of outlet air temperatures using experimental values for the inlet temperature and far field temperature for a point in time and assuming no heat storage or recovery.

Scott et al. (1984) developed a periodic steady state solution which proved to be accurate in predicting results for cyclic fluctuations in summer but gave less precise results when temperatures varied erratically in winter. A design chart was developed based on the model so that suitable pipe size, pipe length and air velocity could be selected using only simple hand computations.

Svec and Palmer (1980) created a two dimensional finite difference program to model a vertical heat exchanger of PVC pipe. The simulated results compared favourably to data collected from a laboratory scale model and a prototype field installation. Their results indicated that the behavior of a ground couple heat exchanger is governed mainly by soil conductivity and to a small degree by the thermal properties of the heat exchanger. As well, they concluded that in compacted saturated soil, convection becomes negligible.

Britton (1981) developed a finite difference program to simulate a horizontal earth-to-air heat exchanger. The program, based loosely on the theoretical analysis by Scott et al. (1965), was used to determine practical pipe lengths, and the effective heat source/sink radius. Steady state conditions with heat flow in the radial direction only were assumed. The analysis included provision for the release of the latent heat of fusion during the freezing of soil moisture. The model was not developed beyond the point where it could be used to show trends.

Puri (1986) presented design curves developed from a time dependent axisymetric finite element formulation of an earth tube system. The program allowed for simultaneous solution of heat and moisture diffusion equations based on a model proposed by Philip and De Vries (1957) and validated by Ahmed et al. (1983). The model considered pipe diameter, pipe length, initial soil moisture content, initial soil temperature, and ambient air temperature. Design curves for both heating and cooling are included based on a 12 m length of tube. For operating times of less than two hours, the tube acted as an infinite length and the system returned to its initial state during two hours of off time. For an operating time of 12 h, the efficiency was reduced to 50% of an infinite length. A spacing of eight tube diameters was suggested to ensure isolation of adjacent ducts.

Diener et al. (1986) simulated the performance of an underground earth tube counterflow heat exchanger system using an energy balance. Two extreme cases, infinite thermal resistance between outer duct and the soil, and zero thermal resistance between the outer duct and the soil were considered. It was expected that actual performance would fall between these two extremes with some heat transfer with the soil occurring. The analysis offered a very simplistic energy balance but did not account for the effects of moisture condensation. A Fortran program was used to compare results of the two cases.

Lei (1985) presented a comprehensive finite element analysis which incorporated pipe diameter, pipe length, air velocity, soil texture, soil thermal properties, convective heat transfer coefficient, and undisturbed soil temperature. The model compensated for hourly, daily and seasonal air temperature variation and was valid for modelling both intermittent and continuous operation. Design recommendations presented included the use of lower airflow rates and shorter pipes and the use of high density and high moisture content soils. He also concluded that the lower thermal conductivity of plastic pipes make copper pipes more efficient. Results were presented in the form of simulated curves and approximate equations to be used for preliminary design of an earth-air heat exchanger.

Spengler and Stombaugh (1983) used a two dimensional finite element analysis to evaluate the thermal and economic performance of earth tube heat exchangers for the ventilation of swine housing. They recommended, as simplified design guidelines, the use of a winter ventilation rate 30% above the minimum recommended rate. With this flow rate and the desired tube diameter, the number of tubes required to maintain winter flow velocity at or slightly below 1.5 m/s can be specified. Next, determine the tube length required to obtain a tube surface area per unit airflow rate of 550 $m^2/(m^3/s)$, neglecting increased area due to corrugations. The authors concluded that the amount of heat exchange achieved was significantly influenced by flow rate and tube diameter. Also, the thermal performance at a given flow rate was not seriously affected by tube material, tube diameter, or soil type as long as the soil moisture was greater than 50% of saturation. Economically, the optimum design would not eliminate the need for supplementary heat but would substantially reduce energy costs.

2.3 GROUND SOURCE HEAT PUMPS

Much of the literature written in the area of using the earth as a heat source/sink focuses of the use of ground source heat pumps. Ground source heat pump systems are simply heat pumps with the evaporator coil buried underground. Unlike the systems previously discussed, the circulating fluid is generally water or an antifreeze solution. Therefore, the fluid is generally circulated in a closed loop as opposed to being used for direct ventilation. The ground coils for the heat pump may be placed either horizontally or vertically in the soil.

Ball et al. (1983) presented a comprehensive review of the historical and current methods for designing ground-coil heat pumps. They reported that a Swiss patent issued in 1912 to Heinrich Zoelly was the first record of using the earth as a heat source for a heat pump. From World War II until the early 1950's, considerable research into the design of such systems for North American conditions took place. Studies were undertaken to examine the effect of pipe size, pipe spacing, depth of burial, and extended surfaces on the soil side of the pipe. Bell et al. (1983) also presented an extensive discussion of design methodologies. A brief explanation of each of the following design techniques were given:

1. Rules of thumb
2. Steady-state and transient analytical models

3. Lumped parameter methods
4. Finite difference in one, two, or three dimensions
5. Finite element in two dimensions

Tables were included which summarized typical overall seasonal performance factors, outlined the effects of heating on the overall heat conductivity, and listed and described existing models. The report also mentioned the shortcomings of many of the existing models. The authors were successful in providing an excellent summary of the evolution of ground source heat pumps, and reported that adequate design guidelines were not publically available at the time.

Smith (1951), Baker (1953), Smith (1956), Vestal and Fluker (1956), Metz (1983), and Klimkowski et al. (1985) investigated the use of horizontal earth coil arrangements. These installations consisted of pipes made of different materials including copper, polyethylene, and PVC, as well as different pipe sizes ranging in diameter from 12.5 mm to 31.75 mm and ranging in length from 45.7 m to 366 m. The ground coils were buried from 1.2 m to 2.0 m below grade. In all installations the importance of being able to accurately determine/predict the soil conductivity was recognized. Klimkowski et al. (1985) reported that because U values were initially high but decreased with running time, the minimum U value should be determined and used in the design process.

Braud, Klimkowski, Baker and Oliver carried out extensive work on ground coupled heat pump applications in Louisianna from 1981 to 1986 focusing on closed loop water source heat pumps, (Braud et al. (1983a,b, 1986), Braud (1983), Klimkowski et al. (1985, 1986)). Studies included monitoring of a horizontal heat exchanger of 31.75 mm diameter PVC pipe, a vertical concentric pipe heat exchanger with a 63.5 mm steel casing and 31.75 mm inner PVC pipe, and a 31.75 mm polyethylene U-bend heat exchanger. Results of all installations showed favorable energy savings.

2.4 SOLAR ENERGY STORAGE

With the recent increase in the use of solar energy has come the challenge of finding an inexpensive and efficient means of storing excess energy for future use. Much research has gone into the possibility of storing solar energy underground. This can be achieved by simply transferring solar energy to the soil via buried pipes, or by storing energy in insulated shallow earth areas, or in specially prepared gravel layers. These methods of storage ensure that much of the solar energy collected in the summer is available for use in the winter when the need is greatest. However, in order for storage to be practical, the cost of retrieving this energy for winter use must be compatible with that of other available fuels.

Shelton (1975), Nicholls (1978), Bose et al. (1979), Metz (1979), Fischer et al. (1979), Andrews and Metz (1979), Schaetzle et al. (1979), Andrews (1981), Simonson and Coleman (1981), Mustacchi et al. (1981), Platell (1981), Spryszynski (1984), and Meyrier and de La Casiniere (1985) studied the various aspects of underground storage of solar energy. Shelton (1975) listed the three most practical problems in designing underground heat storage as:

1. estimating water fluxes,
2. estimating thermal conductivity and heat capacity of soil, and
3. determining the ease of using or suppressing heat flow upward from storage.

Mustacchi, Cena and Racchi (1981) determined that a two year period was required for any system to achieve steady state conditions. Shelton (1975) on the other hand, suggested that steady state conditions could be reached in the order of one year. Mustacchi et al. (1981) also suggested that the heating front seldom moved more than two meters in six months. They suggested 1.0 m to 1.5 m of 25.4 mm diameter pipe should be used per square meter of collector as a useful design guide when using a hairpin loops configuration. Larger pipe sizes showed greater heat flux into the soil but performance had to be weighed against pipe cost in determinimg an optimum design.

Most systems reported in the literature realized favourable energy savings; some found doubtful or no benefit at all. Andrews (1981) studied heat pump systems using energy withdrawn from the ground to determine the benefit of adding solar energy input to the system. Given favorable performance of ground-source heat pumps, he suggested that the addition of solar energy would only be attractive if collector and installation costs could be greatly reduced. Meyrier and de La Casiniere (1985) stated that because of the low mean soil temperature in France long term storage of solar heat by this method was inefficient even for well insulated homes.

Andrews and Metz (1979) presented the results of a three dimensional time dependent FORTRAN program to study heat flow in the soil. The objectives of the study were to compare the results of the computer program called GROCS, to published values, and to determine optimum values for soil thermal conductivity and heat capacity. Experimental results revealed a heat transfer factor, (the rate of heat uptake per unit time, per unit length of pipe, per unit temperature difference between the pipe temperature and far field temperature) ranging from 3.3 W/°C·m to 3.6 W/°C·m.

The possibility of storing solar energy from greenhouse air in the soil bed was investigated in British Columbia, by Staley et al. (1983). The theory used was based on work done in Japan (Yamamota, 1969) and was adaptable to British Columbia due to similar climates. Greenhouse air, solar heated above a set point temperature was drawn through pipes buried in the soil bed. Heat was released to the soil and the cooled air returned to the greenhouse. When the greenhouse air temperature dropped below 17°C, the airflow direction was reversed and heat was retrieved from the soil. The efficiency of the system was dependent on ambient air temperature and the intensity and duration of the solar energy. This study focused on greenhouse performance and no soil temperature data was reported.

Coffin and Alward (1985) studied similar systems at the Brace Research Institute, Macdonald College, Quebec. They monitored two installations of horizontal plastic drainage pipe buried at shallow depths. The first arrangement was placed 30 cm below the soil surface under a solar greenhouse, the second directly beneath a concrete floor slab. Both systems demonstrated the potential to store excess solar energy for short periods and to stimulate plant growth through increased soil temperatures. The authors presented a brief design guide to assist in the construction of similar systems.

2.5 SOIL THERMAL PROPERTIES

Perhaps the most important factors to consider in designing and evaluating any earth-to-air heat exchanger, are the properties of the soil in which it is placed. The performance of the system will be affected by the soil temperature, soil thermal conductivity, soil thermal diffusivity, soil moisture content, ground water movement and the contact resistance at the soil pipe interface. (Lei, 1984). The soil temperature must be such that a favorable temperature gradient exists between the incoming air and the soil and the soil thermal properties must be such that heat transfer due to this gradient can readily occur. (Baxter, 1986). Each of the important parameters will be treated in more detail below.

2.5.1 SOIL TEMPERATURE

Soil temperature is influenced by soil depth, texture, moisture content, surface cover, landscape position and man's manipulation of the site. Temperatures near the surface are more strongly influenced by diurnal and seasonal climatic conditions than temperatures at greater depths. Ground cover will affect soil temperatures due to the insulating effect which is created. Algren (1949) monitored a silt and clay type soil in Minnesota and found that the frost line extended to a 1.0 m depth on a barren site but only to 0.3 m on a sod covered site.

Khatry et al. (1978) presents an explicit expression for soil temperatures as a function of depth and time for Kuwait. The expression was derived from recorded periodic variation of solar radiation and atmospheric temperature rather than the assumed periodic variation of surface temperatures. Costello et al. (1984) used multiple linear regressions to fit a sine wave to average monthly shallow soil temperature data. The regression results along with an estimate of soil thermal diffusivity were applied with simple conduction theory to estimate the soil temperature at any depth. This approach was modified by Lei in 1984, to estimate monthly or daily soil temperatures and soil thermal diffusivity at Glenlea, Manitoba. Assumptions made in developing Lei's model were:

1. soil is a homogeneous medium,
2. thermal diffusivity does not vary with depth, direction, or time,
3. surface temperature is approximated by a sine wave with a period of one year,
4. soil temperature varies with depth only and there is therefore no heat transfer in the horizontal direction, and
5. at arbitrarily large depths, soil temperature approached a constant mean value.

Both studies concluded that the one-dimensional model predicted reasonably accurate undisturbed soil temperatures when compared with local soil temperature data. Baxter (1986), who actually monitored soil temperatures from 300 mm to 2130 mm below grade from 1981 through 1985, found that distinct temperature patterns occurred each year at each depth.

2.5.2 THERMAL CONDUCTIVITY

Kersten (1948) conducted thermal conductivity tests on nineteen different soils representing gravel, sand, sandy loam, clay, crushed rock and a fibrous peat. Tests were conducted over a wide range of mean temperatures, moisture contents and densities.

Temperature was found to be significant only in determining whether the soil sample was above or below freezing. When moisture content increased above 6 to 12%, the conductivity of frozen soil became greater than that of unfrozen soil. On average, for all soils at any temperature and any moisture content, a 16 kg/m^3 (1 lb/ft^3) increase in density caused thermal conductivity to increase by about 3%. An increase in moisture content, up to saturation, also caused increased thermal conductivity. Coarse textured soils generally had a high thermal conductivity while fine textured soils had relativly low conductivity values. Kersten summarized the results of his testing in four charts which gave conductivity values within 25% of their true value. The charts were for frozen and unfrozen sand or sandy soil and frozen or unfrozen silt and clay soils.

Carter (1951) reported on the results of a soil testing program in the Tennessee Valley Area from 1949 to 1951. The program included monitoring of soil temperature and moisture content, and the experimental determination of thermal conductivity and thermal diffusivity of local soils. The author stated that for moisture contents in the 10 to 25% range, thermal diffusivity varied slightly. However, thermal conductivity increased with an increase in moisture content up to a maximum dependent on the liquid and plastic limits of the soil. The increase in thermal conductivity with moisture content was slower for fine-grained soils than for course-grained soils. With increasing moisture contents, density and specific heat increased more rapidly than conductivity.

Smith and Yamauchi (1950) stated that conductivity varied greatly with density and only slightly with moisture content. They determined k using three methods, a cylindrical ring laboratory method, a sphere buried in undisturbed soil, and a cylinder used in a transient type test to measure thermal diffusivity. The cylindrical ring method was the most accurate; however each test took three to four hours to run. If approximate results were sufficient, the transient method was found to be most convenient and time effective.

Parkerson (1951) presented a series of graphs and simple formulae for calculating thermal characteristics of a given soil in its dry-saturated or frozen-saturated state using the dry apparent density of an undisturbed soil sample. Parkerson suggested that since saturated soil maintains good contact, saturated soil characteristics should be used for ground coil heat source design. Alternately, soil around a hot pipe dries out and contact decreases, therefore for a heat sink design, dry soil properties should be used.

Work at the University of Manitoba by Sawatzky (1985) and Janzen (1985) focused on measuring soil thermal conductivity using a simplified version of the guarded hot plate method. Sawatzky designed and built the apparatus but found difficulty in accurately placing the thermocouples. Janzen modified the apparatus and tested clay and sand soils at three moisture contents, two mass flow rates, and two different hot bath temperatures. The testing results compared favorably with published data. Unfortunately, the testing is restricted to laboratory conditions and is time consuming.

2.5.3 THERMAL DIFFUSIVITY

Scott et al. (1965) provided two equations for calculating soil thermal diffusivity. Both equations required undisturbed soil temperature data at any two depths below the surface. The first equation was based on the ratio of temperature amplitudes at any two depths while the second equation was based on the time between the occurrence of maximum temperature at each depth and the distance between depths. Carter (1951) reported actual field measurements of thermal diffusivity for moisture contents ranging from 0 to 30 percent. He found the increase of soil diffusivity with moisture content to be inversly proportional to the liquid and plastic limits of the soil.

2.5.4 CONTACT RESISTANCE

Maximum contact between the pipe and the soil will increase the amount of heat flux possible. When the pipe is warmer than the surrounding soil, the soil moisture is driven away. Sufficient moisture migration will cause the soil near the pipe to become dry, crack and pull away from the pipe. This reduced contact will substantially reduce the heat flux. Pappas and Fresberg (1949) recognized the importance of contact resistance. In an effort to find a material which would maintain good contact, they ran laboratory tests on several mixes of benonite, silica gel and other hygroscopic materials. Results showed that samples with benonite developed large cracks and therefore were considered of little value while plain sand, sand with a small amount of silica gel to attract moisture, and sand mixed with bentonite and oil all tested favorably.

Smith (1956) and later Svec et al. (1983) proposed an apparent thermal conductivity (k_a) to account for the film effect created between the pipe and the soil due to lack of intimate contact. Using field data and a steady state equation Smith determined (k_a/k) in the range of 0.466 to 0.960.

2.5.5 UNFROZEN MOISTURE CONTENT IN FROZEN SOILS

Many people have studied the unfrozen moisture content of frozen soils (Dillon and Andersland (1966), Anderson and Tice (1971), Williams (1972), Anderson et al. (1973), Tice and Sterrett (1980), Frivik (1980), Phukan (1985)). The unfozen moisture exists as a liquid layer separating ice from organic or mineral components. Anderson and Tice (1971) reported that this layer could range in thickness from 3 to 50 A.U. or more. Dillon and Andersland (1966) attributed the thickness of the water layer in frozen clays to the colloidal activity or the suction characteristics of the soil. Since these suction forces are stronger in clay soils than in course-grained soils, the unfrozen moisture content in clays is greater than in coarse-grained soils. Williams (1972) gave the example that at -3°C as much as half of the moisture present in clay may be in the liquid state while for sand at the same temperature, a tenth or less of the moisture present would be liquid. In fact, he further stated that, " . . .some clays contain more liquid water at -2°C than coarse-grained soils when saturated at above freezing temperatures.

An understanding of the unfrozen moisture content of soils is necessary when doing freeze/thaw calculations since if all of the water present in a volume of soil is assumed to freeze at 0°C, errors of 10 to 30% can be expected. Under extreme cases these errors may be as high as 40 to 100% (Williams 1972). The unfrozen moisture content of frozen soils may be determined by the calorimeter method (Anderson and Tice 1971, Anderson et al. 1973), by prediction equations using soil properties such as specific surface area and activity ratio (Dillon and Andersland, 1966), of by nuclear magnetic resonance (NMR) (Tice et al. 1980).

2.6 SUMMARY

A detailed account of literature relevant to this thesis has been given. It has been shown that while several earth-to-air heat exchangers are currently operating effectively around the world, the amount of reliable data on which to base future designs is small. Very little in-field research has been carried out in western Canada where the annual temperature fluctuations may be in the order of 60°C. The effect of these extreme temperatures on the soil surrounding the heat exchanger over extended periods of operation has yet to be determined. As well, of the many models which have been developed to date, most neglect important factors necessary in predicting system performance such as soil thermal properties, thermal properties of the heat exchanger, effect of soil moisture and moisture migration and the effect of condensation. It was for these reasons that Agriculture Canada sponsored a research contract with the Department of Agricultural Engineering in 1984. Mr. Quanmin Lei has developed one of the most complete and comprehensive finite element models to describe the behavior of an earth-to-air heat exchanger as one part of this contract. To enhance his model, this thesis will report the data collected from a full-scale prototype field model of an earth-to-air heat exchanger. So, to restate the objectives of this thesis, they are:

1. to evaluate the performance characteristics of a buried pipe ventilation air tempering system at Glenlea, Manitoba, and
2. to determine which factors among pipe diameter, pipe length, and airflow rate, have the greatest effect on system performance.

The next section will describe the design of the experimental installation used to achieve these goals.

Chapter III

EXPERIMENTAL DESIGN

3.1 INSTALLATION

A site at the University of Manitoba Glenlea Research Station was chosen for the experimental installation. Shelter belts border the south, east and west sides of the plot with a dike running along the north border (Figure 3.1). The location is isolated from pedestrian traffic and is equipped with electrical services.

The design involved burying two 150 mm diameter plastic PVC pipes and two 250 mm diameter plastic PVC pipes, each 30 m long to an average depth of 3.0 m below grade. Pipes were placed parallel at 5 m on center with a slope of 0.8% from the inlet to the outlet end. At the inlet end the ducts extended one meter above grade with a 3.0 m horizontal extension and a screen covered elbow to keep rodents and foreign debris out (Figure 3.2). At the outlet end, the ducts emerged from the ground into a common plywood header allowing for the use of a single centrifugal suction fan. A plywood shed was built around the fan to protect it.

Earth work at Glenlea was complicated by the lack of stability of the Red River Gumbo Clay soil located at the site. On the advice of hired contractor Mr. Paul Nolette, it was decided that the entire site, 25 m x 40 m would be excavated to a depth of 1.0 m and then four, 2.0 m deep

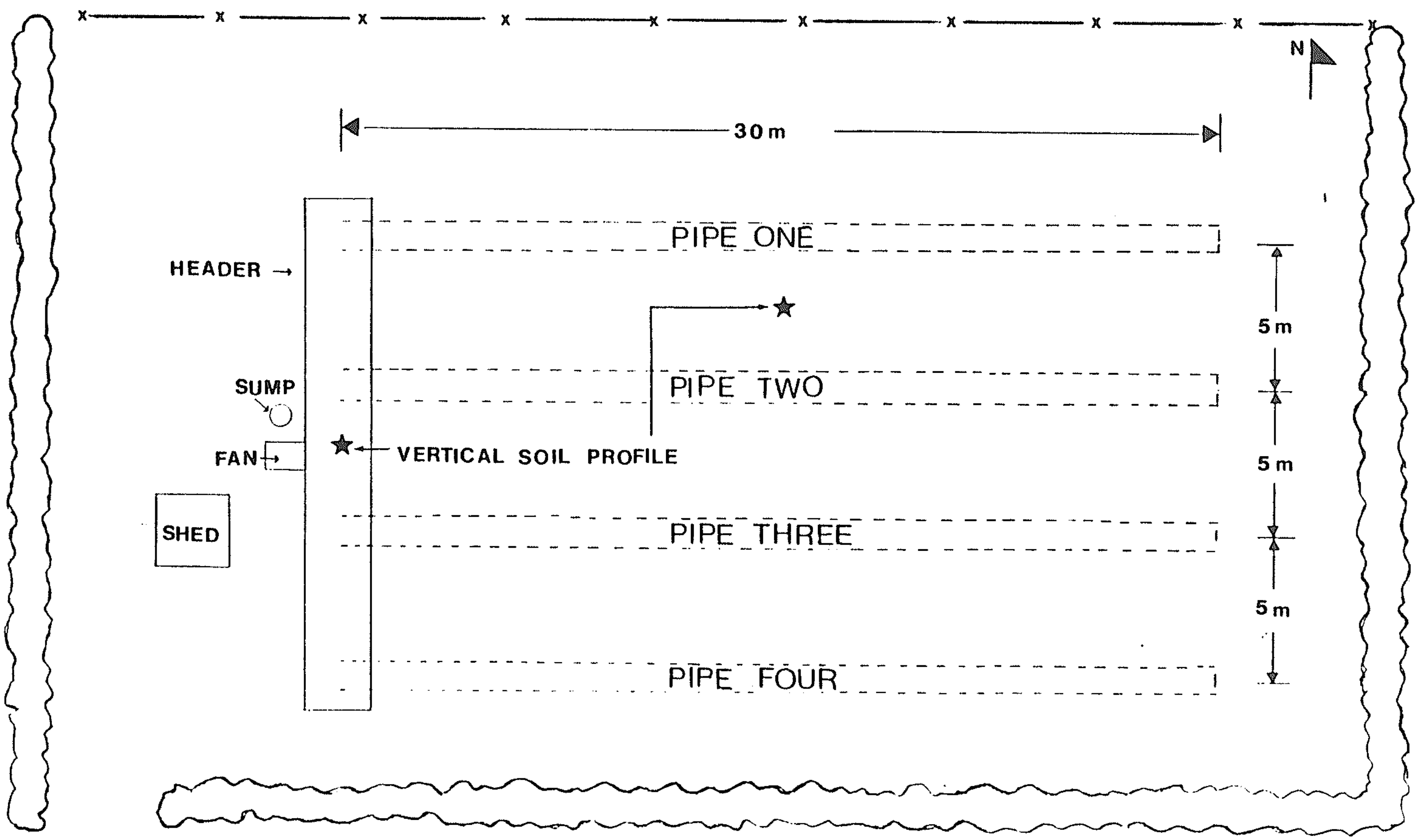

Figure 3.1: Layout of experimental site at the University of Manitoba Glenlea Research station.

PIPE	DIAMETER (mm)	AIRFLOW (m^3/s)
①	250	0.05
②	150	0.15
③	250	0.10
④	150	0.05

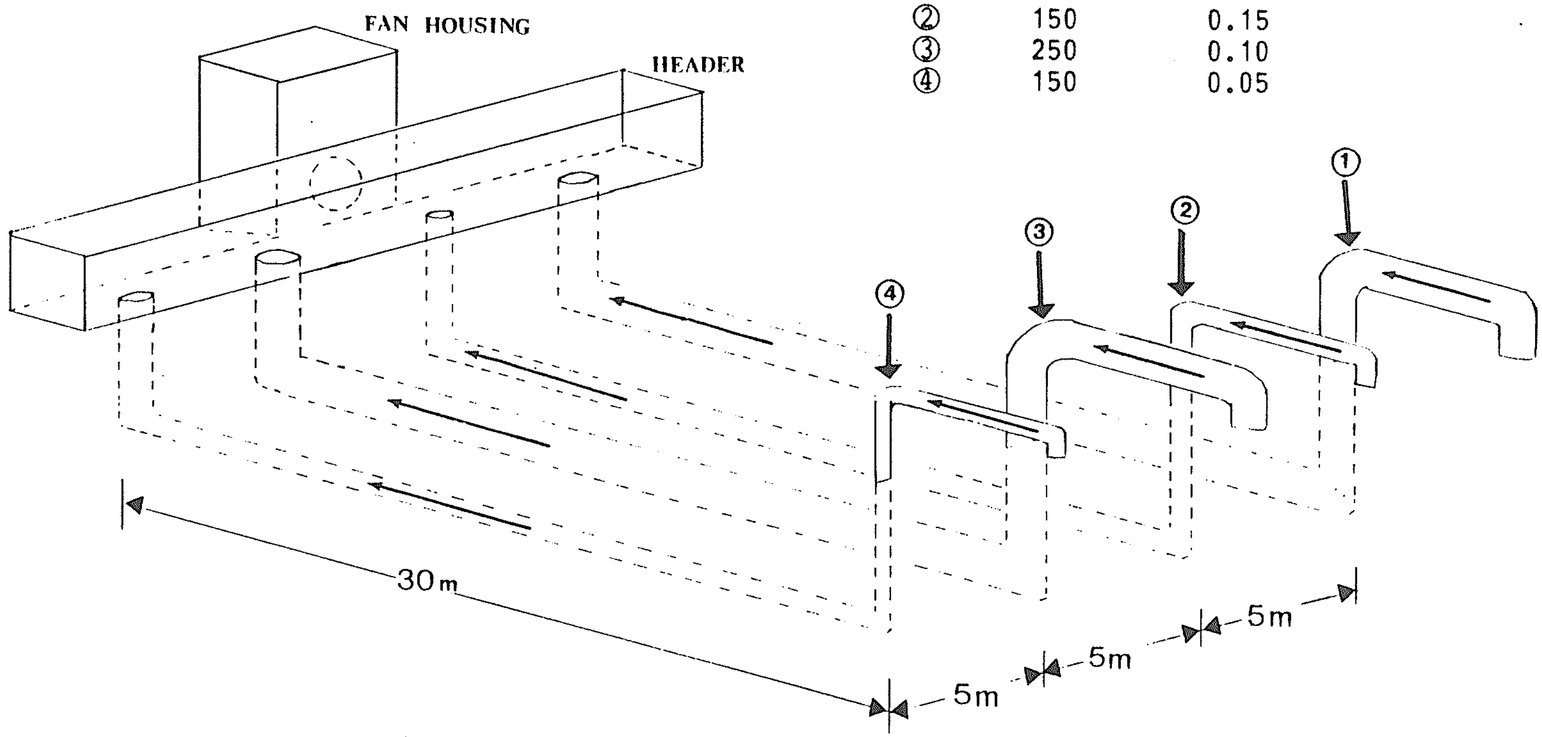

Figure 3.2: Placement of pipes in experimental layout.

trenches, would be dug to receive the pipes (Figure 3.3a). However, after the first trench was excavated, unstable soil caused the trench walls to cave in. To overcome this instability an additional 1 m of soil was excavated from the entire site, and trench depth was reduced to 1 m. This still allowed for the desired burial depth of 3 m below grade. (Figure 3.3b).

With the first trench dug, placement of equipment was about to begin. However, during the next month a record 171.5 mm of rain was recorded at the Glenlea site causing extensive flooding of the excavation. As many as three pumps were operated at once to drain the site and to allow drying to commence. It became evident that the water table had risen and an additional delay would be required before trenching could be resumed. In order to avoid this delay, a basic design change was made.

It was decided that the duct work would be laid directly on the floor of the excavation and an extra 1.0 m of cover would be supplied during backfilling (Figure 3.3c). This would provide the 3.0 m of cover, and minimize scheduling delays. As well, laying the pipes directly onto the excavated surface would facilitate thermocouple placement. (Thermocouple installation will be discussed in the next section.)

The location of each duct was established and shallow trenches were dug with a 0.8% slope toward the outlet end. Four meter sections of pipe were placed in the shallow trench. Sections were cemented together with PVC cement and sealed with silicone to eliminate air leaks. Ninety degree elbow joints were attached to each end of the 30 m ducts to support vertical sections of pipe for the inlet and outlet.

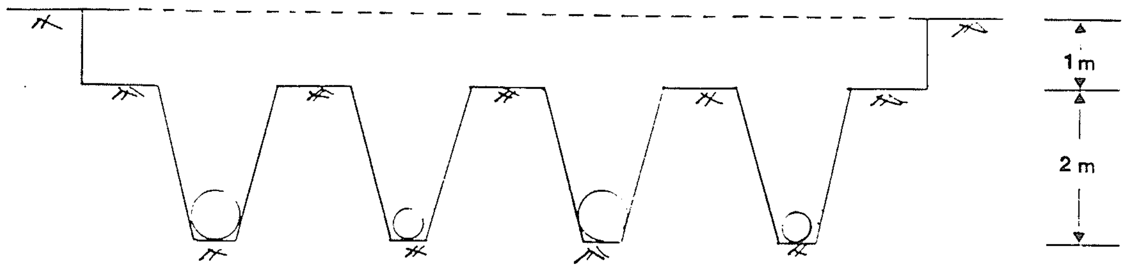

Figure 3.3a: Original excavation plan.

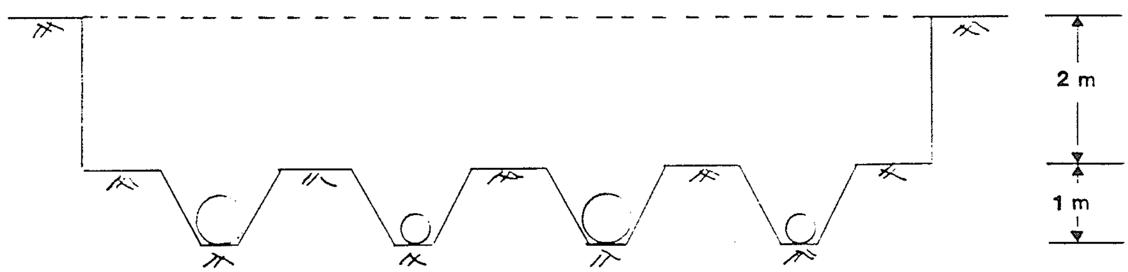

Figure 3.3b: Revised excavation plan.

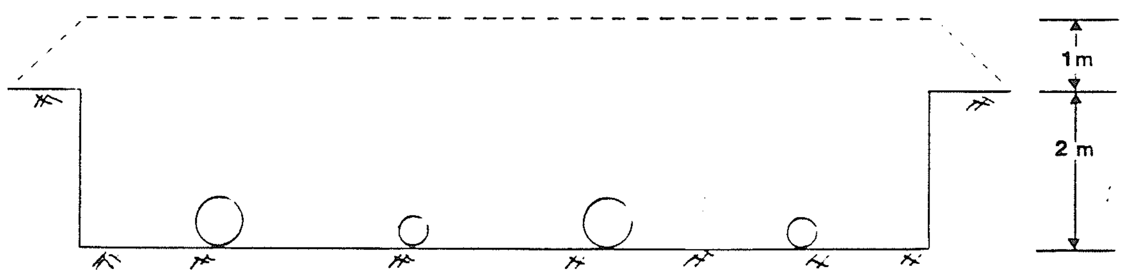

Figure 3.3c: Ultimate excavation plan.

To allow for condensation to drain from the ducts, 19 mm semi-rigid tubing was connected to the bottom of each elbow at the outlet end. The tubing was then fitted to a sump located several meters away. The slope allowed drainage to occur.

Much of the initial backfilling was done by hand to maximize contact between the pipes and the soil, and the thermocouples and the soil. Soil clumps were broken up and the loose soil was compacted around the ducts. Approximately 1.5 m of cover was supplied over the pipes and the thermocouple wires before heavy construction equipment was allowed to move over them. This was done to reduce the possibility of damage due to the weight of the machinery. Large earth moving equipment completed the backfilling. Similar soil from a nearby site was used to provide the additional one meter of cover.

3.2 TEMPERATURE MEASUREMENT

A total of 330 thermocouples were used to monitor soil and air temperatures throughout the installation. All thermocouples were 2.032 mm copper-constantan. The length of each thermocouple lead wire was calculated during initial design so that all wire could be cut in the laboratory. One end of each length of wire was stripped and the copper and constantan leads were soldered together. Each wire was separated, marked with colored tape and numbered to identify its location in the soil. As the thermocouples would be exposed to the elements of soil and air, and temperature and humidity fluctuations it was necessary that they be placed in such a way that they would take accurate readings for the life of the project (3 years) and at the same time remain accurately located despite soil shifting.

To achieve accurate thermocouple placement, a piece of 12.7 mm semi-rigid tubing was cut to approximately 1 m, and small holes were drilled along the tube at 250 mm intervals. Thermocouples were then threaded through the tube and the extended out of the drilled holes and covered with silicone sealant to hold each thermocouple in place and to protect the connection from exposure and corrosion. The tubing was then filled with foam insulation to eliminate any effective heat transfer along the hollow shaft.

From work by Britton (1981) it was believed that little temperature change should occur more than 1.0 m from the pipe surface, therefore no thermocouples were located past this point. Temperature sensing was done in three directions: vertically down from the pipe, vertically up from the pipe, and in a horizontal direction perpendicular to the axis of the pipe. Figure 3.4 shows a detailed layout of thermocouple locations. Air temperatures were also measured at each sensing station. Additional thermocouples were used to measure ambient air temperature on site, and air temperature at the inlet and outlet of each pipe. Monitoring stations were located at 1 m, 5 m, 10 m, 15 m, 20 m, and 29 m from the inlet end along each duct. At the 5 m sensing station, soil temperatures were measured at 250 mm and 500 mm from the pipe surface. At the 15 m sensing station soil temperatures were only measured at a distance of 250 mm from the pipe surface. All other sensing stations had thermocouples located at 250 mm intervals from the pipe surface to one meter from the pipe surface. Thermocouples were concentrated at the inlet end where the greatest temperature change would occur.

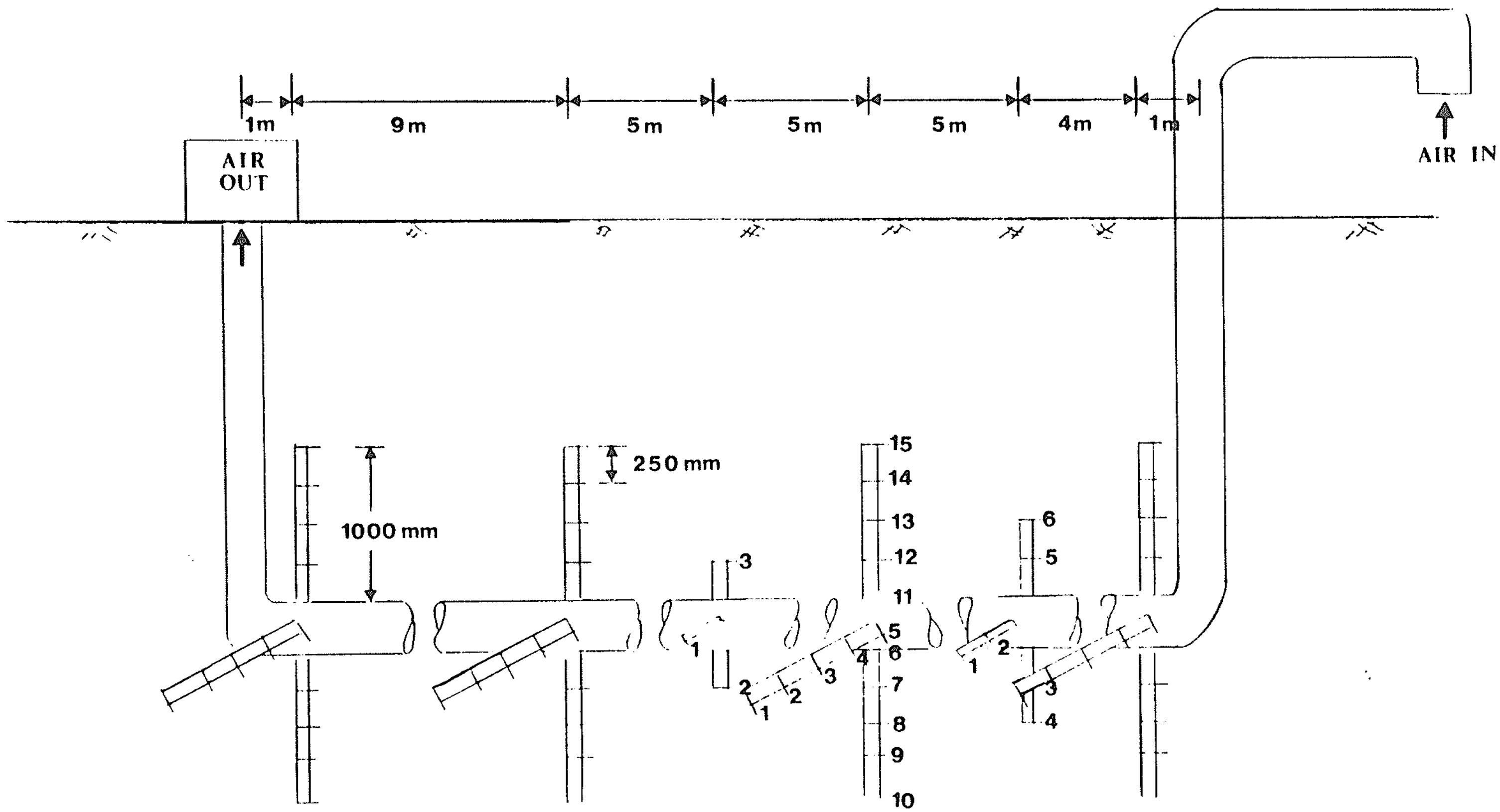

Figure 3.4: Thermocouple placement in soil, identical for each pipe. One thermocouple at the center of the duct at each monitoring station.

Two vertical soil profiles were monitored to provide background soil temperatures, also referred to as far field temperature. Each profile consisted of eight thermocouples situated from 0.5 m below grade to 4.0 m below grade at 500 mm intervals. These thermocouples were again connected to semi-rigid tubing to ensure accurate placement in the soil. One profile was located between the two middle pipes at the outlet end, the other between the two northern pipes 15 m from the inlet end of the pipe (Figure 3.1). This ensured that each profile was located at least 2.5 m from the closest air duct, minimizing the possibility of influence from the system. Once the thermocouples were positioned, the lead wires from each measuring station were threaded into 35.7 mm polyethylene tubing which extended the entire length of the 30 m duct. This provided a means of consolidating the wires to facilitate backfilling. A design change during installation required relocating the 75 wires from each of the two middle air ducts away from the center aisle toward the outside aisles to permit access for heavy equipment during backfilling. These wires were terminated in two junction boxes and extension wires connected to the the instrument shed.

3.3 MONITORING SYSTEM

All thermocouples were connected to a 330 channel Optilog datalogger which was housed in a heated, insulated instrument shed.

The datalogger was attached to a Corona PC which was programmed to automatically read and record the temperature at each location. Monitoring was originally scheduled for 06 00 h, 12 00 h, 18 00 h, and 24 00 h however, this was revised to 01 00 h, 04 00 h, 07 00 h, 10 00 h, 13 00 h, 16 00 h, 19 00 h and 22 00 h once the system was operational. During each data collection cycle the output from every thermocouple was sensed four times. Temperatures were averaged and stored on disk using the Corona PC. A difference of 2°C between any of the four readings was arbitrarily defined as "bad" data. "Bad" data were placed in a separate file for later manual inspection.

3.4 AIRFLOW MEASUREMENT

All four ducts exited into a common plywood header. A 1.1 KW fan drew air through all four pipes simultaneously. A series of baffles inside the header allowed for individual control of the airflow in each pipe. Airflows were balance so that one 150mm diameter pipe (Pipe Four) and one 250 mm diameter pipe (Pipe One) had a volumetric flow rate of 0.05 m^3/s. The remaining 150 mm diameter pipe (Pipe Two) was adjusted to a flow rate of 0.15 m^3/s and the remaining 250 mm pipe (Pipe Three) was adjusted to 0.10 m^3/s. Airflows were initially balanced using a velometer and later verified and occasionally checked using a hot wire anemometer.

3.5 SOIL PROPERTIES

The earth-to-air heat exchanger air ducts were installed in Red River Gumbo clay. Difficulties during construction caused soil temperatures and moisture contents to deviate from undisturbed values. Initial moisture contents were determined from soil samples taken during pipe installation. The soil thermal conductivity and bulk density were determined experimentally by Janzen (1985). Thermal diffusivity was calculated by Lei (1985) according to the method outlined by Scott et al. (1965). Table 3.1 provides a summary of the soil properties at the Glenlea site.

TABLE 3.1

Summary of soil properties at Glenlea Research Station

AVERAGE INITIAL SOIL MOISTURE CONTENT (%)		
Surface	23%	
1.0 m below surface	28%	
2.0 m below surface	27%	
3.0 m below surface	30%	
4.0 m below surface	34%	
BULK DENSITY	1304 kg/m^3	(Janzen, 1985)
SOIL THERMAL CONDUCTIVITY (20% mc)	0.29-0.30 W/m$^2 \cdot$°C	(Janzen, 1985)
SOIL THERMAL DIFFUSIVITY	0.0234 m^2/day	(Lei, 1985)

Chapter IV

RESULTS AND DISCUSSION

Data was collected from December 19, 1984 to February 20, 1987. Continuous operation was interrupted occasionally due to power failures at the site. Disk errors resulted in further data loss. Two major breaks in data collection occured from June 19, 1986 to July 18, 1986 and from October 5, 1986 to November 13, 1986. In both instances the recording equipment was damaged by a power surge believed to have been caused by lightning. Equipment repair was hindered by a difficulty in finding replacement boards for the data logger. Once new boards were installed, extensive testing and adjustments were required before an acceptable level of confidence was regained.

Throughout the testing period the high water table at the test location created problems with the operation of the system. Pipe Two, 150 mm diameter and 0.15 m^3/s airflow rate, failed to drain properly from the onset. Due to its location near the fan housing and instrument shed, and due to the close proximity of 150 thermocouple wires, excavating to rectify the problem was determined to be impractical. Several methods of pumping the water out of the pipe were investigated but were found to be impractical due to the small diameter of the pipe and the apparently rapid rate at which the water was entering. Finally, Pipe Two was abandoned. The remaining three pipes were checked frequently for water, and were pumped out when necessary. Despite pumping, Pipe One, 250 mm diameter and 0.05 m^3/s, had water in it for much of the winter of 1985-1986, and therefore produced no useful data during that period.

Data collected during the first year of operation will be presented in detail to illustrate the system performance since there were only brief periods of interruption during this time. Periods of extreme temperatures throughout the remainder of the testing period will be included to complement the observations of 1985.

4.1 BACKGROUND SOIL TEMPERATURE PROFILES

Of the two background soil temperature profiles one was located directly beneath the plywood header, midway between Pipe Two and Pipe Three, and the second, midway between Pipe One and Pipe Two and halfway along the length of the pipe. Weekly means were calculated for each datum point and the results of the two profiles were averaged to produce one set of background soil temperatures, (Figure 4.1).

Soil temperatures in December 1984 are high, ranging from 6.0 °C at 0.5 m below grade to 15°C at 3.5 m below grade. At 4.0 m below grade the temperature drops to 13 °C. The high soil temperatures 3.0 to 3.5 m below grade were believed to be caused by the exposure to hot summer conditions during excavation. Excavation influenced temperatures at all levels to some degree. By the end of 1985, soil temperatures were 3°C at 0.5 m, 9°C at 4.0 m, and 8°C at 3.0 m below grade.

As evident in Figure 4.1, a time lag exists between annual peak ambient air temperatures and annual peak soil temperatures. Since

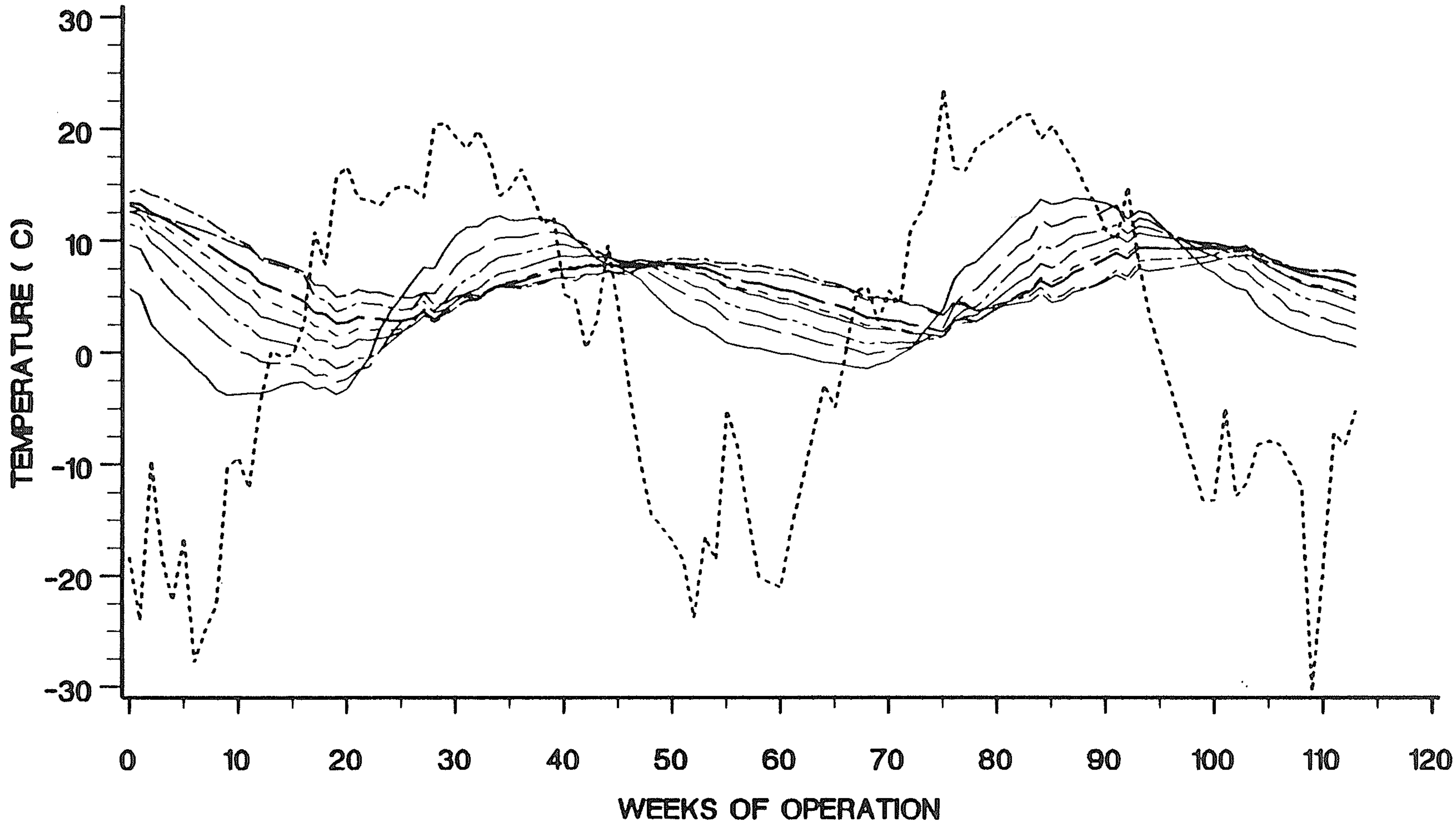

Figure 4.1: Average background soil temperature profile (weekly means). December 19, 1984-February 20, 1987. Depth below grade: (———) 0.5 m, (— — —) 1.0 m, (—·—·—) 1.5 m, (-———-) 2.0 m, (- - - -) 2.5 m, (——— ———) 3.0 m, (——-——-) 3.5 m, (——·——) 4.0 m. Ambient Air (- - - - - -).

temperatures near the surface are more influenced by climatic conditions the time lag here is minimal. The minimum temperature reached at the 0.5 m depth ranges from -4.0°C to -1.5°C and tends to occur in March. The maximum soil temperature at 0.5 m ranges from 12 °C to 13 °C and occurs in August. As depth increases, the time lag also increases. At 3.0 m below grade the soil reaches a minimum temperature of 2°C from the end of April to the middle of May, and a maximum temperature of 9°C around the end of October and the beginning of November. The soil temperature 3.0 m below grade, which is the depth the pipes are buried at, will be referred to as the background or far field temperature from now on.

4.2 HEAT EXCHANGER PERFORMANCE

Performance of an earth-to-air heat exchanger is believed to be dependent on the ambient air temperature, pipe diameter, air flow rate, and the temperature of the soil surrounding the pipe. The soil temperature is in turn affected by the temperature of the air which has previously passed through the pipe.

Results are defined according to the nature of the temperature change which the air experiences as it moves through the duct. Three modes of operation exist, heating mode, cooling mode and transition mode. Heating mode occurs in winter months when ambient air temperatures are lower than the soil temperature surrounding the pipe causing the air to be warmed as it passes through the pipe. Alternately, cooling mode occurs during the summer when the ambient air is warmer than the surrounding soil and is therefore giving up heat as it moves through the pipe. The transition mode occurs in spring and fall when ambient air temperatures and soil temperatures are nearly equal. For each mode, the performance is rated according to

1. actual temperature difference between inlet and outlet air,
2. theoretical exchanger efficiency, (Scott et al. 1965 and Baxter 1986) the ratio of the temperature difference between the inlet and outlet air to the maximum temperature difference if the inlet air were warmed/cooled to far field temperature at the depth of burial, and
3. percent temperature change which is the ratio of the temperature difference from the inlet to a given point along the pipe to the total temperature difference from the inlet to the outlet end.

Next, the overall performance of each pipe over the total testing period is examined. A bar graph is presented to show the expected temperature change in each pipe for inlet temperatures ranging from -35 to 30°C. Finally, plots are presented to demonstrate the percent of the total temperature change as a function of the distance along the pipe.

4.2.1 AIR TEMPERATURE CHANGE

4.2.1.1 HEATING MODE

The winter heating performance of all three pipes for January, February, November and December, 1985 are illustrated in this section. (Pipe One, 250 mm diameter with an airflow rate of 0.05 m^3/s, Pipe Three, 250 mm diameter with an airflow rate of 0.10 m^3/s, and Pipe Four, 150 mm diameter with an airflow rate of 0.05 m^3/s). The maximum possible temperature difference has been defined as the difference between the inlet air temperature and the background soil temperature at 3.0 m below grade. Inlet air temperature is taken as the average of two thermocouples measuring ambient air temperature at the site and therefore is assumed to be the same for each pipe. Outlet air temperature is the temperature recorded at the middle of a pipe, 29.0 m from the inlet end. This temperature was chosen over the temperature recorded at the center of the duct as it exited into the header because of the temperature change which was detected as the air moved up the uninsulated pipe to the header. All temperatures are presented as daily means, calculated from the eight readings taken at 01 00 h, 04 00 h, 07 00 h, 10 00 h, 13 00 h, 16 00 h, 19 00 h, and 22 00 h.

Pipes One and Four, with equal air flow rates demonstrated similar temperature trends (Figure 4.2). Pipe Three, having a larger airflow, was more affected by fluctuations in ambient air temperatures. Also, with the higher airflow rate, a smaller amount of tempering was achieved as the air passed through the pipe. These observations were predictable and consistent throughout the monitoring period.

January and February, 1985 were typical of Manitoba winter weather, (Figure 4.2). The average daily temperature during this two month period ranged from -35°C to 0°C. Average daily outlet temperatures from Pipes One and Four varied from 0°C to -10°C with the low recorded the first week in February when ambient air temperatures were around -35°C. Pipe Three recorded average daily outlet temperatures ranging from -5°C to -20°C with the low of -20°C recorded at the beginning of February. As ambient air temperatures began to rise by the end of February, the temperature difference between inlet and outlet air temperatures for all three pipes decreased.

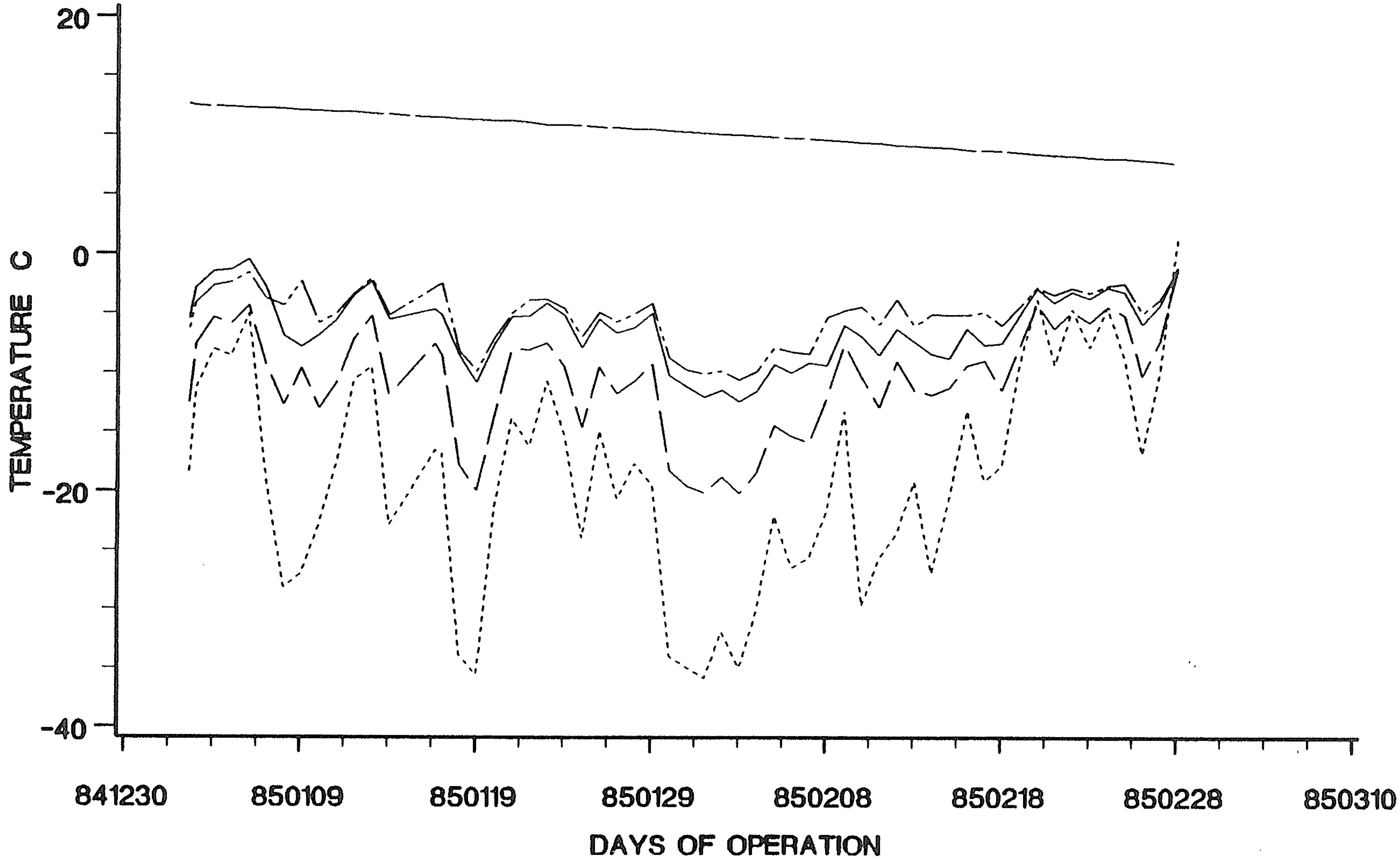

Figure 4.2: Ambient air, background soil, and outlet air temperatures (daily means). January 01, 1985-February 28, 1985. (- - - - - - - - -) ambient air, (——— — ——) background soil, (—————) pipe 1 outlet, (—— ——— —) pipe 3 outlet, (——— -- ———--) pipe 4 outlet air temperatures.

By November, 1985, ambient air temperatures once again dropped below -10°C (Figure 4.3). The daily average temperature fell 27°C from 6°C at the beginning of the month to -21°C by the 25th. At this time there was sufficient water in Pipe One to block the air flow through the pipe, therefore producing unusable data. Average outlet temperatures from Pipe Three stabilized to between -3°C to -7°C by the second week of November after dropping from 6°C in the first week. Average outlet temperatures from Pipe Four decreased from 6°C on the 5th to a level stabilizing between 0°C and -3°C from the 8th to the end of the month. Average daily ambient air temperatures in December, 1985, fluctuated between -5°C and -29°C. During this period the outlet air from Pipe Three fluctuated from -4°C to -14°C while the outlet air from Pipe Four varied from -1°C to -8°C. These temperatures are in the same range as the outlet temperatures of the previous winter, or perhaps slightly higher, possibly due to the somewhat warmer ambient air temperatures.

Two typical weeks of winter operation were considered. The first week, from January 27 to February 3, 1985 shows the far field temperature at the 3.0 m depth, constant at 10°C and the ambient air temperature ranging from a high of -10°C at 10 00 h on January 28, 1985 to a low of -40°C at 04 00 h on February 1, (Figure 4.4). This represents a maximum temperature difference between ambient and far field temperatures of 50°C. Outlet air temperatures from Pipes One and Four were fairly steady at -11°C and -9°C respectively after peaking at -3.5°C and -2°C at 01 00 h on January 29. Pipe Three reached a high of -6°C at the same time and dropped to become steady at around -19°C.

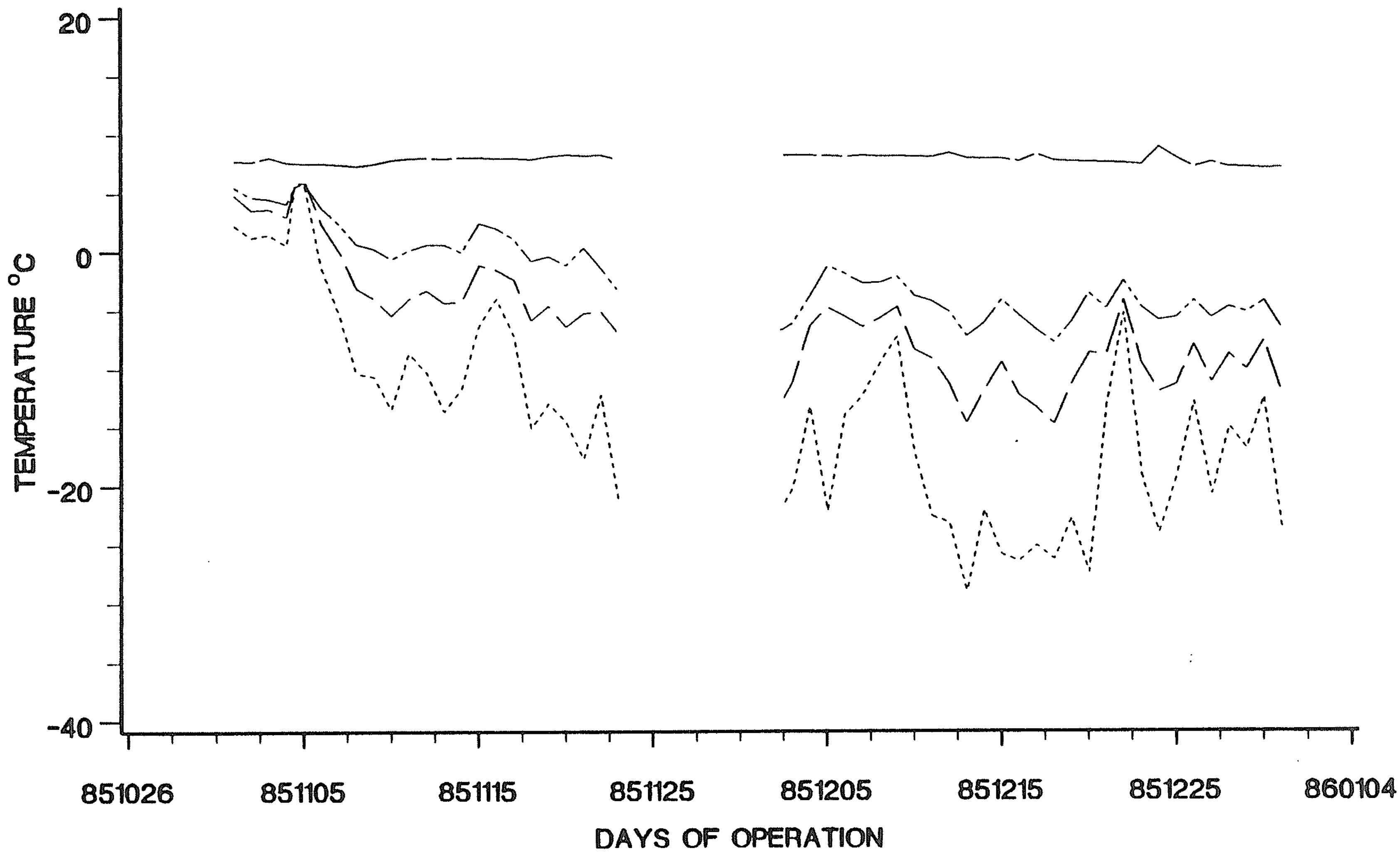

Figure 4.3: Ambient air, background soil, and outlet air temperatures (daily means). November 01, 1985-December 31, 1985. (- - - - - - - - -) ambient air, (——— – ———) background soil, (—— —— ——) Pipe 3 outlet, (—— · · —— · ·) Pipe 4 outlet air temperatues.

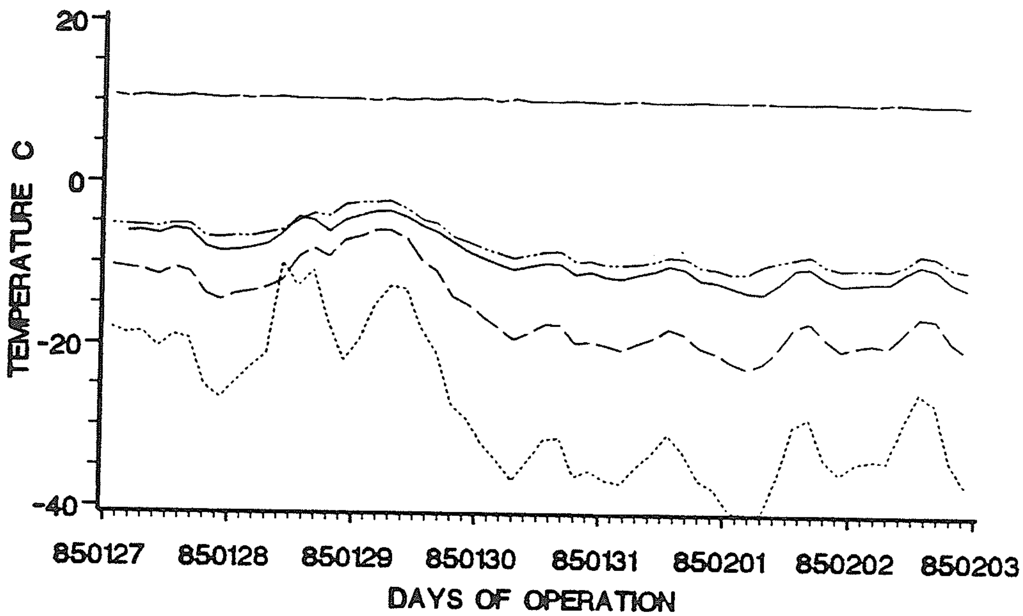

Figure 4.4: Ambient air, background soil, and outlet air temperatures (daily means). January 27, 1985-February 3, 1985. (-----------) ambient air, (—— — ——) background soil, (————————) pipe 1 outlet, (— — — — — —) pipe 3 outlet, (——-- ——- — —) pipe 4 outlet air temperatues.

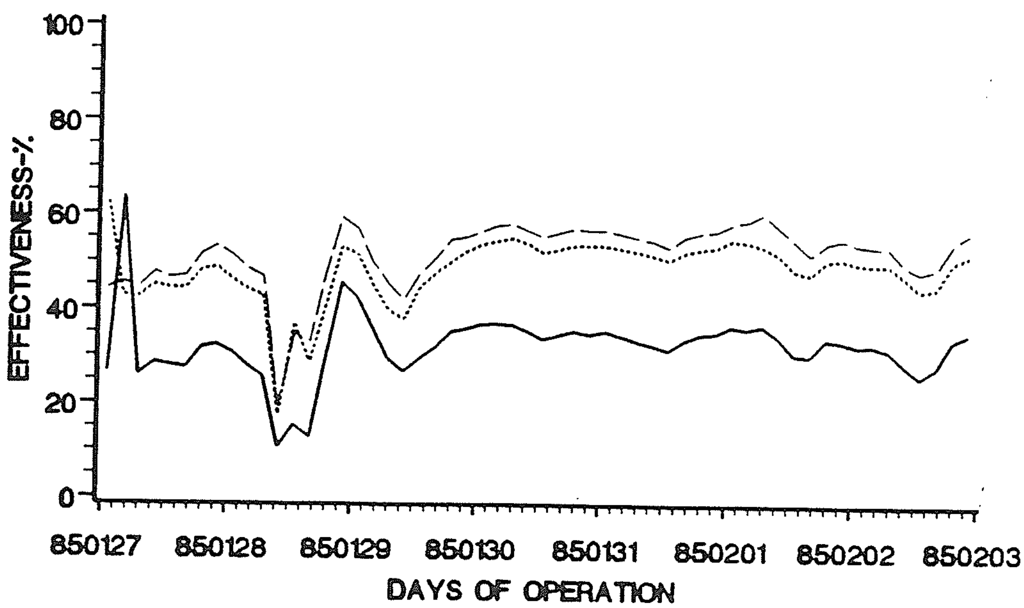

Figure 4.5: Exchanger efficiency for January 27, 1985-February 3, 1985. (----------) pipe 1, (————————) pipe 3, (— — — — — —) pipe 4.

The exchanger efficiency, as defined in Sec. 4.2, for this week of data is plotted in Figure 4.5. The average efficiencies, also referred to as effectiveness, were 52%, 35%, and 57% for Pipes One, Three, and Four respectively. The lower efficiency for Pipe Three was a result of its higher airflow rate and therefore lower temperature change. The drop in efficiency for all three pipes on January 28 was a result of the short period of warmer temperatures. Ambient air at this time increased from -30°C to -10°C while the outlet air temperatures remained relatively constant resulting in a very small temperature difference between the inlet and outlet air for each pipe. Since the efficiency is the ratio of this temperature difference to the temperature difference between the inlet air and the far field temperature, the smaller the temperature change through the pipe, the lower the efficiency.

The percent temperature change has been defined as the ratio of the temperature difference between the inlet air and the air at any point along the duct to the total temperature difference between the inlet and outlet air. The percent temperature change for each duct is shown in Figure 4.6 (a to c). For Pipe One, 85% of the total temperature change occurred in the first 20 m of pipe length. Similarly, 80% of the total temperature change was accomplished in the first 20 m of Pipes Three and Four. In Pipes One and Four most of the temperature change occurred along the first half of the pipe length whereas the temperature change was more evenly distributed along the length of Pipe Three. Eighty percent of the temperature change occurred in the first 15 m of Pipe One, while 77% of the total temperature change occurred in the first 15 m of Pipe Four. However, in Pipe Three only 65% of the total change

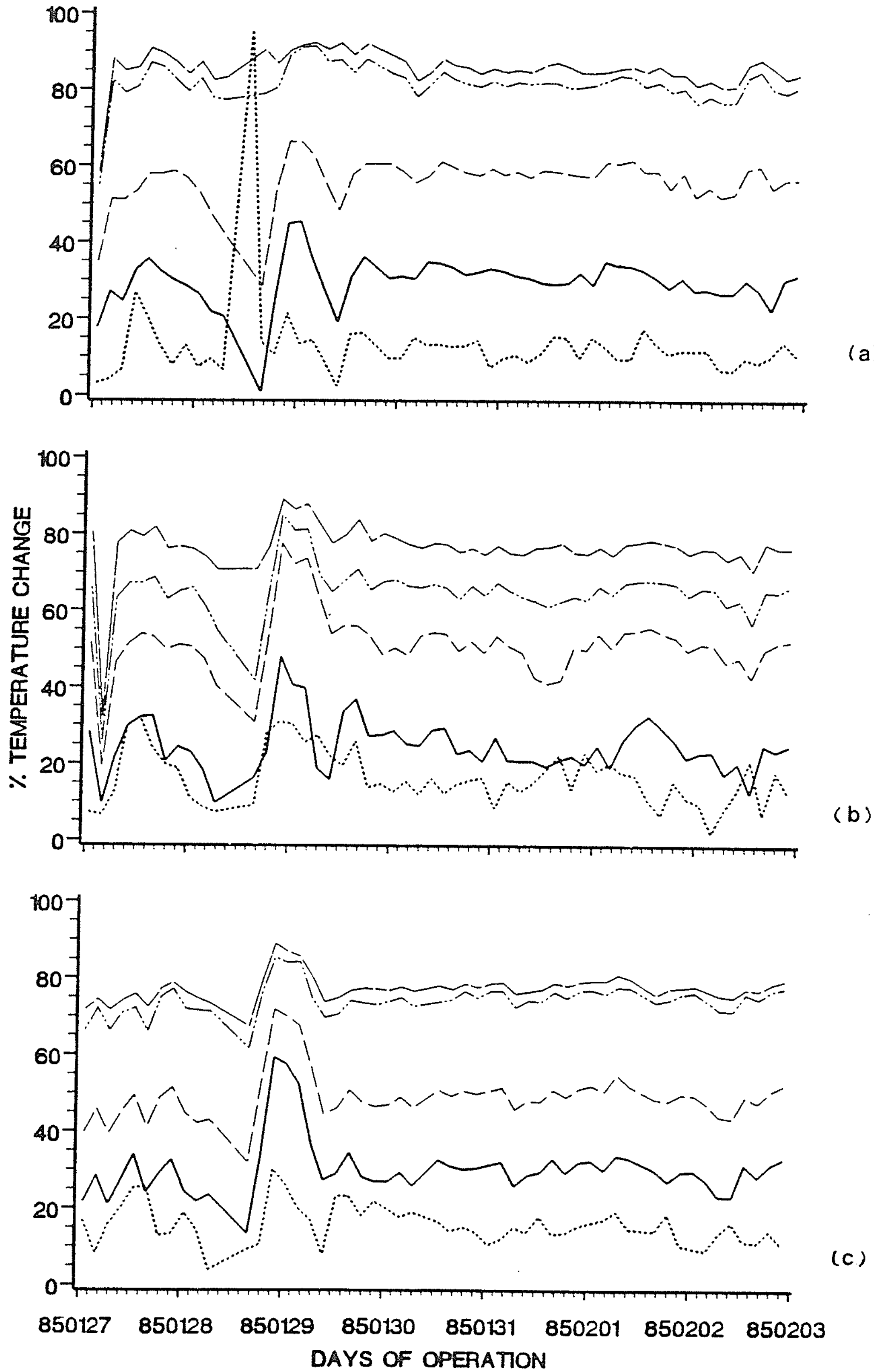

Figure 4.6: Percent temperature change along each pipe. January 27, 1985 - February 3, 1985. Distance along pipe: (---------) 1.0 m, (———) 5.0 m, (— — —) 10.0 m, (-- -- —— --) 15.0 m, (—— - ——) 20.0 m.
(a) Pipe One, (b) Pipe Three, (c) Pipe Four.

occurred in the first 15 m. Since only 20% to 23% of the total temperature change occurs in the last 15 m of length for Pipes One and Four, it may be feasible to use 15 m pipes and accept a slightly reduced performance. This idea would require further investigation.

The second week of winter data are for the week from December 22, 1985 to December 29,1985 (Figure 4.7). The far field temperature is nearly constant at 8°C. This is a drop of 2°C from January 1985 but, as was mentioned previously, the soil temperatures early in 1985 were still quite warm from being exposed to summer conditions during construction. Through the week the ambient air reached a high of 1°C at 04 00 h on December 22 and a low of -28°C at 01 00 h on January 24, 1985. Air flow through Pipe One was blocked by water through this period. Pipes Three and Four maintained relatively constant outlet temperatures at -10°C and -5°C respectively.

The efficiencies of Pipes Three and Four over this week were 35% and 55% respectively (Figure 4.8). These values compare very favorably with the values mentioned earlier. Once again the drop in overall efficiency is evident when the temperature difference between inlet and outlet air is decreased. Figures 4.9a to 4.9c show the percent temperature change at each monitoring station along the pipe. An average of 90% and 80% of the total temperature change is occurring during the first 20 m of Pipes Three and Four respectively. An interesting point to mention is that very little temperature change appears to occur between 15 m and 20 m along Pipe Four.

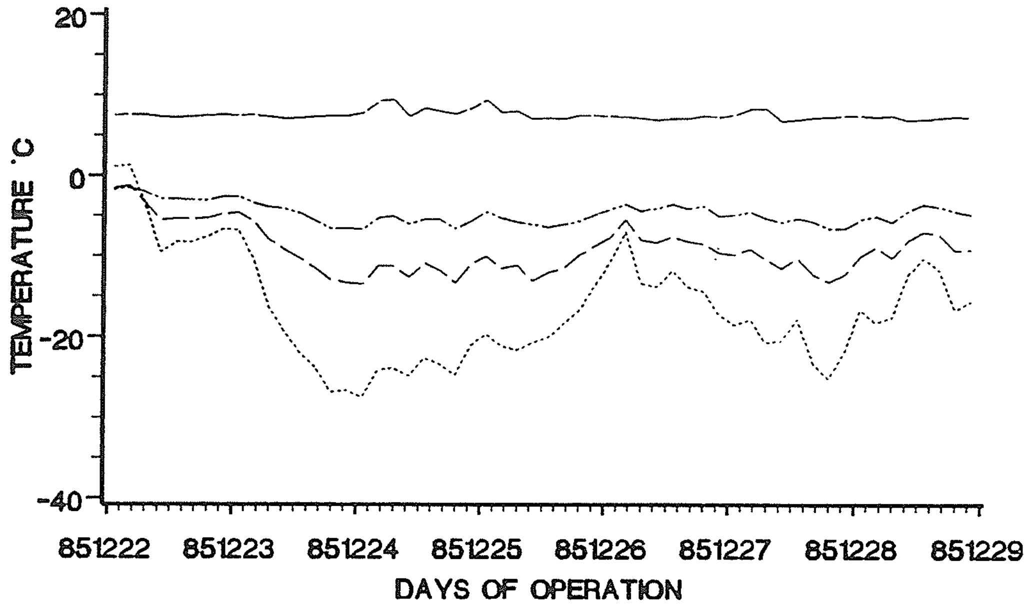

Figure 4.7: Ambient air, background soil, and outlet air temperatures (daily means). December 22, 1985-December 29, 1985. (----------) ambient air, (——— — ——) background soil, (— — — —) Pipe 3 outlet, (——--——--) Pipe 4 outlet air temperatues.

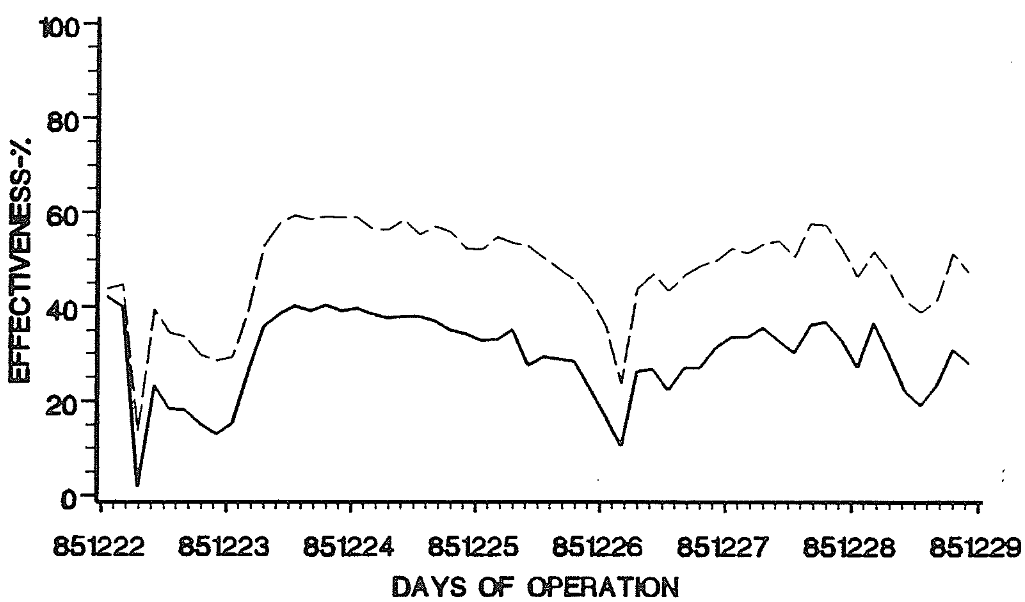

Figure 4.8: Exchanger efficiency for December 22,1985-December 29,1985. (————————) Pipe 3, (— — — —) Pipe 4.

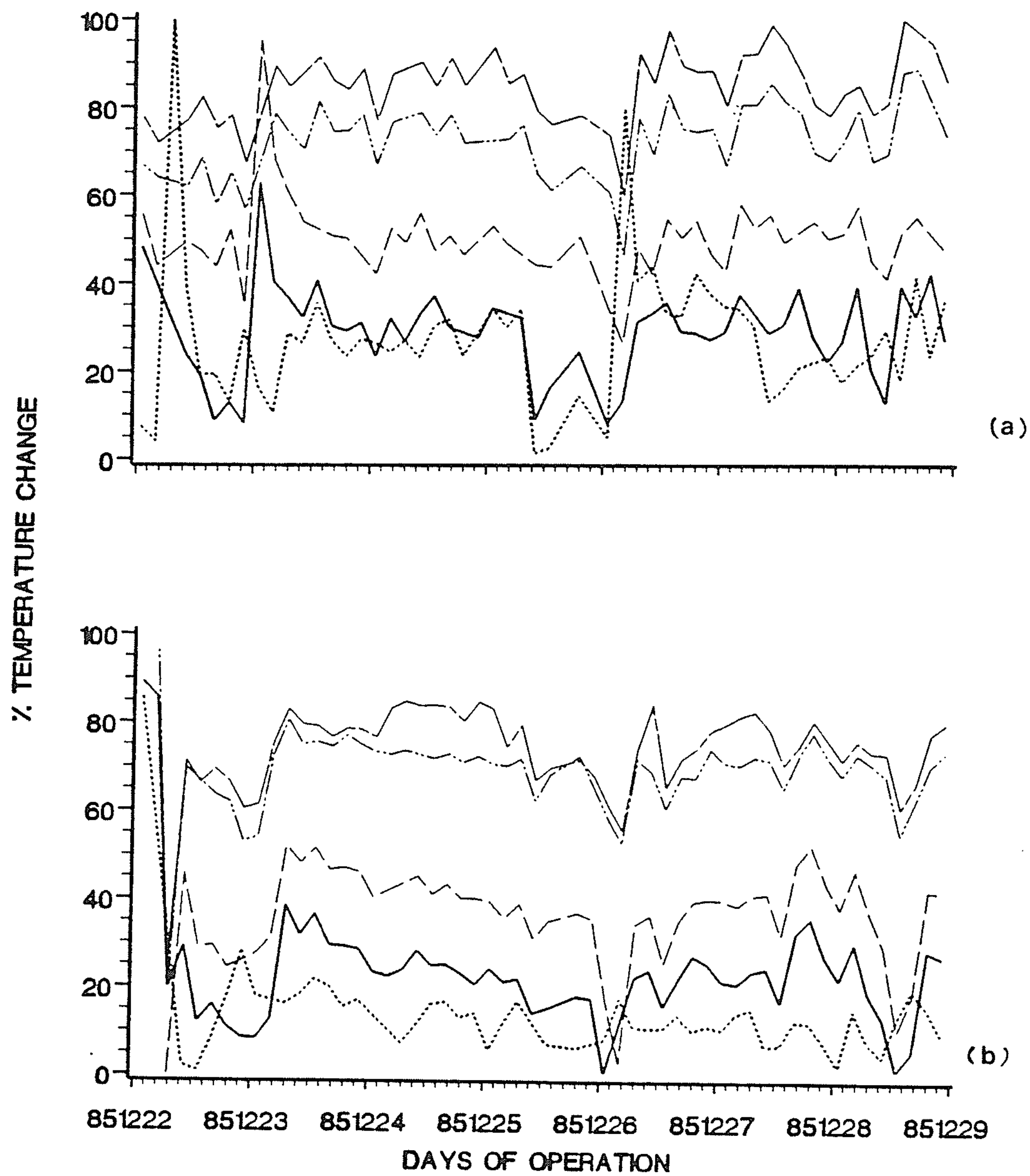

Figure 4.9: Percent temperature change along each pipe. December 22, 1985-December 29, 1985. Distance along pipe: (---------) 1.0 m, (————) 5.0 m, (— — — —) 10.0 m, (——-- ——-) 15.0 m, (—— - ——) 20.0 m. (a) Pipe Three, (b) Pipe Four.

4.2.1.2 COOLING MODE

The next section shows cooling performance of Pipes One, Three, and Four through June, July, and August, 1985. The cooling mode is less defined than the warming mode because of the occurrence of cool days and nights where ambient air temperatures may be close to or lower than the temperature of the surrounding soil. When this condition occurs the system is operating in a transition phase which will be discussed in the next section.

During this period, the daily average ambient temperature ranged from 7°C at the beginning of June to 24°C on the 8th of July and the 2nd of August (Figure 4.10). The weather was cool and wet with few days of extreme temperatures. As was observed in the heating mode, average daily outlet temperatures from Pipes One and Four followed very similar patterns. Outlet temperatures ranged from 5°C at the beginning of June to 14°C in August and from 6°C at the beginning of June to 15°C in August for Pipe One and Pipe Four respectively. Average daily outlet temperatures for Pipe Three varied over this period from 7°C to 17°C. The temperature change between inlet and outlet air temperatures in the summer is generally lower than the temperature change during the warming mode due to the reduced temperature difference between the incoming air and the soil surrounding the exchanger pipe.

Figures 4.11 to 4.13 illustrate actual performance of Pipes One, Three, and Four during the week of July 28, 1985 to August 3, 1985. Far field temperature, still not completely recovered from winter conditions, was steady at 4°C. Ambient air exhibited large diurnal fluctuations with a high of 32°C at 13 00 h July 31, 1985 and a low of

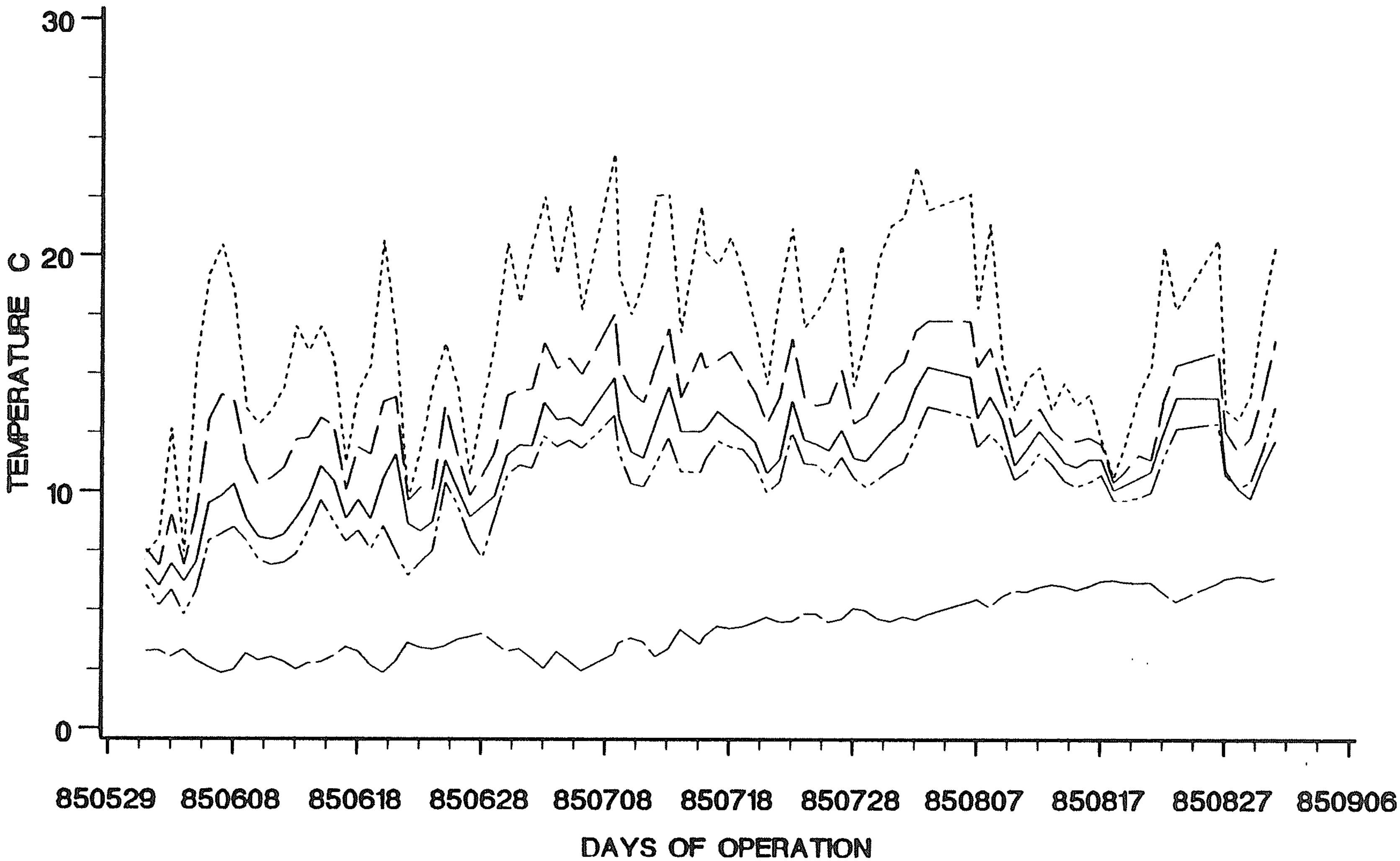

Figure 4.10: Ambient air, background soil, and outlet air temperatures (daily means). June 01, 1985-August 31, 1985. (- - - - - - - - -) ambient air, (——— - ———) background soil, (—————) pipe 1 outlet, (—— ———) pipe 3 outlet, (——- · -——- -) pipe 4 outlet air temperatues.

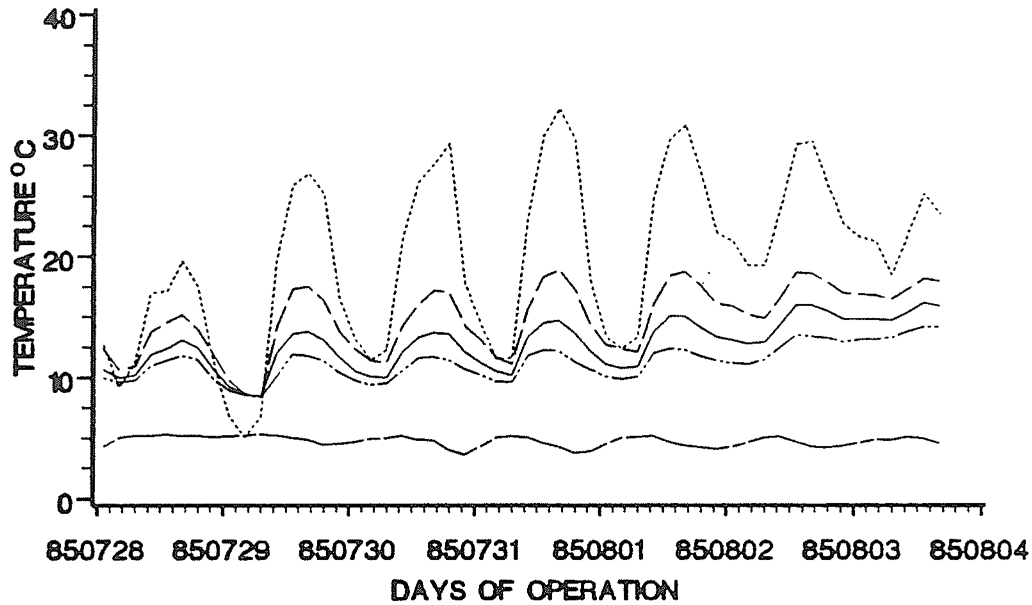

Figure 4.11: Ambient air, background soil, and outlet air temperatures (daily means). July 28,1985-August 3, 1985. (----------) ambient air, (———-——) background soil, (——————) pipe 1 outlet, (— — — — —) pipe 3 outlet, (—··——··—) pipe 4 outlet air temperatues.

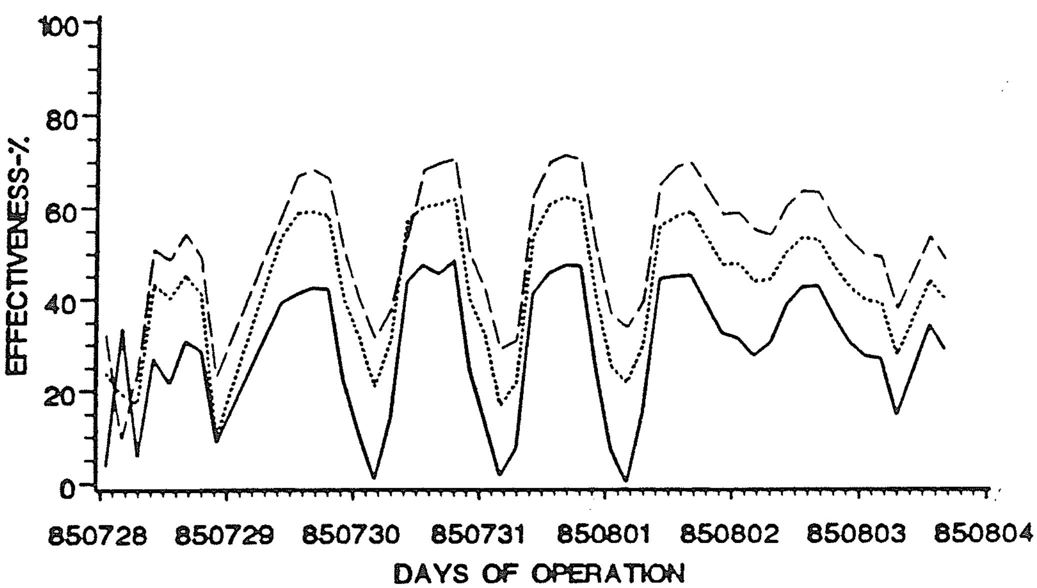

Figure 4.12: Exchanger efficiency for July 28, 1985-August 3, 1985. (-----------) pipe 1, (——————) pipe 3, (— — — — —) pipe 4.

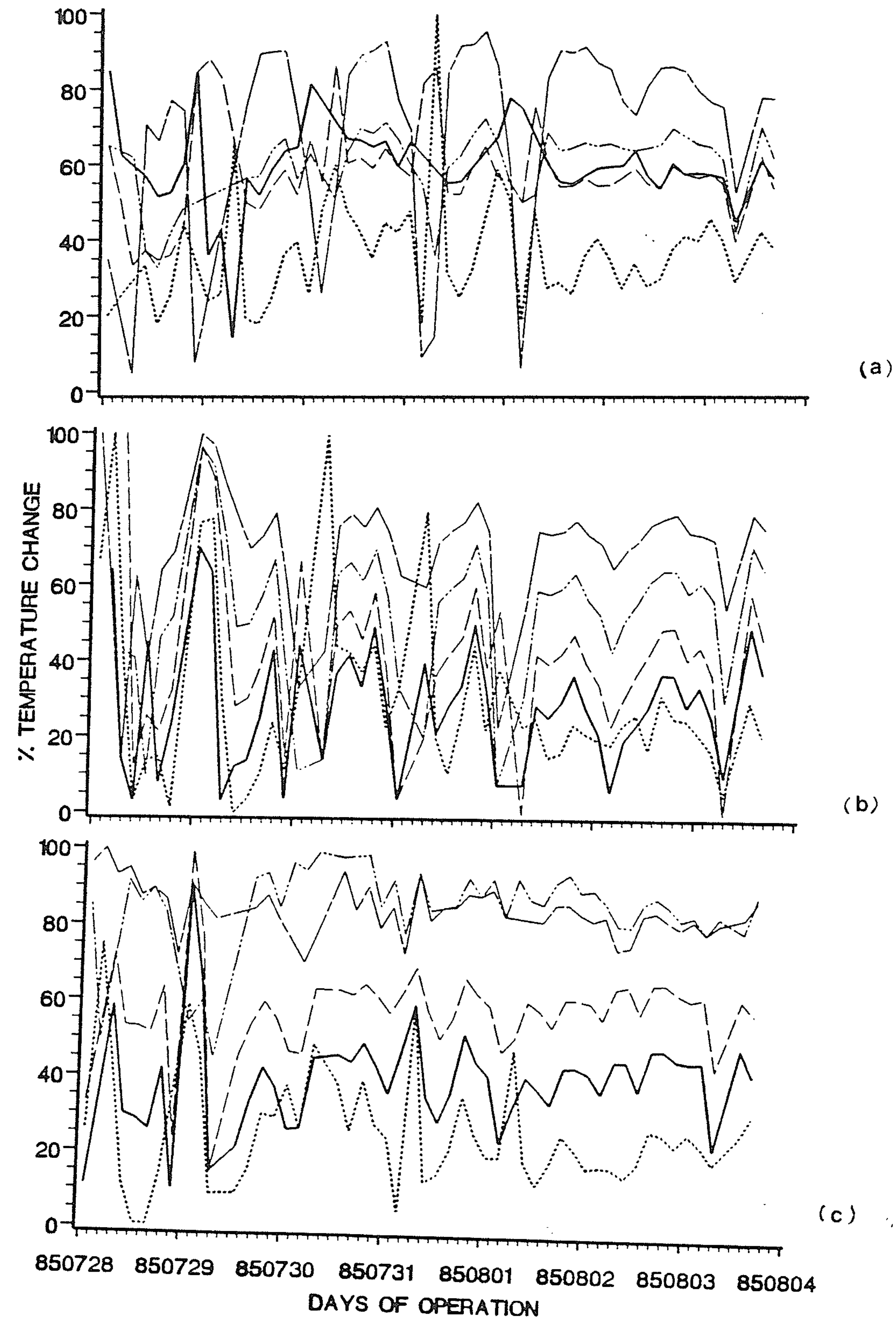

Figure 4.13: Percent temperature change along each pipe. July 28, 1985 - August 3, 1985. Distance along pipe: (--------) 1.0 m, (————) 5.0 m, (— — — —) 10.0 m, (—--—--) 15.0 m, (——————) 20.0 m.
(a) Pipe One, (b) Pipe Three, (c) Pipe Four.

4°C at 04 00 h July 29, 1985. Outlet temperatures ranged from 8.5°C at 07 00 h July 29 for all three pipes, to 16°C, 18.5°C, and 14°C for Pipes One, Three, and Four. The plot of overall efficiency clearly shows the effect of hot days and cool nights. Peak efficiencies of 60%, 45%, and 70% for the three pipes occurred when maximum ambient air temperatures were experienced. These efficiencies were reduced to 25%, 2%, and 35% respectively when ambient air dropped to near 10°C at night. These low efficiencies are typical of what could be expected during the transition mode. The nighttime temperatures are too low to require any cooling and too high to adequately recharge the system. Operation of the system under these conditions provides little practical benefit.

The confusion of the lines in Figures 4.13, a to c, illustrate the problems associated with summer operation. Operation during hot summer days provides substantial cooling of the ambient air as has already been demonstrated. However, operation during cool nights, when the ambient air is at or near the temperature of the soil surrounding the pipes, results in little or no temperature change occurring between inlet and outlet air. Therefore, the percent temperature change occurring at any point along the pipe becomes very erratic. On average though, it can still be said that pipes One and Three achieve about 80% of the total temperature change occurring in the first 20 m of pipe. As the air moves through Pipe Four, nearly 90% of the total temperature change occurs in the first 20 m of pipe.

4.2.1.3 TRANSITION MODE

During spring and fall when ambient air temperatures often range between -10°C and 15°C, the temperature difference between the inlet air and the soil surrounding the pipe is small. Because of this, very little heat transfer occurs and there is minimal benefit in drawing air through the ducts (Figures 4.14 and 4.15) However, since the ambient air at these temperatures is suitable for direct ventilation, the system could be shut down without affecting the operation of the building. This type of controlled shut down would enhance the overall operation of the heat exchanger. Heat stored in the soil during the summer would be preserved for use during extreme winter temperatures instead of being dissipated during the cool days of the fall. As well the cool soil temperatures which are achieved by the end of winter would be preserved during the spring to enhance cooling through the summer months.

Figures 4.16 to 4.18 simply demonstrate the difficulty of measuring performance when an earth-to-air heat exchanger is operated in transition mode. These figures represent actual data collected from March 31, 1985 to April 6, 1985. Far field temperature was constant throughout the week at about 5.5°C. Ambient air varied from -6.5°C at 04 00 h March 31 to 9.5°C at 16 00 h April 2 (Figure 4.16). Outlet air temperatures during this period varied from -1.3°C to 4.3°C from Pipe One, -3.7°C to 4.9°C from Pipe Three and -1.4°C to 3.4°C from Pipe Four. The maximum temperature change of 6.1°C occurred when ambient air at 9.5°C was cooled to 3.4°C while passing through Pipe Four. The additional energy required to draw the air through 30 m of pipe could hardly be justified when ambient air at 9.5°C is adequate for ventilation.

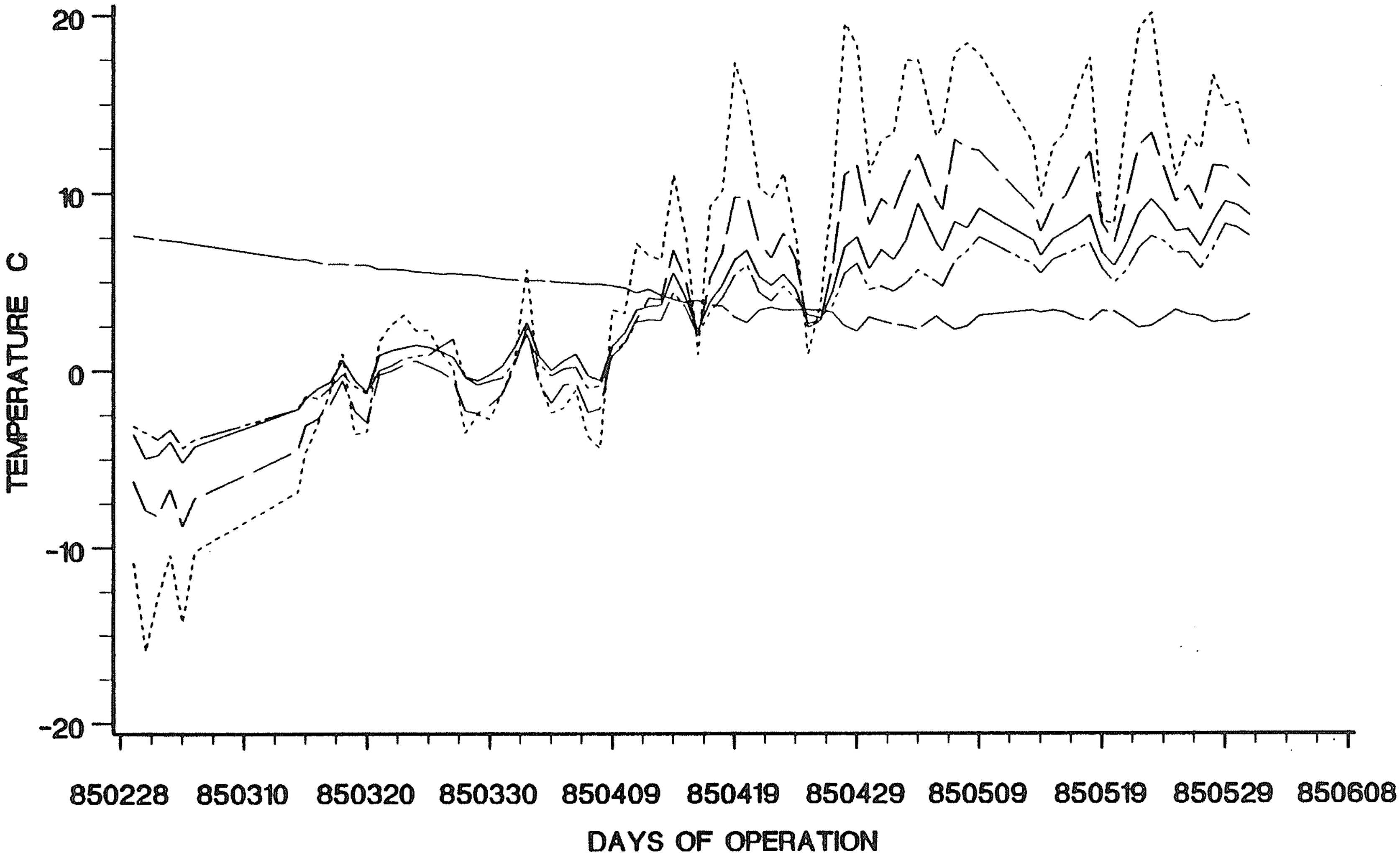

Figure 4.14: Ambient air, background soil, and outlet air temperatures (daily means). March 01, 1985-May 31, 1985. (- - - - - - - - -) ambient air, (—— - ——) background soil, (————) pipe 1 outlet, (—— —— —) pipe 3 outlet, (—- - —— - -) pipe 4 outlet air temperatues.

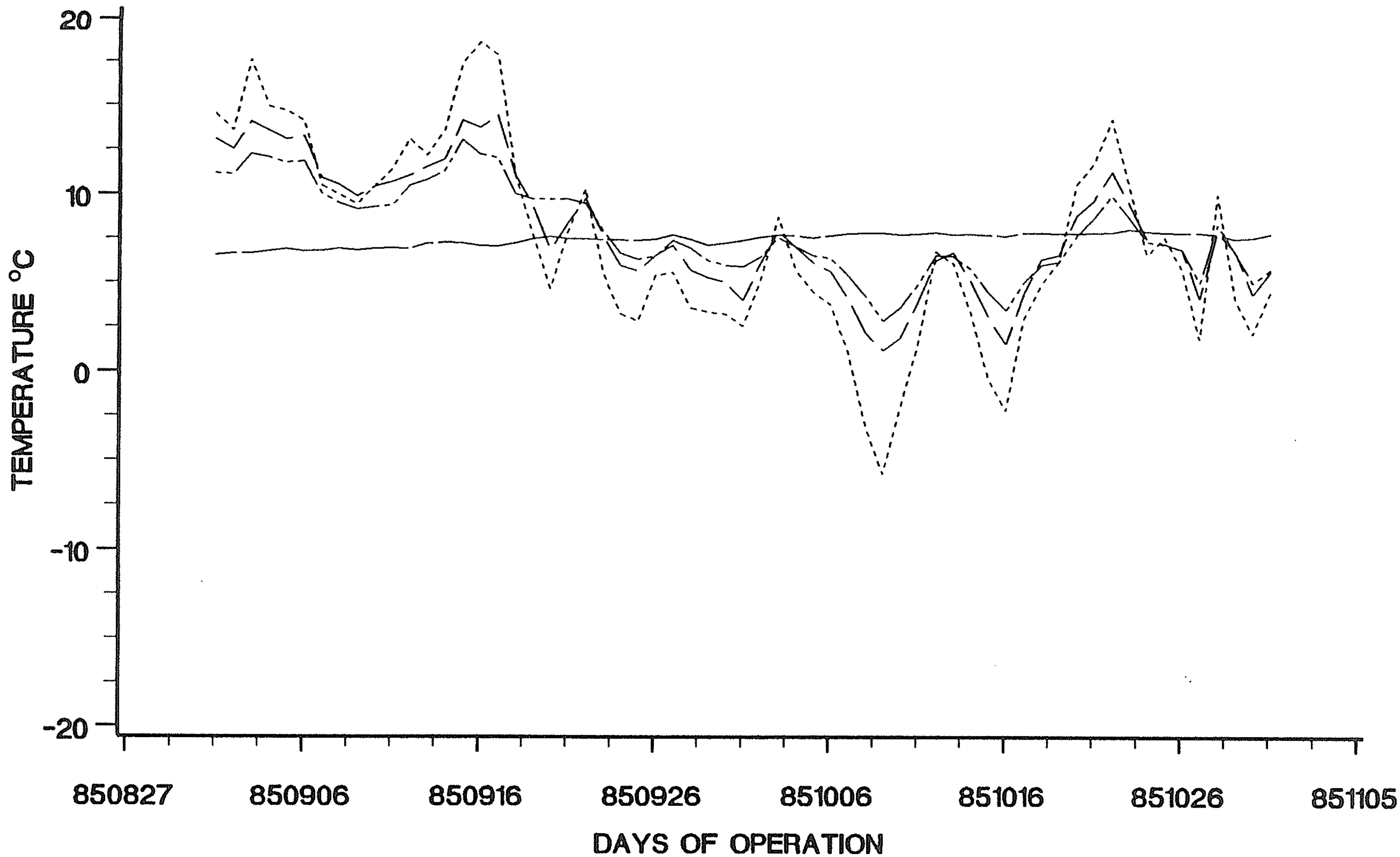

Figure 4.15: Ambient air, background soil, and outlet air temperatures (daily means). September 01, 1985-October 31, 1985. (- - - - - - -) ambient air, (——— - ———) background soil, (—— —— ——) Pipe 3 outlet, (——- - ———- - -) Pipe 4 outlet air temperatues.

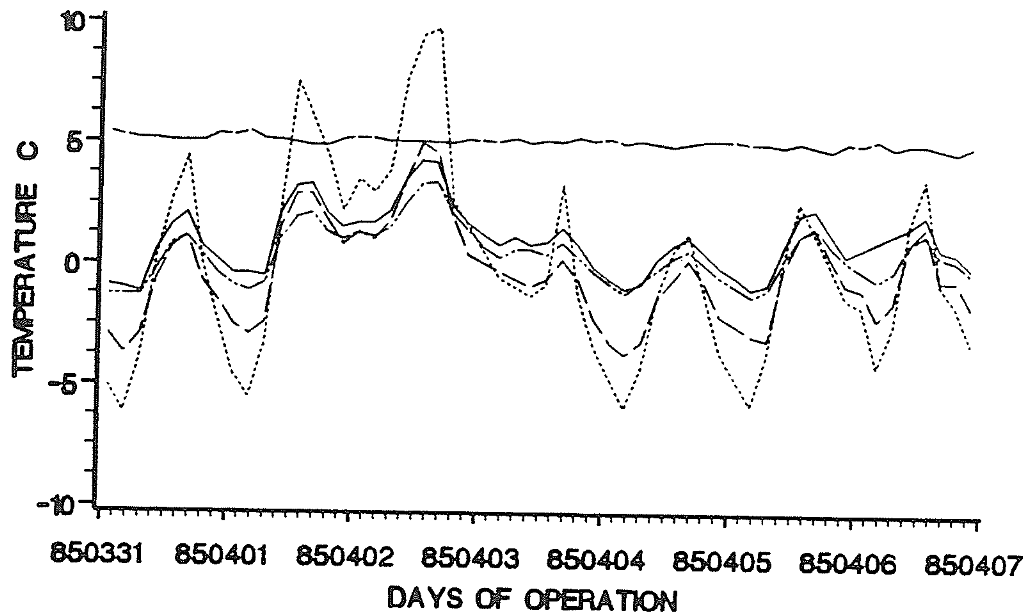

Figure 4.16: Ambient air, background soil, and outlet air temperatures (daily means). March 31, 1985-April 6, 1985.
(-----------) ambient air, (——— — ———) background soil,
(——————————) pipe 1 outlet, (— — — —) pipe 3 outlet,
(—-- ——— --—) pipe 4 outlet air temperatues.

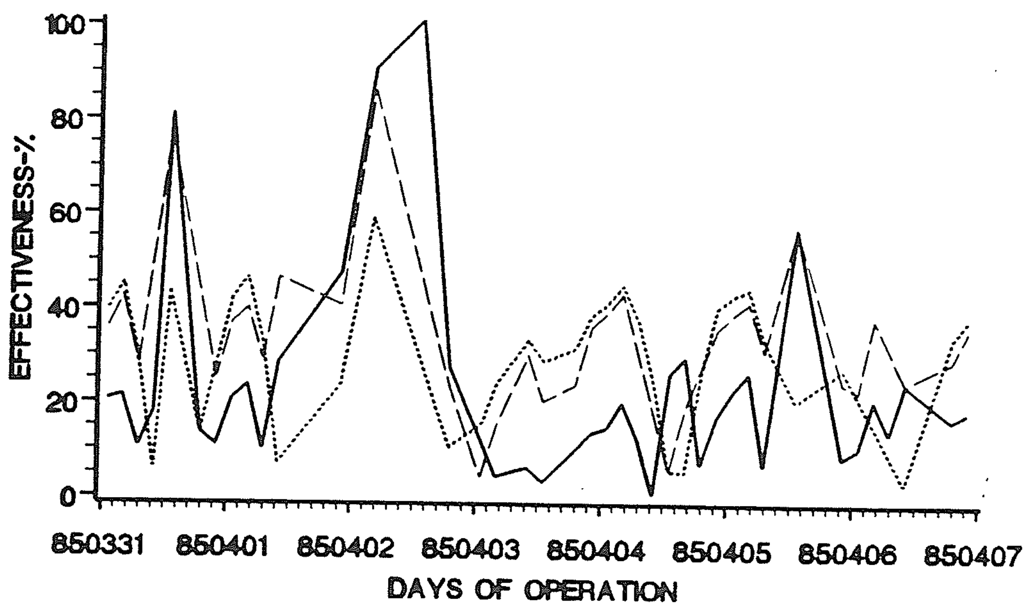

Figure 4.17: Exchanger efficiency for March 31, 1985-April 6, 1985.
(----------) pipe 1, (——————————) pipe 3,
(— — — — —) pipe 4.

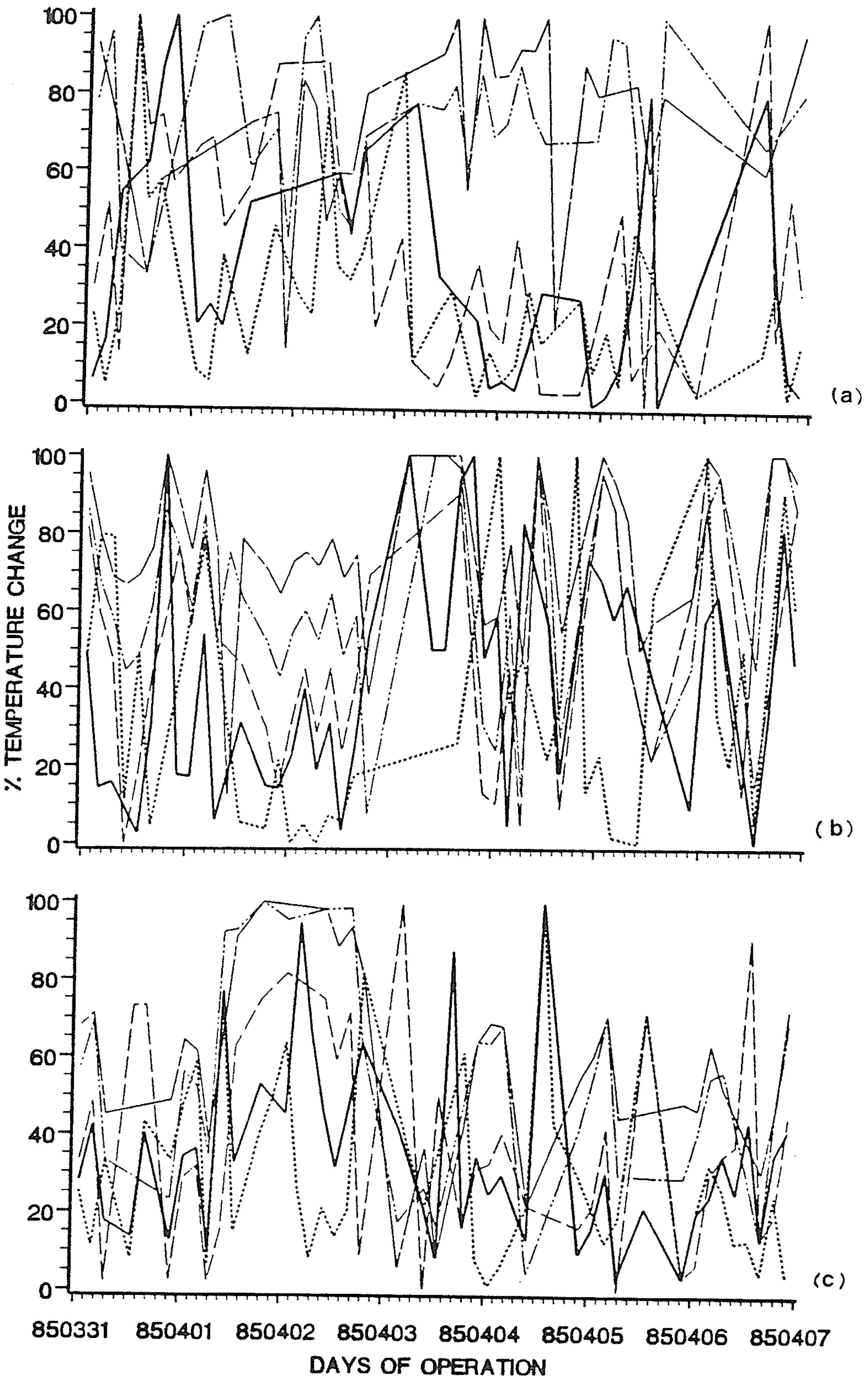

Figure 4.18: Percent temperature change along each pipe. March 31, 1985 - April 6, 1985. Distance along pipe: 1.0 m, 5.0 m, 10.0 m, 15.0 m, 20.0 m. (a) Pipe One, (b) Pipe Three, (c) Pipe Four.

As would be expected, overall efficiency for this period is quite low (Figure 4.17). Pipe Three reached an overall efficiency of 100% on April 2 since the outlet air was at far field temperature. However, this occurred when ambient air at 9.5°C was cooled to 4°C which can not be considered to be exceptional performance. It must be realized that at some point between the heating mode and cooling mode, outlet temperatures will occur at far field temperature due to system design. The importance of overall efficiency is in what can be expected under design, or worst case, conditions and not what happens in the transition mode.

The plots of percent temperature change along each pipe are a clutter of lines (Figure 4.17). Since the air entering the pipe is at nearly the same temperature as the soil surrounding the pipe, and a maximum temperature change of 6°C occurs over 30 m of pipe, the temperature change from point to point is very small. The absence of clearly defined levels of temperature change with distance along pipe further emphasizes the futility of operating the system under these conditions.

4.2.1.4 EXPECTED PERFORMANCE

Figures 4.19 and 4.20 can now be used to predict performance under various ambient temperatures. The bar graphs were created by averaging actual outlet temperatures recorded over the 26 month testing period at inlet temperature intervals of 5°C. The similarity of the effectiveness of Pipes One and Four, (airflow rate of 0.05 m^3/L), is demonstrated as well as the somewhat lower effectiveness of pipe Three, (airflow rate 0.10 m^3/s). It should be noted that the relative implied performance is

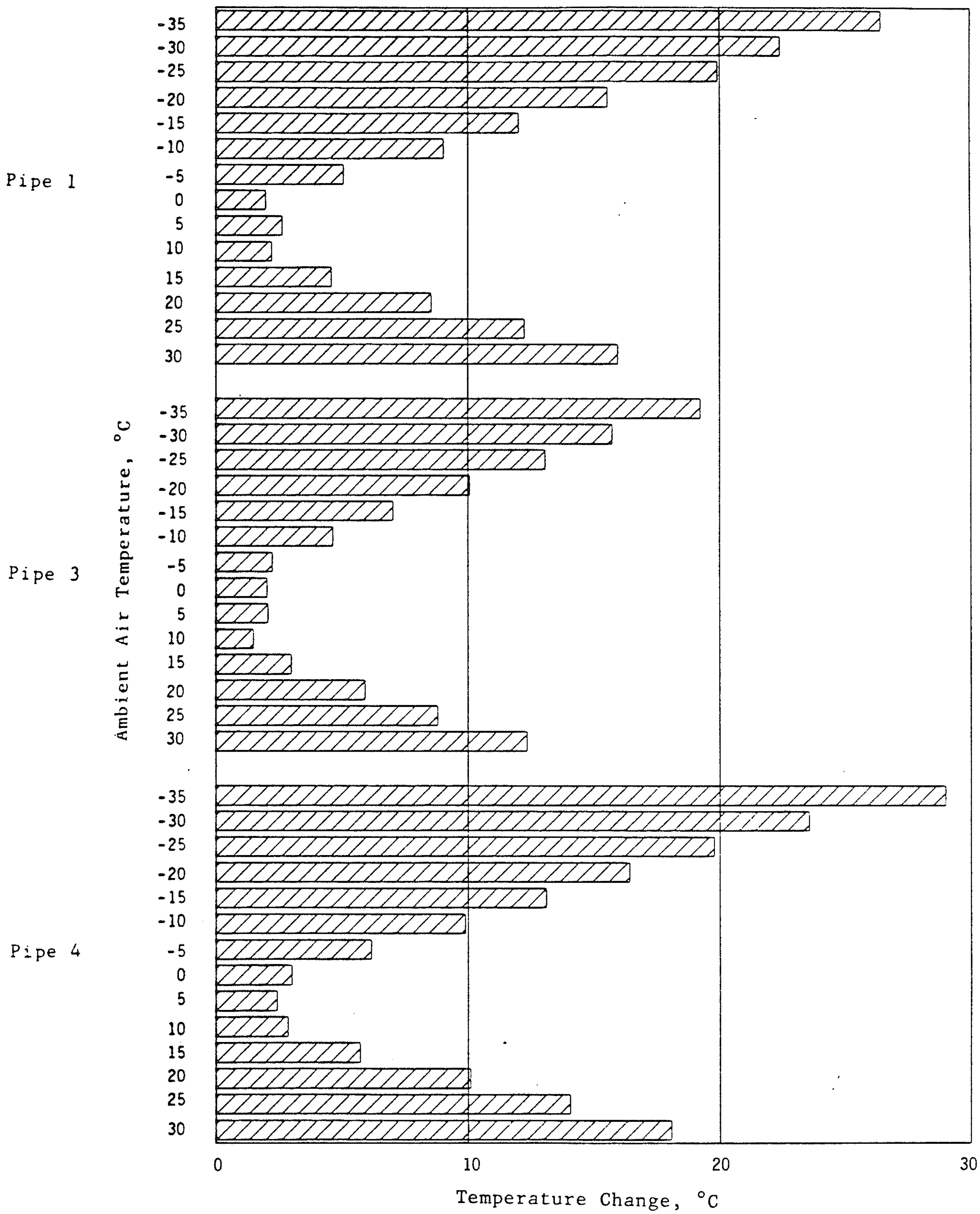

Figure 4.19: Potential temperature change through buried pipe versus ambient air temperature. Based on experimental data from December 19, 1985-February 20, 1987.

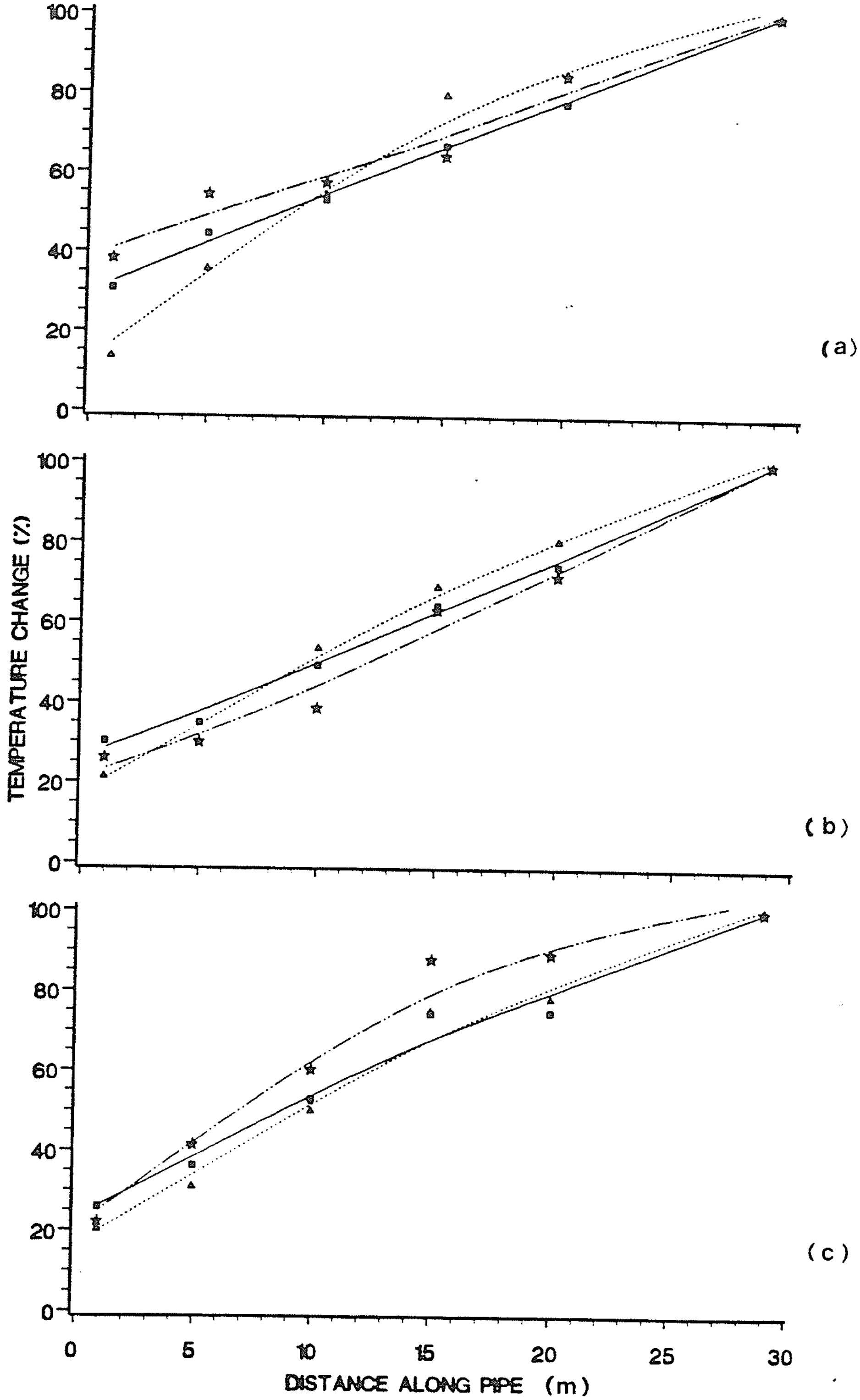

Figure 4.20: Average percent temperature change versus distance along pipe. Based on experimental data from December 19, 1985 - February 20, 1987. (—■———■—) average of all collected data, (--▲-------▲--) winter data only, and (—★·——--★---) summer data only.
(a) pipe One. (b) pipe Three. (c) pipe Four.

based on the on the definition of effectiveness. If the comparison had been made on the basis of total energy transfered, Pipe Three would be defined as having better performance than Pipes One and Four.

During heating mode, a maximum temperature change of 29°C is possible using a 150 mm air duct with an airflow rate of 0.05 m^3/s at an inlet temperature of -35°C. This would mean an outlet temperature of -6°C on average. With this inlet temperature, an average temperature change of 26 and 19°C can be expected from Pipes One and Three respectively. These should be used only as average values because of the many factors which will actually affect outlet temperatures. For example, during an extended cold period, the soil near the pipe will become cool and the temperature gradient between the incoming air and the soil will be reduce. Likewise, if the temperature happens to drop to -35°C at the beginning of the heating season when the soil temperatures are still high from the summer conditioning, the temperature gradient will between the soil and the incoming air will also be high. So, not only is the ambient air temperature an important factor in determinimg the expected temperature change, but the recent history of the ambient air must also be considered. Since this graph is based on average values, the accuracy is also affected by the number of observations which were used in the calculations. Naturally, there were more observations at -30°C than at -35°C. At a less severe ambient temperature of -20°C temperature changes of 15°C, 10°C, and 16°C are possible from Pipes One, Three and Four respectively. These correspond to outlet temperatures ranging from -10°C to -4°C. Therefore, any one of these pipe designs would be capable of tempering ambient air to a level suitable for ventilating livestock shelters.

During the summer season, a maximum temperature change of 18°C is possible from Pipe Four with an inlet temperature in the range of 30°C. This corresponds to an outlet temperature of 12°C which is within the comfort zone for agricultural structures. Pipes One and Three will produce temperature changes of 16°C and 12°C at an inlet temperature of 30°C. All three pipes studied will modify ambient air at 20°C by at least 10°C. Therefore, by integrating an earth-air system into a livestock shelter the risk of summer heat stress could be eliminated.

During transition mode, as was discussed earlier, operation of the earth-to-air heat exchanger will provide only limited benefits. From Figure 4.19, less than a 10°C and in most cases less than a 5°C temperature change can be expected in each duct at inlet temperatures between -10°C and 15°C. Since ambient air in this range is suitable for direct ventilation it is recommended that under these conditions the earth-air heat exchanger not be operated.

In Figure 4.20, the data collected throughout the testing period have been used to generate curves relating precent temperature change to distance along the pipe. Each graph represents one pipe. The three lines on each graph represent the the average of all data collected, the average of all data collected when the ambient air was below -10°C, and the average of all data collected when the ambient air was above 15°C. All three graphs are similar. For all pipes, it can be seen that the curves produced with spring and fall data included in the calculations, lie between the curves generated using only winter or only summer data. As could be expected the curves for Pipes One and Four are slightly more convex, indicating that the amount of temperature change occurring decreases as distance along pipe increases. These graphs would be useful in determinig the optimum pipe length in terms of temperature change and cost per unit length of pipe. An interesting point to make is that for pipe four, under all three conditions, little temperature change seems to occur between 15 and 20 m along the pipe. The cause for this is not known. Perhaps it has to do with the ground water level or the soil properties at that point.

4.2.2 HEAT MIGRATION THROUGH THE SOIL

Heat movement through the soil is assumed to occur in a radial direction from the surface of the air duct. As air temperature changes with movement along the duct, the temperature of the soil surrounding the duct at that point also changes. Three dimensional plots illustrate the movement of the temperature front through the soil. The plots were based on the monthly average of all readings recorded at each monitoring station. A grid was generated from these averages to create a temperature surface extending 1.0 m from the surface of the pipe for the total pipe length. The data collected from the soil surrounding Pipe Four will be discussed in detail in this section. Data from the other two pipes will be referred to illustrate contrasts between airflow rates and pipe diameters.

Figures 4.21 to 4.23 indicate the February, 1985 and July, 1985 behavior of each radial of Pipe Four, vertically up and down from the pipe axis, and horizontal from the pipe axis. The surface representing the soil temperatures vertically upward from the pipe surface reflect the effect from the ambient environment. For February 1985, this

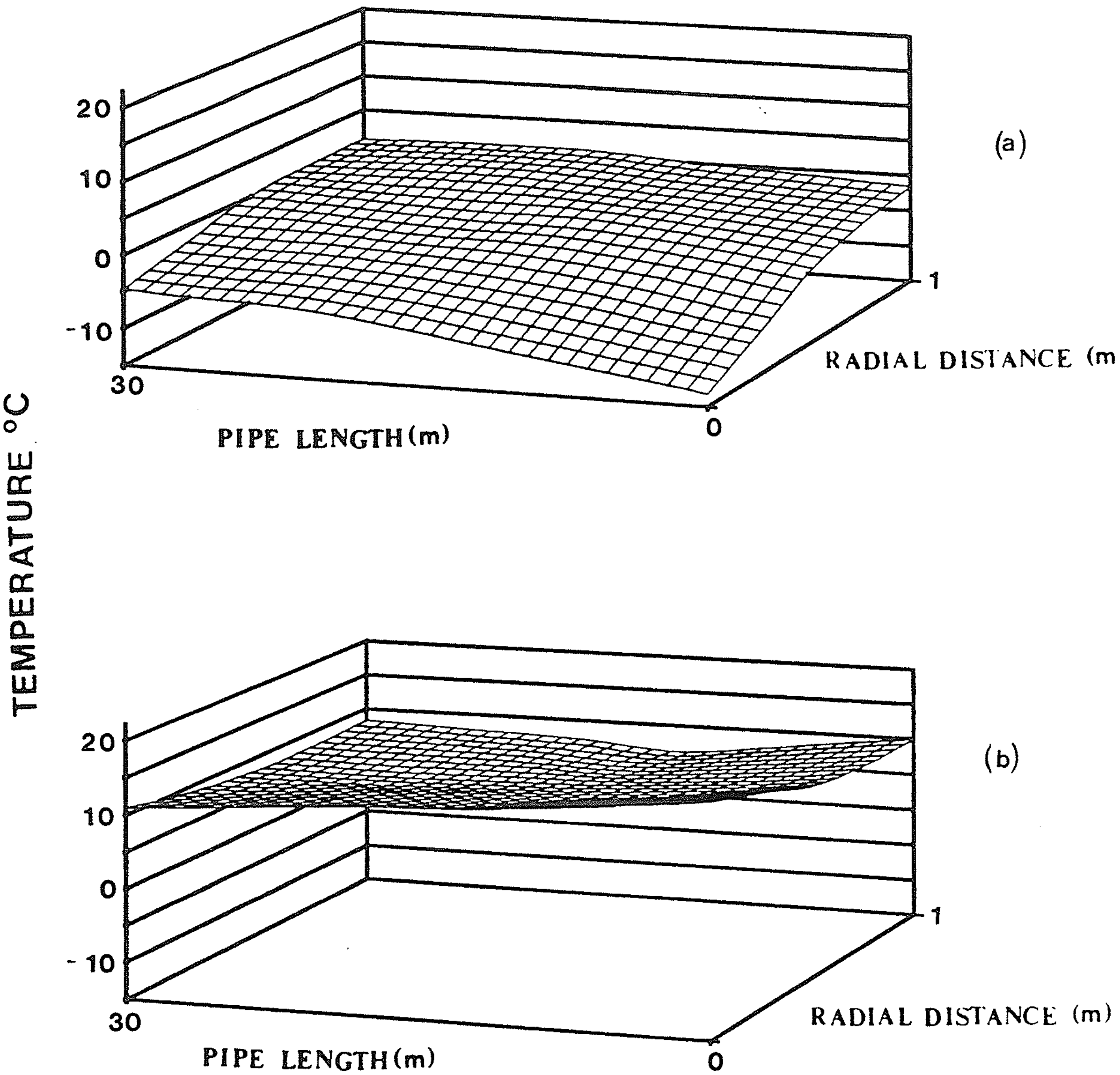

Figure 21: Soil temperature surfaces generated from thermocouples vertically up from surface of pipe Four. (a) February. (b) July.

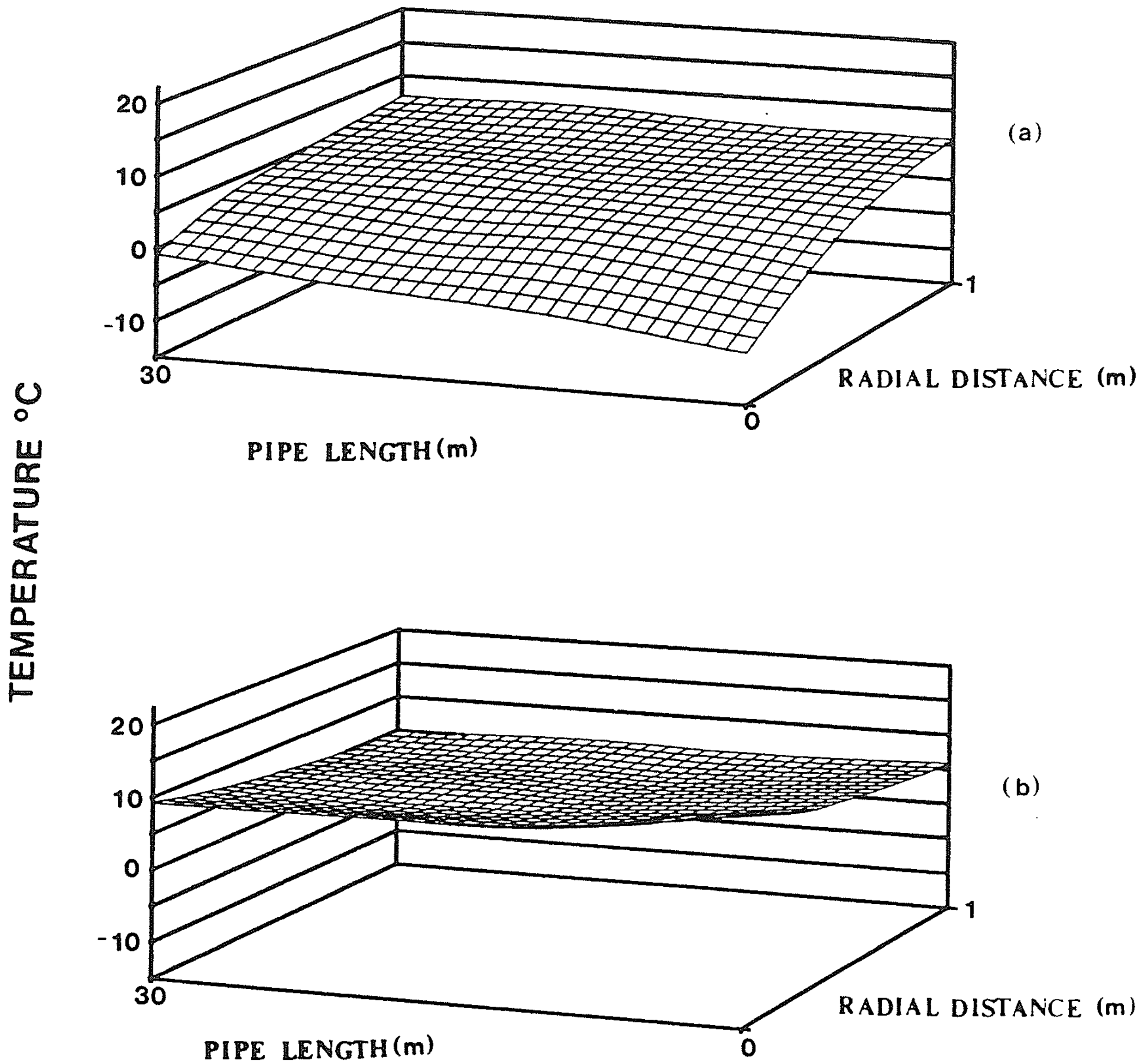

Figure 22: Soil temperature surfaces generated from thermocouples vertically down from surface of pipe Four. (a) February. (b) July.

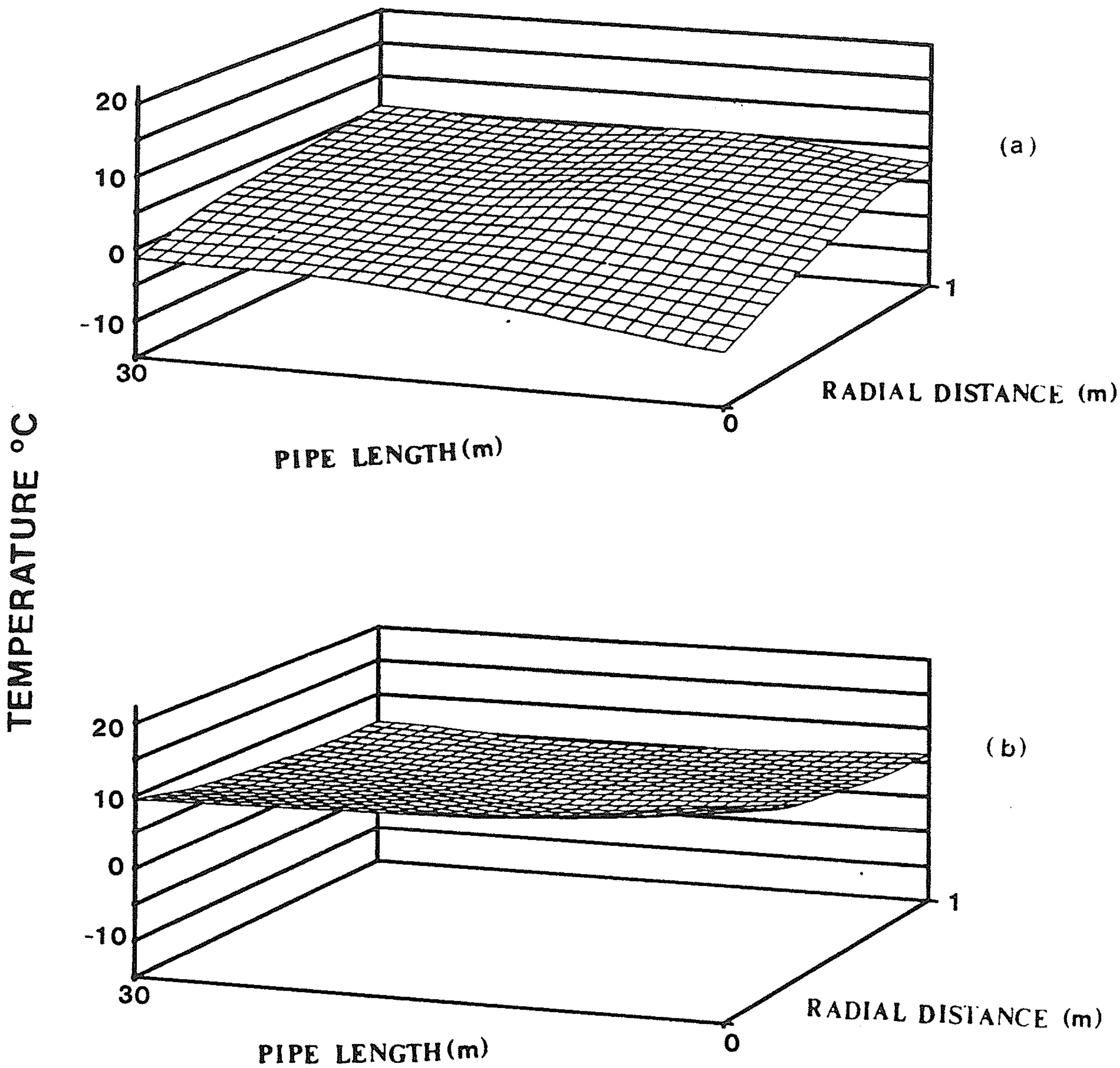

Figure 23: Soil temperature surfaces generated from thermocouples horizontal from surface of pipe Four. (a) February. (b) July.

surface is almost completely frozen. The soil temperature at the inlet of the pipe varies from -13°C at the pipe surface to -1°C at a distance of 1.0 m from the pipe surface (2.0 m below grade). At the outlet end of the pipe, the soil temperature varies from -5°C at the pipe surface to 1°C, 1.0 m from the surface of the duct.

By July, 1985 the soil temperatures vertically up from the pipe surface are all well above the far field temperature which had a monthly mean average of 4°C. Temperatures at the inlet end of the pipe vary from 16°C at the pipe surface to 10°C at 1.0 m vertically up from the pipe surface. Likewise, at the outlet end of the pipe, soil temperatures at the pipe surface are close to 11°C while 1.0 m upward they are 6°C.

The temperature surface vertically down from the air duct varied at the inlet end from -8°C at the pipe surface to 6°C, 1.0 m from the pipe surface in February. At the outlet end, the monthly mean temperature at the pipe surface was -1°C and at 1.0 m from the pipe surface it was 7°C. The soil was frozen for a distance of 400 mm from the pipe surface at the inlet end and to about 50 mm from the pipe surface at the outlet end. This temperature surface is warmer due to the reduced influence of the ambient environment. Whereas thermocouples 1.0 m directly up from the pipe are protected by only 2.0 m of cover, thermocouples 1.0 m directly down from the pipe are protected by 4.0 m of cover. In July, these soil temperatures range from 13°C at the pipe surface of the inlet end to 6°C at a distance of 1.0 m vertically down from the pipe surface. Temperatures at the outlet end varied from 9°C at the pipe surface to 5°C, at a distance 1.0 m away.

The behavior of the temperature surface extending horizontally from the pipe falls between the extremes of the other two radials, although these extremes are somewhat less evident under summer conditions. This will be the radius which will be discussed in detail as it gives an average indication of the performance at the 3.0 m depth. It is still important to realize that surface effects do occur and if the performance of the exchanger is reduced because of these effects then the depth of burial should be increased. However, based on the system performance observed at 3.0 m depth, it is doubtful whether additional benefits could be justified in light of increased excavation costs required for deeper burial.

Figure 4.24 illustrates the average monthly soil temperature surfaces for each pipe, for February 1985, as generated from the horizontal soil profiles. The higher airflow rate of Pipe Three causes the temperature change to be distributed along the entire pipe, whereas the lower airflow rate in Pipes One and Four causes a large portion of the temperature change to occur at the inlet end. The soil temperature surfaces show that the soil near Pipe Three ranges from -9°C to -5°C from the inlet to the outlet end. Meanwhile, the soil temperatures 1.0 m from the duct surface remain steady at 5°C. Along Pipe One, temperatures near the duct vary from -10°C to -2°C from the inlet to the outlet end, while temperatures 1.0 m from the duct varied between 3°C and 6°C along the duct. Similarly, temperatures along Pipe Four ranged from -8°C to -1°C and from 3°C to 6°C from the inlet to the outlet end, at the pipe surface and 1.0 m from the pipe surface respectively.

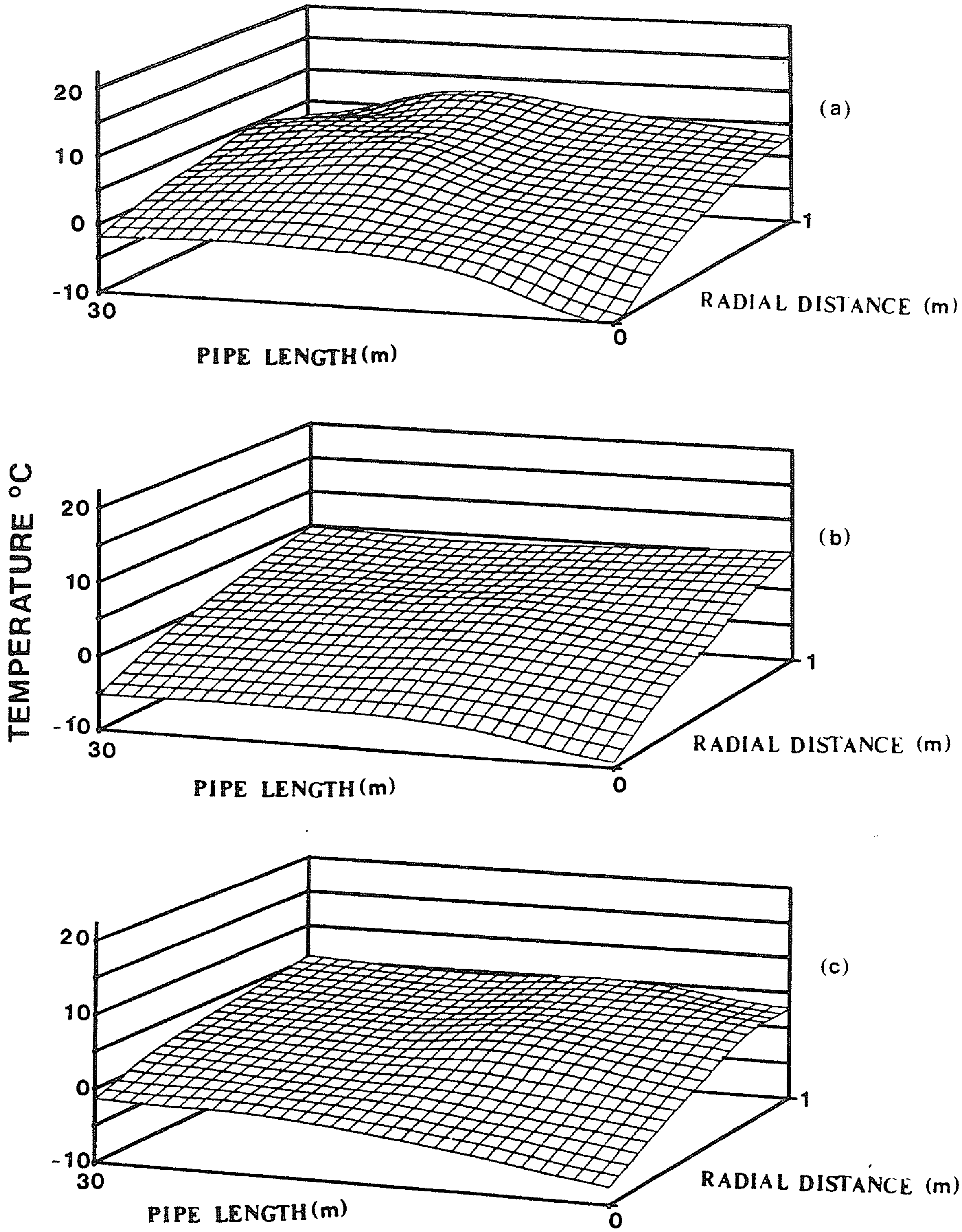

Figure 24: Soil temperature surfaces generated from thermocouples horizontal from the surface of each pipe, February, 1985. (a) pipe One. (b) pipe Three. (c) pipe Four.

Similar soil temperature surfaces for each pipe for July, 1985 are shown in Figure 4.25. Once again, the even distribution of heat along the surface of Pipe Three is evident. Soil temperatures along the surface of Pipe Three range from 15°C at the inlet to 12°C at the outlet. At a distance 1.0 m horizontal to the pipe surface, the soil temperatures range from 8°C at the inlet to 6°C at the outlet. Meanwhile, soil temperatures along the surface of Pipe One and Four both vary from about 13°C at the inlet to 10°C at the outlet. One meter from the surface of Pipe One, soil temperatures range from 5°C at the inlet to 8°C at the outlet end. One meter from the surface of Pipe Four, the temperature of the soil remains relatively constant at 5°C. The soil temperatures surrounding Pipes One and Four appear low compared to the soil temperatures near Pipe Three. This is especially true at the inlet end where higher soil temperatures are expected due to the lower flow rate. However, remember that because of these lower flow rates, to soil at the inlet end of Pipes One and Four was colder than corresponding soil near Pipe Three. Therefore, this soil requires a longer recovery period than that near Pipe Three.

Although the differences between the three soil temperature surfaces are slight, they clearly demonstrate the influence of airflow rate on the behavior of temperature change along the pipe. The soil temperatures near the surface of the air duct at the outlet end are more affected by a high airflow rate. While the soil temperatures at a distance of 1.0 m from the surface of the duct at the inlet end are more influenced by a lower airflow rate.

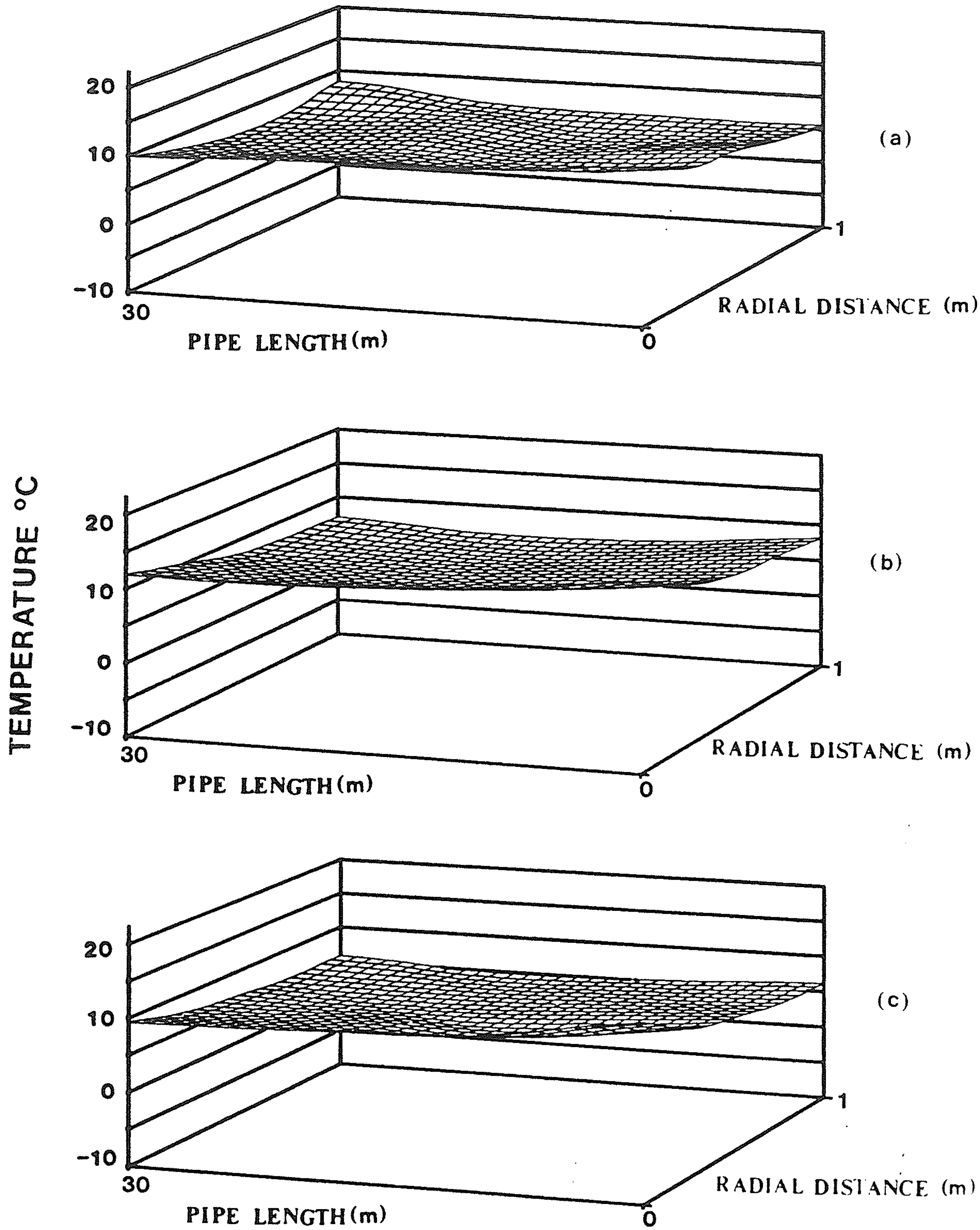

Figure 25: Soil temperature surfaces generated from thermocouples horizontal from the surface of each pipe, July, 1985. (a) pipe One. (b) pipe Three. (c) pipe Four.

Figures 4.26 (a to l) illustrate, in detail, the movement of the temperature front horizontally from the surface of Pipe Four through 1985. Far field temperatures at the 3.0 m depth are included to demonstrate how the heat exchanger has caused soil temperatures to deviate from their natural state. Table 4.1, summarizes the major findings of these graphs and Table 4.2 and 4.3 summarize the minimum and maximum soil temperatures around Pipes One, Three, and Four.

The first graph, that of December 1984, is included to establish the soil temperature surface before any air was drawn through the duct. Except for the soil very close to the duct, the entire temperature surface is very near far field temperature, 13°C. At the pipe surface, temperatures appear to be somewhat lower. This is probably caused by natural convection of cold ambient air through the pipes even before the fan was turned on.

Soil temperatures at the duct surface and 1.0 m horizontally out from the duct surface at both the inlet and outlet end, for each month in 1985 are summarized in Table 4.1. As well, the table gives a brief description of the extent to which the soil around the duct is frozen, as determined visually from the soil temperature surfaces. Soil temperatures at the pipe surface decrease steadily, reaching minimums in February of -7°C at the inlet end and -2°C at the outlet end. Movement of the temperature front away from the pipe surface occurs at a slower rate. A minimum temperature of 1°C occured 1.0 m from the surface at the inlet end in March and April, while at 1.0 m from the surface at the outlet end, a minimum of 3°C occurred in April and May. This is a time lag of two to three months after the minimum temperature occurred at the pipe surface.

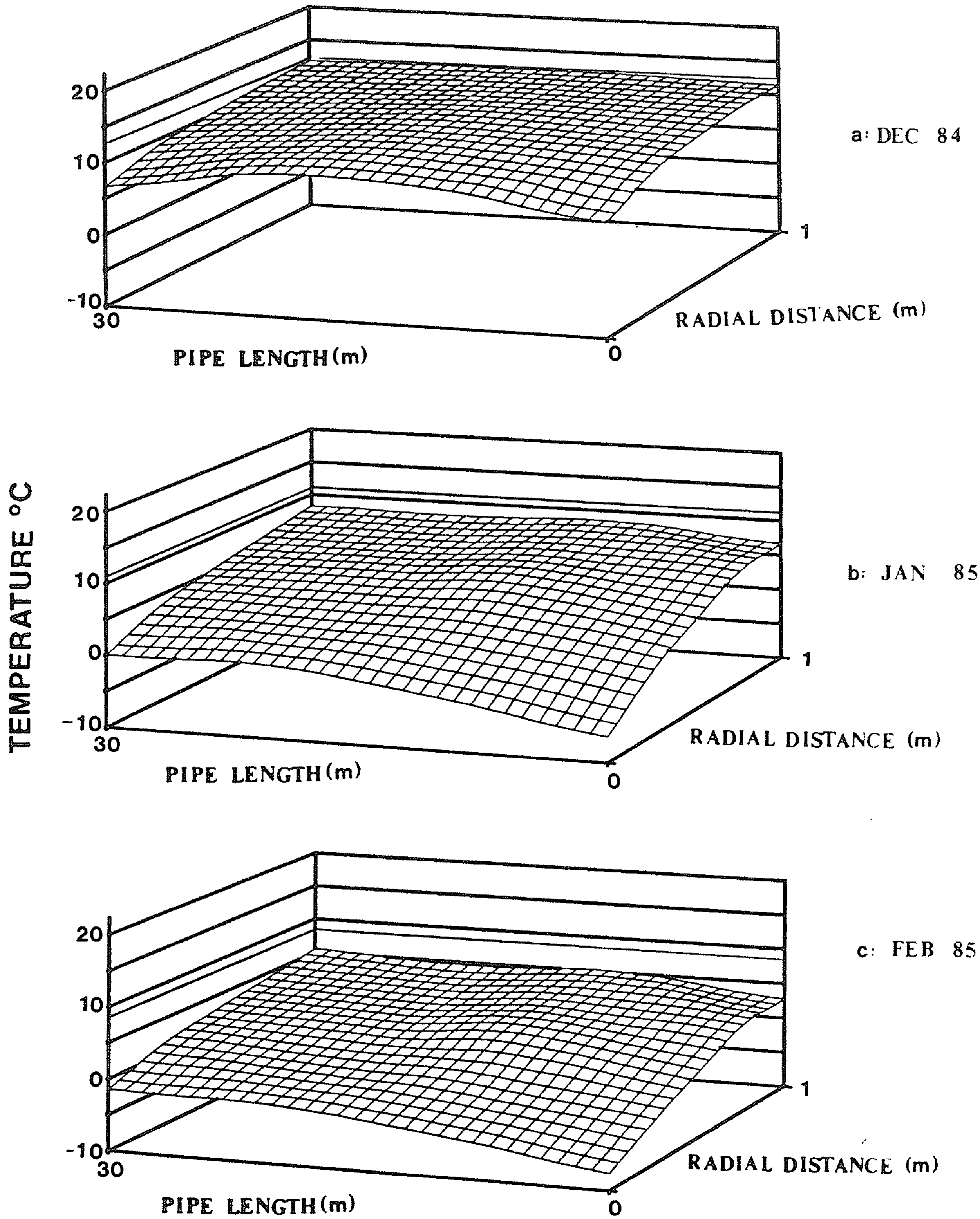

Figure 26: Monthly soil temperature surfaces generated from horizontal thermocouples from the surface of pipe Four. December, 1984 to December, 1985.

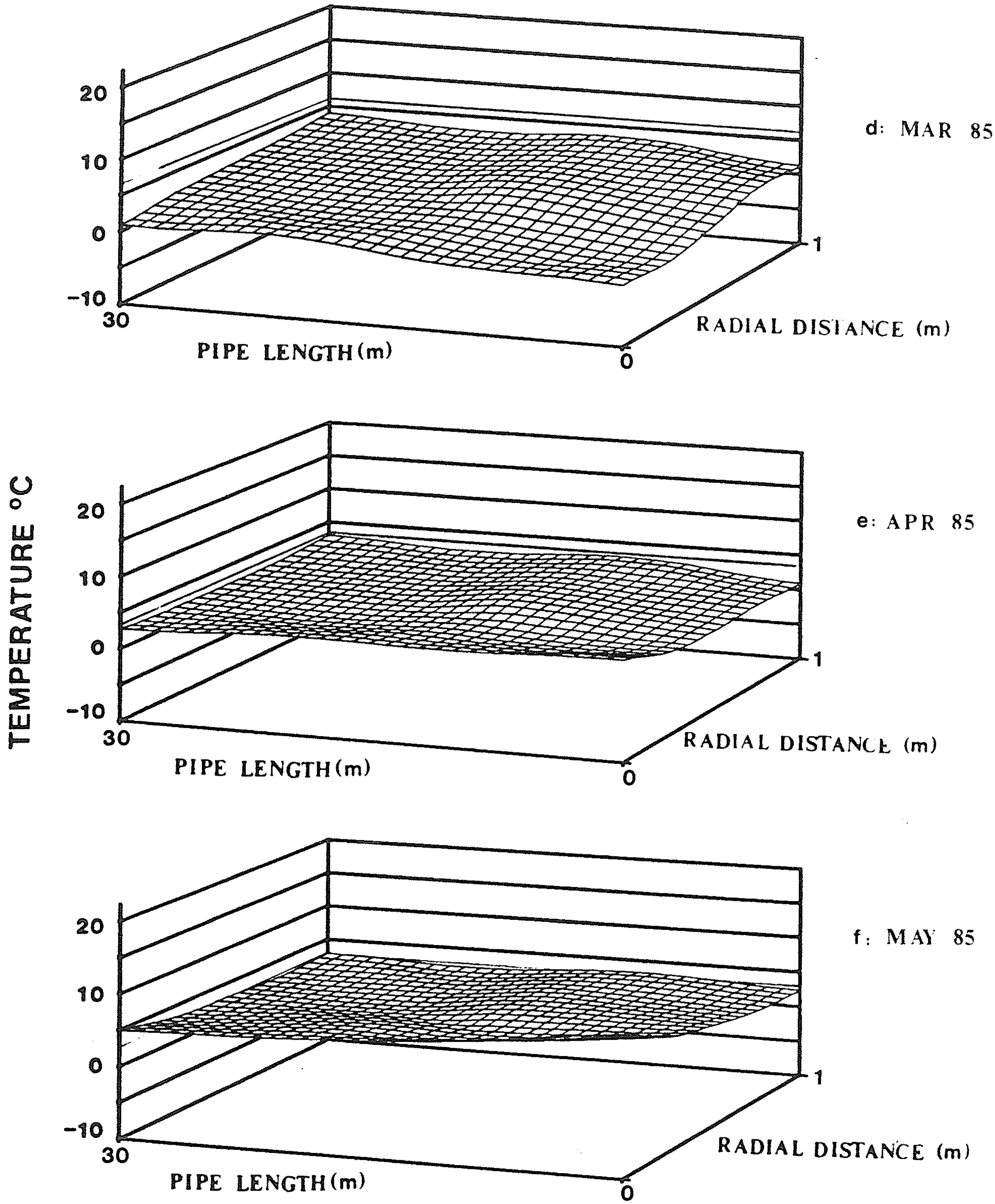

Figure 4.26: (continued).

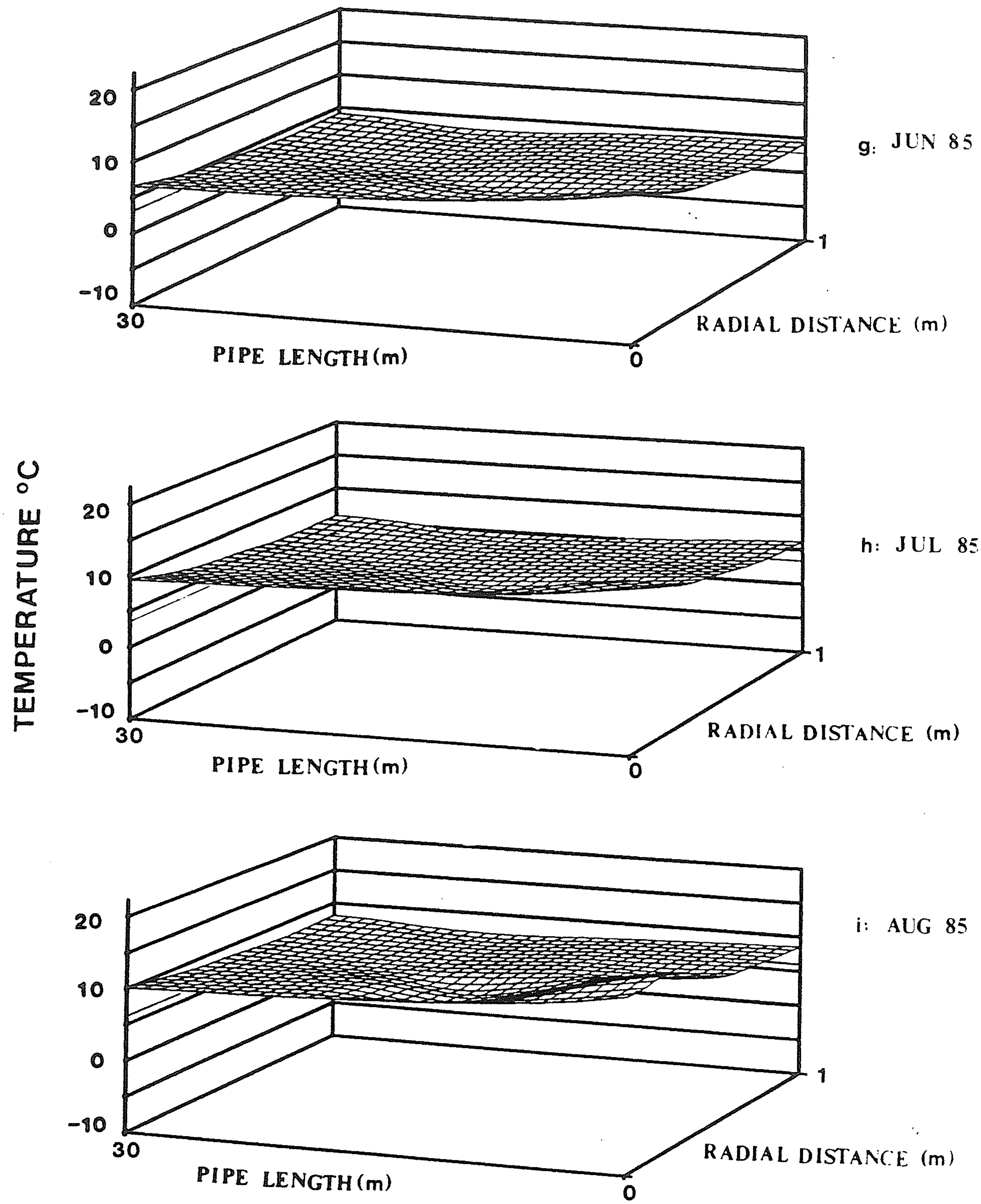

Figure 4.26: (continued).

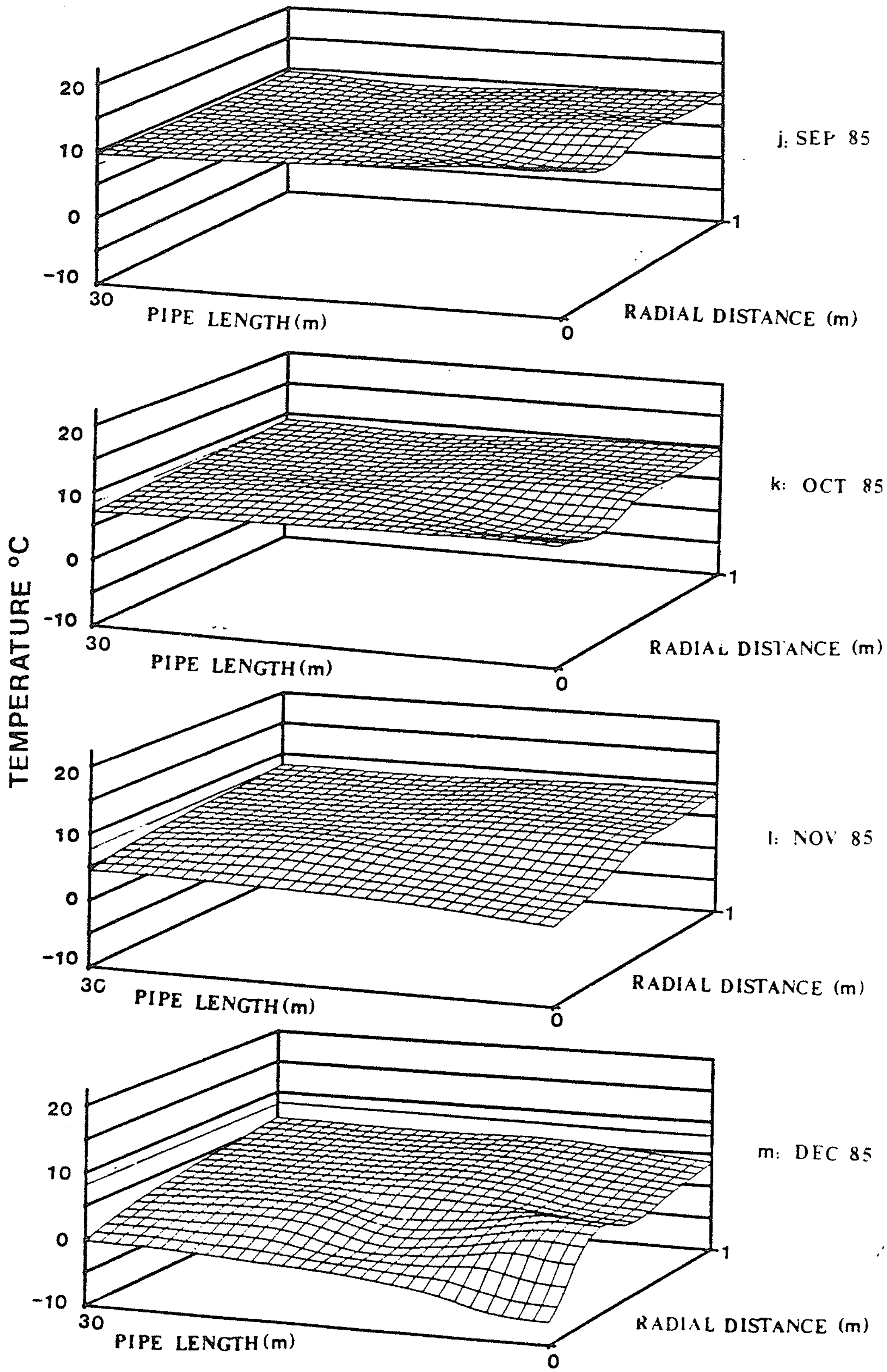

Figure 4.26: (continued).

TABLE 4.1

Summary of 1985 soil temperatures as measured horizontally from the surface of Pipe Four.

MONTH	SOIL TEMPERATURES (°C)					EXTENT OF FROZEN SOIL
	FAR FIELD	INLET		OUTLET		
		SURFACE	1.0 m	SURFACE	1.0 m	
Dec'84	13	6	12	7	12	none
Jan'85	11	-6	7	0	9	250 mm from pipe at inlet to 19 m along duct.
Feb'85	9	-7	3	-2	6	500 mm from pipe at inlet, 100 mm from pipe at outlet
Mar'85	6	-2	1	1	4	500 mm from pipe at inlet to 13 m along duct.
Apr'85	4	4	1	3	3	none
May 85	3	9	3	5	3	none
June 85	3	10	5	7	4	none
July 85	4	14	6	9	6	none
Aug'85	6	14	9	10	8	none
Sep'85	7	11	10	10	10	none
Oct'85	8	8	10	7	9	none
Nov'85	8	2	9	5	8	none
Dec'85	8	-7	4	0	6	150 mm from pipe at inlet to surface at outlet.

TABLE 4.2

1985 Maximum soil temperatures (°C), (monthly means)

POSITION	PIPE ONE MONTH	PIPE ONE TEMP(°C)	PIPE THREE MONTH	PIPE THREE TEMP(C)	PIPE FOUR MONTH	PIPE FOUR TEMP(C)
Far Field	Oct-Dec	8	Oct-Dec	8	Oct-Dec	8
Inlet						
(surface)	July-Aug	13	July	15	July-Aug	14
(1.0 m)	Sept-Oct	8	Sept-Oct	11	Sept-Oct	10
Outlet						
(surface)	August	11	July-Aug	12	Aug-Sept	10
(1.0 m)	October	10	Sept-Oct	10	September	10

TABLE 4.3

1985 Minimum soil temperatures (°C), (monthly means)

POSITION	PIPE ONE MONTH	PIPE ONE TEMP(°C)	PIPE THREE MONTH	PIPE THREE TEMP(C)	PIPE FOUR MONTH	PIPE FOUR TEMP(C)
Far Field	May-June	3	May-June	3	May-June	3
Inlet						
(surface)	February	-10	February	-9	February	-7
(1.0 m)	April	1	Mar-Apr	4	Mar-Apr	1
Outlet						
(surface)	February	-2	Jan-Feb	-5	February	-2
(1.0 m)	March	4	April-May	3	April-May	3

Far field temperature reaches a minimum of 3°C in May and June. Although the soil 1.0 m from the pipe varies from far field temperature, the minimum average monthly temperature at this point is nearly the same. This confirms that soils 1.0 m from the air duct are not undergoing drastic temperature changes. The soil 1.0 m from the pipe surface, for the length of the duct, is always within 5°C of far field temperature.

The soil temperature surface for January indicates that the soil at the inlet end is frozen to a distance of 250 mm from the duct surface and to a distance of 19 m along the duct from the inlet end. In February, the frozen soil has extended to 500 mm from the surface at the inlet, along the pipe, and to a distance of 100 mm from the surface at the outlet end. By March the soil has began to thaw. A 500 mm radius of frozen soil still exists around the duct at the inlet end but the frost only extends to 13 m along the duct from the inlet end. The soil was completely thawed by April.

By March, 1985 soil temperatures near the duct surface begin to rise. At this time, the soil temperatures 1.0 m away are still decreasing as the cold temperature front makes its way through the soil. Naturally, soil temperatures rise most quickly at the inlet end where the ambient air is at its warmest. As the soil temperatures rise, the corner of the temperature surface near the duct at the inlet end "flips up". Then, as the warm temperature front moves through the soil, this flip becomes a ripple which continues to move outward from the pipe.

The soil at the duct surface reaches a maximum of 14°C at the inlet end during July and August, and 10°C at the outlet end in August and September. Meanwhile, the soil at a distance of 1.0 m from the duct surface reaches a maximum of 10°C at both the inlet and the outlet end in September and October. Far field temperature reaches a maximum of 8°C in October and maintains this temperature until December.

The behavior of these soil temperature surfaces can also be used to support the argument of not using the earth-to-air system during the spring and fall. From Table 4.1, one can see that the minimum soil temperatures occur in February and March. By June the soil has been warmed substantially, therefore less temperature potential exists for cooling the summer air. If average monthly soil temperatures for March could be preserved even to May, the difference between the soil temperature available for cooling would decrease 11°C from 9°C to -2°C at the surface of the inlet end. At the surface of the outlet end, soil temperatures could be in the order of 1°C rather than 5°C. Likewise, if August soil temperatures could be preserved for use in November, soil temperatures at the pipe surface would be 9°C and 5°C warmer at the inlet and outlet end respectively.

Some temperature change will occur during spring and fall, even if the system is turned off, as the energy stored near the pipe will dissipate throughout the soil volume. However, the potential does exist to substantially increase system performance by adopting a schedule which does not include using the earth-to-air heat exchanger during the spring and fall.

4.3 ESTIMATED HEAT TRANSFER RATES

This section estimates the rate of heat transfer between the soil surrounding each duct and the air moving through each duct using basic heat transfer equations. The movement of energy within the system is calculated using the three methods presented by Lawand et al. (1983).

The rate of energy gained or lost by the air as it moves through the duct was calculated according to Equation 4.1

$$q = \frac{1}{n} \sum_{1}^{n} \frac{m\ C_p\ \Delta T}{L} \qquad (4.1)$$

where:
- q = the rate of energy gained/lost to the air (W/m)
- n = the number of discrete observations in the testing period
- m = mass flow rate (kg/s)
- C_p = specific heat of air in duct (J/kg°C)
- ΔT = measured temperature difference (°C) between inlet air (at 1.0 m station) and outlet (at 29.0 m station).
- L = length of duct between inlet and outlet air temperature sensors (m).

Note: All air properties are taken from Kays and Crawford (1980) for an air temperature of −10°C for heating and 20°C for cooling.

The heat transfer to the air was calculated for each temperature reading and averaged over the test period.

The energy change in the soil is calculated using measured soil temperatures. The soil surrounding the duct was divided into eight concentric rings, and the length of the duct was subdivided into thirteen sections as shown in Figure 4.27. These subdivisions were made, rather than working with the four concentric rings and six sections defined by the thermocouples, to reduce the volume of soil in

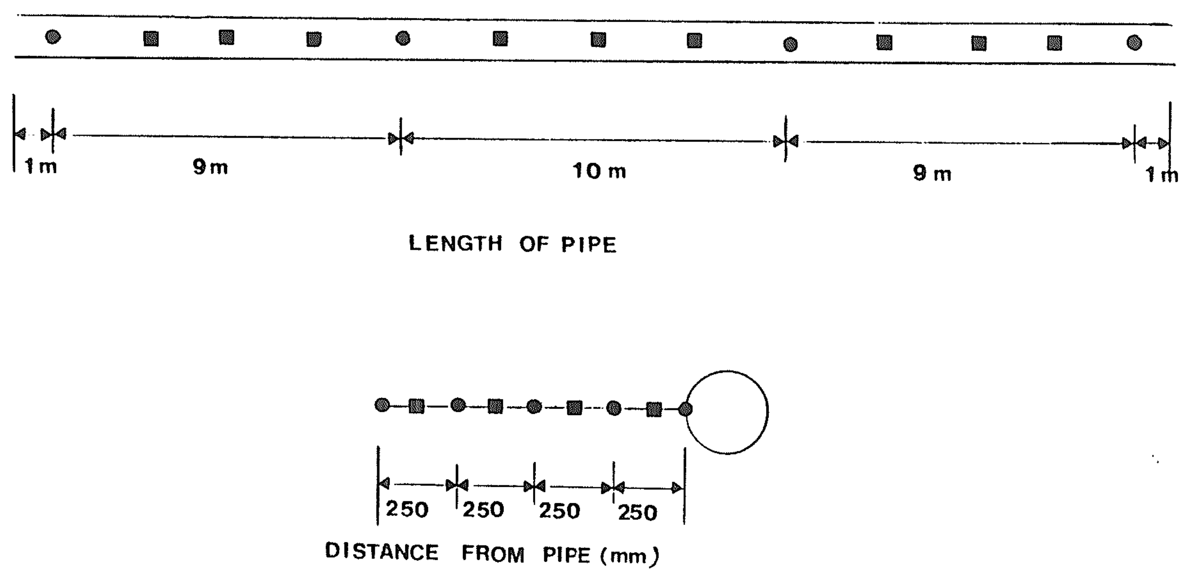

Figure 4.27: Subdivision of soil volume for heat transfer calculations. (●)actual thermocouple locations, (■) location of interpolated points.

each cylinder and to smooth out the influence of temperature from one section/ring to the next. In effect, a finer grid was created on which the heat transfer calculations were performed. The temperature at the center of each ring was calculated by linear interpolation between thermocouple locations. The heat flux was calculated from the difference in the average temperature of each volume over the testing period. The heat transfer rate was calculated using Equation 4.2

For $T_1 \geq 0$ and $T_2 \geq 0$ (4.2a)

$$q = \sum_1^j \sum_1^i \{[Cp_s(1-M_c)+Cp_w(M_c)]\ V_{ij}\ BD\ (T_2-T_1)\}\ /\ (\theta L)$$

For $T_1<0$ and $T_2<0$ (4.2b)

$$q = \sum_1^j \sum_1^i \{[Cp_s(1-M_c)+Cp_w(M_c)(1-F_c)+Cp_i(M_c)(F_c)]\ V_{ij}\ BD\ (T_2-T_1)\}\ /\ (\theta L)$$

For $T_1 \geq 0$ and $T_2<0$ (4.2c)

$$q = \sum_1^j \sum_1^i \{[Cp_s(1-M_c)(T_2-T_1)+Cp_i(M_c)(F_c)(T_2)+Cp_w(M_c)(1-F_c)(T_2)+ Cp_w(M_c)(-T_1)+LHF(M_c)(F_c)]\ V_{ij}\ BD\}\ /\ (\theta L)$$

For $T_1<0$ and $T_2 \geq 0$ (4.2d)

$$q= \sum_1^j \sum_1^i \{[Cp_s(1-M_c)(T_2-T_1)+Cp_i(M_c)(F_c)(-T_1)+Cp_w(M_c)(1-F_c)(-T_1)+ Cp_w(M_c)(T_2)+LHF(M_c)(F_c)]\ V_{ij}BD\ \}\ /\ (\theta L)$$

where: q = the rate of energy change in the soil (W/m)
i = number of concentric rings
j = number of sections along pipe length
Cp_s = specific heat of soil (J/Kg·°C)
M_c = soil moisture content at 3.0 m depth
Cp_i = specific heat of ice (J/Kg·°C)
Cp_w = specific heat of water (J/Kg·°C)
V_{ij} = volume of soil cylinder (m^3)
BD = bulk density of soil (kg/m^3)
F_c = percent of soil moisture which is frozen
T_1 = soil temperature at beginning of test period (°C)
T_2 = soil temperature at end of test period. (°C)
LHF = latent heat of fusion (J/kg)
L = length of pipe (m)
θ = length of testing period (s)

The heat transfer was computed over each time interval, summed, and divided by the total time of the testing period under consideration to establish an average heat gain/loss to the soil.

Finally, the convective heat transfer to/from the air was calculated using Equation 4.3

$$q = \frac{1}{n}\sum_{1}^{n} [A_s(T_a-T_w)] \,/\{[1/h_a+t/k_{pipe}]L\} \tag{4.3}$$

where: q = the rate of energy gained/lost to the air (W/m)
n = the number of discrete observations in the testing period
A_s = surface area of section of pipe (m^2)
T_a = air temperature at center of grid (°C)
T_w = mean surface temperature on outside pipe wall (° C)
t = thickness of pipe (m)
k_{pipe} = thermal conductivity of PVC pipe (W/m·°C)
h_a = heat transfer coefficient (W/m^2·°C)
L = total length of pipe (m)

where: $h_a = \frac{k_a}{d} Nu = \frac{k_a}{d} * 0.023 * R_e^{0.8} Pr^{0.3}$

k_a = thermal conductivity of air (W/m·°C)
d = diameter of duct (m)
Nu = Nusselt number
Re = Reynolds number
Pr = Prandtl number

Once again the heat transfer to the air was calculated for each temperature reading and averaged over the testing period.

Initially, a testing period of one day was chosen and the three heat transfer rates were calculated for the first day of each month. This was adequate for calculating the energy gained or lost to the air using both the mass flow and the convective heat transfer coefficient equations. However, when calculating the energy change in the soil, the slight changes in soil temperatures at any position over a monitoring interval caused the calculated heat transfer to fluctuate creating discrepancies in the energy balance. These discrepancies were especially evident when the soil was at or near freezing.

When soil moisture freezes it releases the latent heat of fusion at a rate of 334.93 kJ/kg. However, in clay soils where the soil moisture can be bound by cohesive forces, the soil moisture will freeze at different temperatures and at different rates. Therefore, within a 24 hour testing period where the soil temperature at one point may fluctuate from slightly above to slightly below freezing, the latent heat of fusion could be released and absorbed by the soil several times using traditional calculation methods. To account for a gradual freezing rate, only one-third of the soil moisture was assumed to freeze/thaw as the temperature of the soil passes 0°C. Table 4.4 summarizes the results of the heat transfer calculations for February 01, 1985.

TABLE 4.4

Summary of calculated heat transfer for February 01, 1985.

	HEAT TRANSFER RATE (W/m)		
	PIPE ONE	PIPE THREE	PIPE FOUR
Heat transfer to air (mass flow basis)	52.85	65.34	56.53
Heat transfer from the soil:			
no fusion	-43.19	-48.66	-38.37
fusion	-178.44	-343.48	-139.32
1/3 fusion	-88.27	-146.92	-72.01
Heat transfer to air (convective heat basis)	44.45	80.25	45.52

The heat transfer to the air is dependent on the mass flow rate and the temperature difference between the inlet and outlet air. Therefore the rate of heat transferred to the air in Pipes One and Four, both with an airflow rate of 0.05 m^3/s, is similar at 52.85 W/m and 56.53 W/m respectively. The larger airflow rate of Pipe Three, 0.1 m^3/s, creates a larger heat transfer rate, 65.34 W/m, despite the lower temperature difference between inlet and outlet air than that of the other two pipes.

When the energy change in the soil is compared to the heat gained by the air during this period, some interesting observations are made. If the latent heat of fusion is neglected and the heat transfer is calculated with regard to the energy change in the water/ice and soil only, between 68 and 82% of the heat supplied to the air is accounted for within the 1.0 m radius of the duct surface. However, once the latent heat of fusion is account for the results indicated that the soil is generating energy. Since this is impossible, the procedure for calculating the energy change in the soil surrounding the pipe must be faulty. Two areas where this procedure could be improved are in the calculating of the amount of soil moisture which is actually freezing and in the calculating of the change in soil temperature at any point over a desired time interval.

Looking first at the method of calculating the latent heat of fusion. The FORTRAN program assumes that as soon as soil at a temperature above freezing drops to 0°C or below, the soil moisture in that volume of soil freezes and the latent heat of fusion is calculated. However, not all of the soil moisture will freeze at 0°C. In fact the freezing process may not be complete until the soil temperature reaches -1°C or lower. To account for this, only 1/3 of the calculated heat of fusion was used in the energy change equation. A factor of 1/3 was choosen to give a rough estimate of the amount of frozen soil moisture based on the available literature. However, this once again resulted in a gross over approximation of the actual heat transfer, 127-225% of that calculated for air.

Next the method used to determine the temperature change in the soil at any point will be adjusted. Now rather than using actual recorded data, at three hour intervals over a 24 hour period, weekly means are calculated over a five week period and the temperature change was determined as the difference between the temperature at week five and the temperature at week one. This adjustment increased the testing period and decreased the effect of small fluctuations in individual thermocouples. By using a larger testing period and mean temperatures, the problems previously associated with calculating the amount of fusion are eliminated. It is now assumed that if the average soil temperature decreases to or below freezing during the 5 week period, that all of the soil moisture in the representative soil volume has become frozen. Likewise, if the average temperature increases from below 0°C to above 0°C, the latent heat of fusion is absorbed by the soil. Therefore, the total 334.93 kJ/kg of energy is used in the energy balance. Since the temperature front in the soil moves slowly, the use of weekly means, over a five week period will provide a good indication of the heat transfer which is occurring. The results of two, five week periods are presented in Tables 4.5 and 4.6. The first time interval demonstrates heating mode performance from January 27, 1985 to March 02, 1985 while the second demonstrates cooling mode performance from July 28, 1985 to August 31, 1985.

The heat transfer to the air in Pipes One and Four in heating mode are again very similar, 28.57 W/m and 28.49 W/m, while the heat transfer to the air in Pipe Three is 35.79 W/m. Now, the calculated heat transfer from the soil, including fusion, ranges from -29.11 W/m, -22.71 W/m, and -15.30 W/m for Pipes One, Three and Four respectively. Using these values, 102% of the heat gained by the air passing through Pipe One can be accounted for by calculating the energy change within a 1.0 m radius of the pipe surface. This analysis is intended only to estimate

TABLE 4.5

Summary of calculated heat transfer in heating mode.

January 27, 1985 to March 02, 1985

	HEAT TRANSFER RATE (W/m)		
	PIPE ONE	PIPE THREE	PIPE FOUR
Heat transfer to air (mass flow basis)	28.57	35.79	28.49
Heat transfer from the soil:			
no fusion	-8.44	-5.74	-5.86
fusion	-29.11	-22.71	-15.30
1/3 fusion	-15.32	-11.40	-9.01
Heat transfer to air (convective heat basis)	24.08	44.18	24.28

TABLE 4.6

Summary of calculated heat transfer in cooling mode.

July 28,1985 to August 31, 1985

	HEAT TRANSFER RATE (W/m)		
	PIPE ONE	PIPE THREE	PIPE FOUR
Heat transfer to air (mass flow basis)	-6.93	-13.80	-11.31
Heat transfer from the soil	3.88	4.21	5.52
Heat transfer to air (convective heat basis)	-7.29	-10.88	-5.72

the heat transfer which is occurring between the soil and the air, so this two percent over estimation is not alarming.

The values calculated for Pipe Three and Four correspond to 63.5 and 53.7% of the heat transfer to the air in Pipe Three and Four. The difference between the heat transferred to the air and the heat given up by the soil is probably a result of the heat transfer occurring over a larger radius than the 1.0 m from the pipe surface which was originally assumed. Table 4.7 gives the results of calculations where the energy

TABLE 4.7

Comparison of heat transfer rates, soil to air (mass flow basis), and air to air (convective heat basis, mass flow basis)

		PERCENT OF HEAT TRANSFER RATE TO THE AIR USING (MASS FLOW BASIS)		
		PIPE ONE	PIPE THREE	PIPE FOUR
Heat transfer from the soil (assuming fusion):				
Heating:	1.00 m radius	101.9	63.5	53.7
	1.25 m radius	116.8	75.1	67.3
	1.50 m radius	134.5	86.7	83.7
Cooling:	1.00 m radius	56.0	30.5	48.8
	1.25 m radius	84.8	49.7	74.1
	1.50 m radius	119.9	72.6	104.1
Heat transfer to air (convection basis):				
Heating:		84.3	123.4	85.2
Cooling:		105.2	78.8	50.6

change in the soil was calculated using: (1) the soil temperature profile measured within a 1.0 m radius of the pipe surface, (2) the same soil profile but extending it to 1.250 m from the pipe surface and assuming that the the temperature at that point is equal to far field temperature, and (3) the same soil profile but extending it to 1.500 m from the pipe surface and assuming that the temperature at that point is equal to far field temperature. Now, 87% and 84% of the heat transferred to the air in Pipes Three and Four respectively, can be accounted for by assuming a radius of influence of 1.500 m (Table 4.8).

In cooling mode, the heat transferred to the air is -6.93 W/m, -13.80 W/m and -11.31 W/m for Pipes One, Three, and Four respectively. Pipes One and Four no longer have similar heat transfer rates. Possible reasons for this are that these equations do not account for condensation which occurs as the warm summer air is cooled. Since condensation was not monitored during operation, there is no way to speculate on how it may influence the heat transfer rate. As well, the distribution of the heat transfer in the soil surrounding pipe One has been shown to be different than that of Pipe Four. This causes the temperature of the soil near the pipe to vary between pipes. Since the temperature change of the air is influenced by the temperature of the soil near the pipe, and since the heat transferred to the air is calculated according to the temperature change of the air, this causes deviations between pipes.

When considering the energy change in the soil, the calculated values varied from 3.88 W/m, 4.21 W/m and 5.52 W/m for Pipes One, Three, and Four respectively. These values, while substantially lower than the

TABLE 4.8

Calculated heat transfer assuming larger radii of influence.

		HEAT TRANSFER RATE (W/m)		
		PIPE ONE	PIPE THREE	PIPE FOUR
Observed heat transfer from the soil:				
heating:	no fusion	-8.44	-5.74	-5.86
	fusion	-29.11	-22.71	-15.30
	1/3 fusion	-15.32	-11.40	-9.01
cooling:		3.88	4.21	5.52
Calculated heat transfer assuming no influence 1.25 m beyond pipe:				
heating:	no fusion	-12.69	-9.90	-9.74
	fusion	-33.37	-26.86	-19.18
	1/3 fusion	-19.58	-15.55	-12.90
cooling:		5.88	6.86	8.38
Calculated heat transfer assuming no influence 1.50 m beyond the pipe:				
heating:	no fusion	-17.76	-14.85	-14.43
	fusion	-38.44	-31.02	-23.86
	1/3 fusion	-24.66	-20.50	-17.57
cooling:		8.31	10.02	11.77

winter heat transfer rates, reflect the lower temperature differences between ambient air and soil temperatures under summer conditions. In comparing the calculated energy change in the soil with the calculated heat transfer from the air, a 1.0 m radius of soil accounts for 56, 31, and 49% of the total heat transfer from the air for Pipes One, Three and Four respectively. However, speculating on the effect of increasing the radius of soil which is influenced by the duct would indicated that the heat transfer occurs at a radius of between 1.250 and 1.500 m for Pipes One and Four but an even larger radius of influence probably exists for Pipe Three.

When the heat transfer to the air is calculated on a convective heat transfer basis rather than a mass flow basis, the convective heat transfer equation predicts the heat transfer to 84, 123, 85% of that calculated using the mass flow basis for Pipes One, Three, and Four respectively in heating mode. In cooling mode the convection equation accounted for 105, 78, and 51% of that calculated using the mass flow equation. Once again the summer data do not account for the occurrance of condensation in the pipes.

In all instances, in calculating the heat transfer to the air, the air temperature in the duct was measured at each station by only one thermocouple rather than by a grid of thermocouples across the duct which would be desirable. The accuracy of the recorded air temperatures is therefore dependent on the placement of that thermocouple and on the mixing of the air as it flows through the duct.

Chapter V

CONCLUSIONS

All conclusions are based on data from a full-scale, field model earth-to-air heat exchanger located at the University of Manitoba Research Farm at Glenlea, Manitoba. This unit consisted of three isolated PVC pipes (Pipe One, a 250 mm diameter pipe with a 0.05 m^3/s airflow rate; Pipe Three, a 250 mm diameter pipe with a 0.10 m^3/s airflow rate; and Pipe Four, a 150 mm diameter pipe with a 0.05 m^3/s airflow rate) buried to an average depth of 3.0 m below grade in Red River gumbo clay.

On the basis of data from the above mentioned unit, the following conclusions can be drawn.

1. The background or far field soil temperature at the 3.0 m depth ranged from 2°C during April and May to 9°C during October and November.
2. Temperature change as the air passes through the pipe is airflow dependent. A greater airflow rate will reduce the temperature difference between inlet and outlet air and result in a less consistent outlet temperature.
3. Under winter conditions, at an average inlet temperature of -35°C, average temperature changes of 19°C to 29°C can be expected.
4. Under summer conditions, at an average inlet temperature of 30°C, average temperature changes of 12°C to 18°C can be expected.

5. At ambient air temperatures in the range -10°C to 15°C little useful tempering occurs.
6. Exchanger efficiencies of 35 to 57% were determined with winter conditions, 45 to 70% with summer conditions.
7. On average, approximately 80% of the total air temperature change occurs within the first 20 m of pipe length.
8. The soil temperature profile vertically up from the pipe surface reflects the influence of the ambient environment. However, due to the observed performance at a 3.0 m depth of burial, it is doubtful that the additional cost of a deeper depth of burial could be justified.
9. Airflow rate has the greatest effect on system performance of the factors studied, that is of airflow rate, pipe diameter, and pipe length.
10. A higher airflow rate will result in the temperature change in the soil surrounding the pipe being distributed more evenly along the length of the pipe. A lower airflow rate will cause the temperature change to be concentrated near the inlet end of the pipe and extend to a greater radius from the pipe surface.
11. A time lag in the order of two to three months exists between the occurrance of the minimum ambient air temperature and the minimum soil temperature 1.0 m from the surface of the pipe.
12. Summer operation will allow the soil to recover from the effects of winter operation. In fact soil temperatures near the pipe will exceed the background soil temperatures by the end of the cooling season.

13. The unfrozen moisture content of frozen soils must be given careful consideration when doing discrete calculations of the heat transfer between the soil and the air.
14. Classical heat transfer equations can be used to describe the rate of heat transfer between the air and the soil.
15. Heat transfer to/from the air is in the range of 28 to 36 W/m under winter conditions and 7 to 14 W/m under summer conditions.
16. Providing adequate drainage of condensation from the pipes and ensuring that all joints are properly sealed to eliminate the possibility of soil moisture entering the pipes should be considered one of the most crutial elements in the design of any earth-air system.

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