THE FEASIBILITY OF USING A SOIL-AIR TEMPERING SYSTEM WITH A FARROWING-NURSERY OPERATION

by

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A Thesis
Submitted to the Faculty of Graduate Studies
in Partial Fulfilment of the Requirements
for the Degree of

Master of Science

Department of Agricultural Engineering
Universtity of Manitoba
Winnipeg, Manitoba

ISBN 0-315-77844-X

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Abstract

A feasibility study was conducted to determine the economic viability of using a soil-air tempering system with a swine farrowing-nursery operation. Soil-air tempering is a heat exchange process in which the thermal energy of the soil is transferred to an airflow passing through buried pipes. A three year database of soil and air temperatures was obtained from a tempering pilot project at the University of Manitoba (Murray, 1987). This system consisted of PVC pipes buried to a depth of 3.0 m, which compared differing pipe diameters (150 mm and 250 mm), pipe length, and airflow rate (50 L/s and 100 L/s) against heat recovery.

The feasibility study made use of a computer simulation which modeled the heating and ventilation of a 250 sow, farrowing and nursery structure.Data for winter and summer operation were modeled for six soil-air tempering systems, and a conventional ambient air ventilation system. The tempering systems were sized to meet the demand of winter operation. The winter model provided the design heat load and energy consumption of the period for each of the systems. The summer model calculated the resultant room temperatures within the structure.

The conventional ventilation system had a winter design heat load of 75 kW, and consumed 20,321 kW·h of electricity over the study period. The most productive soil-air tempering system had a design heat load of 22.8 kW and consumed 7,990 kW·h of

energy. The summer simulation did not demonstrate any appreciable temperature control, as compared to the conventional system.

Net present value (NPV) and equivalent annual worth (EAW) were calculated for each of the systems based upon a 5% real rate of interest, and a 20 year service life. The cost of electricity was assumed to be \$0.03/kW·h. Cost estimates were prepared for each of the soil-air tempering systems, and they ranged from \$52,217 to \$119,866. Based on these values, the NPV of the conventional system was determined to be (\$19,325), while the tempering systems ranged from (\$458,200) to (\$124,785). The EAW of the conventional system was (\$1,578), while the tempering systems were between (\$4,685) and (\$10,025).

Acknowledgements

I would like to thank my wife and family for standing behind me and providing the encouragement to finish this thesis. If it had not been for your efforts, I never would have finished.

I gratefully acknowledge Dr. M.G. Britton and the Department of Agricultural Engineering for the patience and support they gave me over the years it took to prepare this thesis.

I would also like to thank Dr. B.E. Prentice of the Transport Institute and Dr. Q. Zhang of the Department of Agricultural Engineering for consenting to and sitting as my thesis committee.

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1.0 INTRODUCTION

The purpose of this thesis is to determine if there is economic justification to pursue the use of Soil-Air Tempering as a means of supplementing the heating and cooling requirements of a swine farrow to nursery operation. Previous work in Canada and the United States have indicated a potential saving of heating and cooling costs. The majority of work in the United States has concentrated on cooling benefits, but some work in the northern states have presented heating results. Very few systems are in operation in Canada, and as a result there is little substantiating cost data. Potential interest in the systems could come from industries requiring large quantities of heated or cooled ventilation air. Within the agricultural sector this could include any type of livestock confinement housing or storage facilities.

The thesis modeled a typical swine farrowing and nursery structure to determine the design heat loads and system requirements for a conventional ventilation system and a number of soil-air tempering configurations. Based upon the projected performance of each system, capital cost estimates and projected cash flows were prepared for each system. A comparison of the net present values and equivalent annual worth demonstrated the economic viability of each of the systems as compared to the conventional system.

The basis of the comparisons was a typical Western Canadian operation which has a significant heating and ventilation requirement. The selected system was a 250 sow, farrow and nursery operation. The structure represented a typical structure built to recommended specifications for Western Canadian conditions. The heat transfer model took into consideration the sensible and latent heat transfer between the animals, the structure and ambient conditions. Heat and moisture production rates of the animals were obtained from current animal science literature. The climatic database was the result of two and one half years of temperature measurement at a pilot soil-air tempering system (Murray, 1987). The database consisted of 330 discrete measurements of ambient, soil and air flow temperatures taken on a 3 hour interval over the 3 year life of the project.

The computer models simulate the heat balance of the system for both winter and summer operation. The models assume the soil-air tempering systems have been installed for some time, and that their performance is equivalent to the results of the pilot system. By modelling a variety of soil-air tempering configurations, the economic effects of pipe size, air flow rate, and lateral lengths were determined. Each of the systems was modeled using the same temperature database, with the only difference being outlet temperature for each configuration.

The economic evaluation compared each of the systems on an equal basis over the lives of the systems. Net present value of each system illustrated the long term financial impact of each decision. Equivalent annual worth of each system showed the annual operating cost or benefit. Cost estimates were prepared for each system using current material, labour, and energy costs representative of Manitoba and Saskatchewan. In this manner a simulation of continuous operation of a commercial scale operation was made using a number of equivalent systems.

2.0 LITERATURE REVIEW

2.1 Soil-Air Heat Exchanger Systems

The concept of using the earth as a heat source or heat sink to temper ventilation air is far from new. A number of cases have been recorded in the past describing unique situations. The object in recent times has been to reduce the dependence on a unique phenomenon and develop a reliable source of in-situ data and design criteria applicable to the general conditions. Among the first to take a serious look at Soil-Air Tempering was Scott et al. (1965) at Cornell University. At the time of their study, the economic feasibility of such systems was doubtful, and little useful work was attempted until the late 1970's when the cost of energy renewed interest in energy conservation..

Weisbecker, Jacobson and Jordan (1980) reported on a system for a 24 sow swine farrowing house in Blue Earth County, Minnesota. Their system consisted of 5 - 130 mm diameter pipes and 5 - 150 mm diameter pipes. The pipes were plastic, nonperforated, drainage tile with a lateral length of 61 m. Laterals were buried to a depth of 2.5 m. Ventilation air was drawn through the pipes to a header, which lead to a plenum in the barn. An auxiliary fan was installed in the plenum to supply additional summer ventilation. Their results were presented as a series of graphs and a calculation of the economics of the system. The graphs showed outside temperature, barn temperature and plenum temperature for typical days from December, 1979 to August, 1980. The authors calculated an annual heat savings of $270 US/year. Total cost of the system was $2000, which was approximately $1/ft of lateral length. This system had a payback period of 7.4

years based solely upon heat savings.

Goetsch, Peterson and Muehling (1981) presented a study on four earth-tube heat exchangers for heating and cooling the ventilation air for swine farrowing and nursery buildings. Their paper described the four systems, the results of data collection from the summer of 1980 to the summer of 1981, and some design recommendations based upon their observations. The four systems followed the same basic concepts in terms of pipe selection, depth of bury, and ventilation systems. All of the operators selected nonperforated plastic drainage tubing. The primary reasons for this selection were cost and availability. Depth of bury was fairly consistent in a range between 2.4 m and 3.5 m deep. All of the systems used negative exhaust pressure to ventilate the rooms, and centrifugal fans to draw air through the heat exchangers.

Differences between the four systems were pipe diameter, lateral length and configuration, airflow rates and soil type. Three different diameters were used; 130 mm, 250 mm and 300 mm. Lateral lengths were between 30.5 m and 79.5 m. Two of the systems used a radial configuration for the laterals, the other two systems had parallel laterals spaced 1.2 m apart. Airflow velocities were between 1.11 m/s and 2.67 m/s. Three of the systems were placed in silt/clay soils, while the other system was placed in sandy soil.

Data were collected from all four systems. The results from the first two systems were used in the design of the last two systems. Data collected consisted of air velocity, air temperature and relative humidity measurements at the inlets and outlets. On the fourth system a series of temperature measurements were made along the centre tube of the lateral layout.

Based on their findings the authors made a number of design recommendations for soil-air heat exchangers. The first recommendation was that silt or clay soils were preferable to sandy soils. Heavier soils had better surface contact with the pipes. The second recommendation was that an airflow of 2.54 m/s be used, based on the most effective of the systems tested. Related to this the authors recommend that a surface area to flow ratio of 393 $m^2/m^3/s$ be maintained to achieve optimum results. There was not a preference between either the radial or parallel pipe configurations, and they suggested that this decision be based upon the particular situation. Pipes laterals should be buried to a depth range of 2.1 m to 3.7 m. Depth of bury must be balanced between surface effects and the cost of deeper excavation. Pipe laterals should be spaced 2.4 m to 3.0 m apart, if a parallel configuration is used, and the pipes should have a minimum slope of 0.25% to facilitate drainage.

In a paper by Goetsch and Muehling (1984), three systems installed in the early 1980's were described and the early results discussed. One of the systems discussed in the report was also part of an earlier paper (Goetsch et al., 1981). At the time of the system's installation nearly 40 units were in operation in the state of Illinois, mainly in swine facilities. The authors indicated that there was a need for further performance data and construction guidelines to optimize the design of these systems. The three systems and their results are discussed in the following paragraphs.

The first system was built in 1981, for a 30 crate farrowing operation. The structure had an enclosed area of 174 m^2. The heat exchanger consisted of 5-300 mm diameter, 79.2 m long, parallel laterals buried to a depth of 3.1 m. Air was drawn through the laterals by a centrifugal fan, and delivered to a plenum. A winter airflow rate of 0.4 m^3/s and a summer airflow rate of 1.0 m^3/s was used. Ventilation air was supplied to the rooms via ceiling ducts. Additional exhaust fans were used to provide increased summer ventilation.

The second system was built in 1982, for a combined farrowing-nursery operation. Combined area of the two structures was 437 m^2. The heat exchanger consisted of 8-300 mm diameter, 61 m long, parallel laterals buried to a depth of 3.4 m. Average airflow rate for winter and summer operation was 0.74 m^3/s. All of the winter ventilation air for the farrowing barn and some of the nursery requirement was preconditioned before entering the rooms. Each structure was provided with a duct system to distribute the ventilation air through the buildings.

The last system discussed was built in 1982, for a 31 crate farrowing operation. The farrowing house had an enclosed area of 179 m^2. The system consisted of 6-300 mm diameter, 70 m long, parallel laterals buried to a depth of 3.1 m. Winter airflow rate was 0.6 m^3/s, and the summer airflow rate was 1.1 m^3/s. An insulated enclosure equipped with a fan and a heater drew air through the system. A duct system in the structure was used to distribute the air within the structure. Additional ventilation was provided by exhaust fans and eave inlets.

Results for the first two systems were based on thermocouple measurements and the energy costs of operating the system. Temperature measurements of the airflow through the first system showed that the first 30.5 m of the laterals contributed two thirds of the total daily average of the sensible cooling load. The authors concluded that the last 18.2 m of pipe could be omitted, if the final 4% to 9% of the potential cooling was not necessary. Winter operation resulted in a savings of 41% of the entire ventilation heating load, a savings of $535 US. The heat exchanger installed in the second system replaced 34% of the total ventilation heating load, for a total savings of $313 US. There was no temperature or cost results presented for the third system.

In general, the authors recommend lateral lengths and airflows for three diameters of pipe. For 300 mm diameter pipe, a 61 m lateral length for summer cooling, and a length of between 61 m and 72.9 m for winter heating were recommended. An airflow rate of 0.19 m^3/s was recommended for the 300 mm diameter pipe. For 254 mm diameter pipe a lateral length of 64.0 m and an airflow of 0.13 m^3/s were suggested. For 154 mm diameter pipe a lateral length of 30.5 m to 38.1 m and an airflow of 0.05 m^3/s were recommended. It was found that silt or clay soils required shorter lateral lengths as compared to sandy or lighter soils. Moisture content of the soil also had some effect on the performance of the heat exchanger but the authors did not quantify this effect.

Other design criteria mentioned in the paper were surface to flow rate ratios, depth of laterals, lateral spacing, line slope, and header insulation. Based on the above systems a surface to flow rate ratio of 314 $m^2/m^3/s$ to 393 $m^2/m^3/s$ was recommended. This ratio represents the total inside surface area of a smooth wall pipe to the total airflow. The suggested depth of bury for the laterals was between 2.1 m and 3.7 m. Parallel laterals should be spaced at 2.4 m to 3.0 m apart, and a minimum line slope of 0.25% should be maintained for all systems. The header which connects the laterals to the structure should be insulated to RSI-3.35 (R-19) to a depth 1.8 m below grade.

Based upon these and other cases, the authors attempted to determine the costs and benefits of these systems. The estimated cost of a system was between \$1.18 US/m^3/s to \$1.79 US/m^3/s (\$2 US to \$3 US per ft^3/min) of air capacity. At the time of writing the systems had an estimated payback period of 4 to 6 years based on heat savings, excluding increases in fuel costs and interest. Though it is far more difficult to quantify the benefits of summer cooling, Goetsch and Muehling (1984) felt that summer cooling could result in reduced sow heat stress, more efficient sow milk production, and larger pig weaning weights.

Borg (1987) reported on a system installed in a twelve room veal barn in Haynes, Alberta. This was a full size system consisting of 12-300 mm diameter laterals, 30 m long. The laterals consisted of corrugated non-perforated drainage pipe buried to a maximum depth of 3.66 m. The lateral length was based upon current literature. Air was drawn through the laterals into a preheat attic. The system air velocity was between 1.25 m/s and 3.25 m/s. Two centrifugal fans, with airflow rates of 2.0 m^3/s/unit, were used to draw air through the lateral pipes. The air was then further heated to room temperature using a hot water unit heater in the attic. Air was drawn into the rooms from the plenum through wall slots by exhaust fans mounted in each of the rooms. The entire system had a capital cost of $15,672.42 which included materials and labour. Though there was little supporting documentation, it was felt that there was great potential for winter heating and summer cooling using the Soil-Air Heat Exchanger.

Murray and Britton (1985) present the background and the development of the soil-air tempering project at the University of Manitoba. This paper described the results collected after the first year of operation of their pilot system. Representative data was presented showing soil temperatures, and relative tempering effects on the three pipes in operation.

Initial results suggested that lower airflow rates would result in a greater tempering effect, and ambient temperature was a principle factor in the degree of tempering. In general, the report showed that favourable heating and cooling could be achieved by the system and that further work was justified. This work was further developed and discussed in the thesis presented by Murray (1987).

Murray (1987) reported on a Soil-Air Heat Exchanger at the University of Manitoba Research Station, located at Glenlea, Manitoba. The heat exchanger consisted of 4-30 m PVC pipes buried to an average depth of 3 m. Two pipes had a diameter of 250 mm, with air flows of 0.05 m^3/s and 0.10 m^3/s, respectively. The other two pipes had a 150 mm diameter, with air flows of 0.05 m^3/s and 0.15 m^3/s, respectively. Thermocouples were installed to measure temperatures at 3 hour intervals over 3 years of continuous operation. The temperatures which were monitored included ambient air, air temperatures along each pipe, outlet air temperature, soil temperatures along each pipe and soil background temperatures.

Air temperature difference measured between inlet and outlet had recorded maximum changes of 29° C for winter operation, and 18° C operating under summer conditions. System regeneration was accomplished through year round operation of the system. Murray was able to describe the system operation through the use of classical heat transfer equations.

Through this research work a number of observations and conclusions were made in regard to the operation of a Soil-Air Heat Exchanger:

1. The far field temperature (3.0 m from the pipes) had a low temperature of 2° C in the months of April and May, and a high temperature of 9° C in the months of 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 will result in less consistent output temperature.

3. Under winter conditions , at an average inlet temperature of -35° C, temperature changes ranging from 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 temperatures in the range of -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 buried depth, it is doubtful that the additional cost of a deeper depth of bury could be justified.

9. Airflow rate had the greatest effect on system performance of the factors studied, that was of airflow rate, pipe diameter and pipe length.

10. Higher airflow rates resulted in more evenly distributed temperature changes in the surrounding soil along the length of the pipe. A lower airflow rate caused the temperature changes to concentrate 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 2 to 3 months exists between the occurrence 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 W/m to 36 W/m under winter conditions, and 7 W/m 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 crucial elements in the design of any soil to air system.

2.2 Soil-Air Heat Exchanger Models

Weisbecker et al. (1980) used a classical heat transfer model to analyze parameters of buried drainage tile to heat and cool ventilation air, for Minnesota conditions. Benefits to animal productivity as a result of cooling air in swine operations were examined. Their analysis found that ventilation accounted for 80% of the energy required in confinement housing. The authors estimated that summer cooling during breeding and gestation could provide an increase of 1 pig per litter.

Spengler and Stombaugh (1983) studied and reported on the theoretical and economic performance of Soil-Air Tempering systems for 10, 20, 30, and 40 sow swine farrowing houses in the Ohio area. The heat transfer analysis for the tempering system was performed using a finite element model. The economic analysis consisted of examining pipe lengths (10 m to 120 m using a 10 m interval), the number of pipes installed (1 to 24 pipes), and pipe diameter (0.10 m to 0.38 m). Based on this model the authors determined that the major influences on outlet temperature were airflow rate, pipe length and inlet temperature. Changes due to differing pipe diameters were found to be inconsequential.

Using an assumed system configuration, initial pipe, fan, heater and installation costs were estimated. Equivalent Annual Costs (EAC) were calculated assuming a 10 year economic life and an interest rate of 15%. Only one half of the initial costs were charged to the heating season, as they assumed that summer cooling would recover the other half of the installation costs. This assumption would only be valid if provisions for summer cooling are normally made for swine farrowing houses.

An EAC comparison between soil-air tempering and a conventional systems were made for winter and summer operation in a 20 sow swine farrowing house. The EAC for the conventional system was found to be $611 US through the winter, and $200 US for summer operation. It was not clear from the paper how the EAC for the conventional system was determined. Of the soil-air tempering systems, the most efficient system consisted of 7 - 30 m laterals with a 150 mm pipe diameter. This system had an EAC of $483 US, or an annual savings of $128. The authors arbitrarily selected a winter airflow rate 30% greater for the soil-air tempering system. This additional airflow was to account for moisture added to the air while passing through the buried pipes. By their calculations, use of a similar airflow rate would result in an EAC of $284 US. For summer operation, the authors simulated use of a tempering system with 9 laterals, with a length of 30 m. The EAC of this system was $300 US, $100 more than the conventional system.

Puri (1984) presented a finite element simulation of a Soil-Air Tempering System, and design curves for both system heating and cooling operations. His analysis evaluated the thermal performance of the system as a function of pipe diameter, pipe length, initial soil moisture content and temperature, and ambient air temperature. This model considered that heat diffusion occurs by both moisture migration and conduction. His model indicated that the key variables affecting Soil-Air Tempering are soil moisture, temperature and tube length. The results suggest that for soil moisture in the range of 20% to 30%, there was no significant difference in overall performance. A lower threshold value of 13.5% soil moisture content was suggest as the limit for useful operation. Puri also suggested that lateral spacing be a minimum of 8 diameters to minimize effects on soil moisture and temperature around neighbouring pipes.

Puri modeled his system for a typical 20 sow swine farrowing unit, for both cooling and heating operation. He used 10 - 0.1 m diameter pipes with a maximum pipe length of 12 m in the models. Performance curves were presented for both models giving outlet temperature as a function of pipe length and hours of operation. His system operated intermittently for a maximum duration of 12 hours. He stated that intermittent use of 2 hours or less at a time, resulted in the system approximating an infinite length model. That is outlet temperatures were maximized and neared the undisturbed soil temperature. Running the systems for intervals greater than 2 hours would exceed the soils regenerative capacity, and as a result the amount of heat recovered from the soil was reduced. The off cycles were assumed to be the same length as the on cycles.

The performance curves were then used to derive an Equivalent Annual Cost (EAC) comparison between the Soil-Air Tempering System and a conventional ventilation system for the swine farrowing operation. His calculations assumed that the ventilating year was divided equally between heating and cooling. The EAC for the Soil-Air Tempering system was $1,475 US, and the conventional system had an EAC of $1,850 US. For this particular system Puri showed an Equivalent Annual Worth of $375, with a system payback period of less than 5 years.

Lei et al. (1985) presented a theoretical study of variable factors contributing to the effectiveness of a Soil-Air Tempering system using finite element analysis. His model analyzed the mechanisms of changing thermal properties of the soil, latent heat exchange within the air and the soil, and soil moisture migration. The analysis described the relationships of key variables of the system to its effectiveness. The variables examined were airflow velocity, pipe length and diameter, system heat capacity, soil density and soil moisture content. Lei presented the mathematical model and the derivation of the differential equations necessary for the three dimensional transient heat transfer by conduction, convection and radiation. The results of the simulation are given as equations for the exchanger effectiveness, system heat capacity, and the temperature differential between inlet and outlet temperatures. The finite element model had close agreement with actual field data supplied from the experimental site located at Glenlea, Manitoba (Murray and Britton, 1985);(Murray, 1987).

2.3 Animal Environment

The most critical factor in a swine farrowing and nursery house is the environment. Environmental requirements of the young animals and sows differ greatly. To provide a productive environment the needs of the farrows, the nursery age animals and the sows must all be addressed. A number of scientific and government publications from around the world describe the key elements of temperature, humidity, and ventilation rate.

Temperature is a key factor for animal welfare and productivity which varies greatly depending upon the age of the animal. The general trend, which is shown in Table 2.1, is that optimum temperatures are inversely proportional to age. Temperatures beyond the

Table 2.1 Farrow, nursery and sow temperature requirements.

Temperature Range (° C) Farrows	Nursery	Sows	Author
25-35	24	16-19	Esmay & Dixon (1986)
24-35		18-21	Agriculture Canada (1976)
28		13-21	Agriculture Canada (1979)
21-30	21-27	21	Kennedy (1985)
	26.7		Curtis & Morris (1982)
	28		LeDividich et al. (1982)
28	15		Owen (1982)
	21-26.7		Vaughan, Holmes & Bell (1980)
24-35	24	21	Sainsbury & Sainsbury (1979)

ranges shown will lead to decreased productivity. The farrowing room, which must accommodate the sow and the farrows, is usually maintained at a temperature comfortable for the sow, approximately 21 ° C. Heated creeps provide a warm area for the piglets to rest. The creeps are usually maintained at 30 ° C or higher by using a heat lamp, or other heat source. Nursery areas are slightly easier to manage as all the animals require essentially the same thermal environment.

A number of authors have investigated the effects of extreme temperature on the performance of swine. Esmay and Dixon (1986) reported that feed efficiency decreases during extremely cold or hot environmental conditions. Temperatures below optimum leads to increased bodily heat loss, which in turn reduces feed efficiency. This is more critical for the younger animals which have a greater surface area to volume ratio, and hence greater convective losses. Farrows are born with a poorly developed ability to regulate body heat. By four days of age, the farrows temperature regulatory mechanisms are working properly (Agriculture Canada, 1976). Chilling young animals makes them weak and sluggish which may lead to losses by crushing or starvation. As well cold stressed pigs are more susceptible to infectious diseases, such as diarrhea (Feenstra, 1982; Agriculture Canada, 1976).

Feenstra (1982) determined that the lower air temperature limit of 4 week old weaned pigs was 24° C in uncovered pens, and 18° C in covered pens. At these temperatures good health, animal comfort, acceptable weight gains and feed conversions were maintained. At temperatures below these levels supplementary heat, in the form of heat lamps, was necessary. LeDividich, Noblet and Aumaitre (1982) state that the critical temperature for early weaned pigs raised in deck cages is approximately 28° C. With younger animals the maintenance of a constant environmental temperature is as important as the actual temperature itself. They found that a temperature fluctuation of $\pm$ 3° C resulted in a reduction of the daily gain by 9%, and an increase in the feed gain ratio by 7%. In older animals, diurnal temperature variations showed no reductions in performance when compared to constant temperature environments.

Hot temperatures, though having a lesser effect on young animals, drastically affect the performance of older animals. Heavier animals are less able to dissipate latent heat at higher temperatures. This phenomena leads to decreased feed efficiency, heat stress and weight loss. High temperatures may lead to higher rates of embryonic mortality in sows, resulting in smaller litters (Agriculture Canada, 1976; Kennedy, 1985).

The effects of relative humidity on animal performance are dependent on the accompanying air temperature, and are much harder to quantify. There does not appear to be an optimum relative humidity, but rather a range of acceptable values. Humidities beyond these ranges will lead to decreases in animal performance. LeDividich and Aumaitre (1977) suggest that the optimum operating range is between 40% and 60% relative humidity. Sainsbury and Sainsbury (1979) suggest an operating range of 30% to 60%. Combined with high air temperatures, relative humidity affects the performance of early weaned piglets (LeDividich & Aumaitre, 1977). High relative humidities reduces the amount of latent heat lost through respiration and evaporation, and increases the animals susceptibility to disease. Low relative humidities increase the animals susceptibility to disease and skin irritations.

Airflow rate is a significant factor of the performance of young animals. The minimum continuous airflow rate should be designed to remove all water vapour produced by the animals at a winter temperature that is exceeded 97.5% of the time (Esmay and Dixon, 1986). Higher airflow rates must be used to remove excess sensible heat during all seasons. This excess sensible heat includes radiant heat absorbed by the structure, heat produced by equipment and lights, and sensible heat produced by the animals. Esmay and Dixon (1986) have recommended airflow rates per animal as follows; for winter, 1 L/s; for spring and fall, 5 to 7 L/s; and for summer, 12 to 20 L/s. Recommended airflow velocities for given air temperature ranges as stated by Owen (1982), are shown in Table 2.2.

Farrows are extremely sensitive to draughts, which could lead to chilling. An increase in air velocity from 10 cm/s to 56 cm/s would be equivalent to a decrease in temperature of 4°C in still air for 2 kg pigs (LeDividich et al., 1982). Young pigs (2 to 3 weeks old) react unfavourably to air velocities over 12 cm/s (LeDividich and Aumaitre, 1977). LeDividich and Aumaitre found that draught free pigs grew 6% faster on 25% less feed than draught exposed pigs, when air temperature varied from 4° C to 19° C. Too little ventilation can lead to heat stagnation represented by lack of appetite, poor adaptability

Table 2.2 Suggested air velocities and critical temperatures for swine (Owen, 1982).

Type of Animal	Weight (kg)	Airflow (m/s)	Critical Temperatures (° C) Lower	Upper
Lactating Sow	145	0.3	9	27
Farrow	1.5	0.25	25	32
		0.6	28	34
Nursery/ Growers	25	0.3	10	24
		0.6	13	28
	80	0.3	9	26
		0.6	12	29

to weather changes, and susceptibility to chilling (Sainsbury and Sainsbury, 1979). LeDividich et al. (1982) found that increasing air velocity improves the performance of piglets through increased feed intake at temperatures above 31°C.

2.4 Energy Estimating

2.4.1 Winter Conditions

Steady heat flow through a wall is generally not possible, since that would imply that there is no change relative to time of the air temperature and motion, and the radiation to and from the surfaces of the inner and outer walls. Mackey and Wright (1943) suggest that steady flow was adequate for heating season predictions, though unsteady conditions had to prevail in the cases of intermittent heating or cooling. The basis of this assumption is that the difference between indoor and outdoor temperatures is very much greater than the daily range of outdoor temperature. As well the effect of solar insolation had a lesser proportionate effect during winter conditions.

The calculation of heating load is principally a function of heat losses due to transmission, air infiltration, and ventilation (ASHRAE, 1984). Weisbecker et al (1980) estimated that, for Minnesota conditions, 80% of the energy requirement for confinement housing is for the heating of ventilation air. The purpose of adding this ventilation is to remove excess moisture which may lead to sanitary and health problems within livestock confinement housing.

Esmay and Dixon (1986) recommend that housing for young animals should be designed to remove all the water vapour produced by the animals, at a winter temperature that is exceeded 97.5% of the time. The ASHRAE Fundamentals (1984) recommend using the

97.5% design temperature for minimum air exchange within a building. The recommended winter design temperatures for the Winnipeg area are given as -33°C for a 97.5% level, and -34°C for a 99% level (ASHRAE, 1985).

Transmission losses are calculated by equations which approximate Fourier's Equation for conductive heat transfer. The empirical formulae and tables are presented within ASHRAE (1984), for a variety of design heat transfer coefficients (U, $W/m^2{\cdot}K$), and building configurations. The formulae are straight forward and easy to use, requiring only a familiarity with local design conditions and material properties.

ASHRAE (1985) calculates infiltration heat losses for both the sensible and latent heat components of the incoming air. The sensible component is calculated as a function of the volumetric air flow (L/s), the inside air density (kg/m^3), the specific heat of air (.240 J/kg·K), and the difference in temperature of the inside and outside air (° C). The latent heat component is calculated as a function of the volumetric air flow (L/s), the inside air density (kg/m^2), the latent heat of vapourization of the inside air (2340 kJ/kg), and the difference in the absolute humidity ratios of the inside and outside air (kg/kg_{da}).

Ventilation using outdoor air must be taken into consideration. The calculation of its components of sensible and latent heat takes the same form as that for the infiltration rate. In the case of a livestock structure, the ventilation rate will generally exceed the infiltration rate. A well built structure may have an infiltration rate of 0.2 air changes per hour (ASHRAE, 1984). By comparison a minimum winter ventilation rate of 7.0 L/s/litter (VIDO, 1986) would translate into approximately 2.3 air changes per hour. For the purposes of a livestock structure with a high ventilation requirement, infiltration rate has been neglected, because it is difficult to calculate and has a lessening effect as ventilation rates increase.

As was the case with ASHRAE, Esmay and Dixon (1986) suggest partitioning the heat balance into sensible and latent components. The latent component results from the introduction of outside air to reduce the absolute humidity of the inside air. The introduction of this colder air will in turn reduce the sensible heat or enthalpy of the air.

The design heat load then is the amount of sensible heat which must be added to maintain a balance between the transmission losses, the ventilation losses, and the sensible heat gain from the internal processes or animals. Given the psychrometric relationships for air, the material properties of the building, and assuming a constant indoor temperature, and thus constant sensible heat production by the animals, the heat balance of a structure may be predicted as a function solely of the outside air temperature (Esmay and Dixon, 1986).

2.4.2 Summer Conditions

Methods of calculating the summer heat load are more complicated than for winter conditions. The effects of solar insolation, relative humidity, and vapour pressure must all be taken into consideration as functions of time. The influence of time would therefore imply that unsteady heat flow must be considered for calculations through the summer months. The procedure recommended by ASHRAE (1984) for the calculation of space heating load is the Total Equivalent Temperature Differential (TETD) Method. This method was first described by Mackey and Wright (1943, 1944), and later by Stewart (1948). The theory behind the procedure was that the summation of all heat gains may be converted into an instantaneous rate of heat transfer.

Stewart (1948) listed the variables involved in determining the heat gain through a structure as follows;

1. Outside air film coefficient of heat transfer, (h_o)
2. Inside air film coefficient of heat transfer, (h_i)
3. Thermal conductivity of the material, (k)
4. Density of the material, (p)
5. Specific heat of the material, (c)
5. Thickness of the material, (l)
6. Adsorptivity of outside surface for solar radiation, (α)
7. Incident solar radiation intensity as a function of time, (I)
8. Orientation of the wall or roof,
9. Room air temperature, (°C)
10. Outdoor temperature as a function of time, (°C)
11. Hours of operation of the cooling system.

With the equivalent temperature differential method, the heat flow rate is the product of the temperature gradient and the overall heat transfer coefficient (U, $W/m^2{\cdot}K$). This method accounted for both heat gain due to air temperature and solar insolation. Stewart noted several advantages when using this method.

1. All wall and roof structures could be summarized in representative classes.
2. The total sensible heat was easily calculated.
3. The calculations could be easily modified for specific structures and differing conditions.
4. Adjustments could be made easily for conditions other than those presented by the tables.

Mackey and Wright (1943) presented an empirical method of estimating unsteady heat flow using a form similar to that for steady heat flow. Their model assumed that temperature was cyclic over a 24 hour period, and the heat flow was through a single homogeneous material. The thermal storage of a material was a function of the material's characteristic time lag and thermal resistance ratio. The paper also presented the concept of an equivalent outdoor air temperature, which was later defined as the Sol-Air Temperature (Mackey and Wright, 1944). The authors introduced the concepts of the fundamental time lag (σ_0), the thermal resistance ratio (L_0), and the fundamental lag angle (ϕ_0). For fixed values of the indoor and outdoor air film heat transfer, these quantities were functions of the material's thermal conductivity, density, specific heat, and thickness. For their purposes, the authors used values of 22.7 W/m^2K (4 $Btu/hr{\cdot}ft^2{\cdot}F$) for the outdoor air film coefficient, and 9.4 W/m^2K (1.65 $Btu/hr{\cdot}ft^2{\cdot}F$) for the indoor air film coefficient.

The fundamental time lag (σ_0) was defined by equation 2.1. The derivation of the constant factor was not presented by the authors, so the constant value of equation 2.1 is only valid for English units. Mackey and Wright (1944) presented a graphical solution which proved to be more conducive for use with the SI system.

$$\sigma_0 = \left[\frac{.1309 \cdot p \cdot c}{k} \right]^{-\frac{1}{2}} \tag{2.1}$$

where; $k =$ Thermal conductivity, (W/mK) or (Btu/hr·ft·F)
$p =$ Material density, (kg/m^3) or (lb/ft^3)
$c =$ Specific heat, (kJ/kg·K) or (Btu/lb·F)

The thermal resistance ratio or decrement factor (L_0), a dimensionless quantity, was defined by Mackey and Wright (1944) as shown by equation (2.2).

$$L_0 = \left[\frac{2}{(F^2+G^2)} \right]^{-\frac{1}{2}} \tag{2.2}$$

where;

$$F = (\pi_1 + 1)\cdot c_1 + \frac{c_3}{\pi_3} + 2\cdot\pi_1\cdot\pi_3\cdot c_4$$

$$G = (\pi_1 + 1)\cdot c_2 + \frac{c_4}{\pi_3} + 2\cdot\pi_1\cdot\pi_3\cdot c_3$$

and

$$c_1 = \cos \pi_2 \cdot \cosh \pi_2 + \sin \pi_2 \cdot \sinh \pi_2$$
$$c_2 = \cos \pi_2 \cdot \cosh \pi_2 - \sin \pi_2 \cdot \sinh \pi_2$$
$$c_3 = \cos \pi_2 \cdot \cosh \pi_2$$
$$c_4 = \cos \pi_2 \cdot \sinh \pi_2$$

$$\pi_1 = h_i/h_o$$
$$\pi_2 = \sigma_0 l$$
$$\pi_3 = \frac{k\sigma_0}{h_i}$$

h_i = Indoor air film coefficient of heat transfer, (W/m^2K)
h_o = Outdoor air film coefficient of heat transfer, (W/m^2K)
l = Material thickness, (m)

The fundamental lag angle (ϕ_0) defined the length of time for the flow of heat to pass through the material. The fundamental lag angle was defined by equation 2.3.

$$\phi = \tan^{-1} \left[\frac{F - G}{F + G} \right] \tag{2.3}$$

Mackey and Wright (1944) defined the Sol-Air Temperature as;

> the temperature of the outdoor air which is contact with a shaded building surface, would give the same rate of heat transfer and the same temperature distribution through the material as exists with the actual dry bulb temperature of the outdoor air and the actual intensity of sol radiation incident upon that surface.

The temperature was defined for either steady or unsteady heat flow as;

$$t_e = t_a + \frac{\alpha \cdot I}{h_o} \tag{2.4}$$

where;
t_e = Sol-Air Temperature, (°C)
t_a = Dry Bulb Ambient Air Temperature, (°C)
α = Solar Adsorptivity of the Surface, dimensionless
I = Intensity of Solar Radiation, (W/m^2)

Stewart (1948) expanded upon the principles presented by Mackey and Wright to include composite structures. The concept of equivalent homogeneous construction was introduced as a simple homogeneous wall or roof which will have the same variation with time of indoor surface temperature as does the actual composite wall or roof under identical ambient conditions.

The fundamental time lag (hours) was first determined for each distinct layer by calculating the equivalent thermal conductance $(k/l)_e$ and kpc_e, and the use of Figure 2 of Mackey and Wright (1944). The sum of the time lags of the layers was defined as the time lag of the composite wall.

The equivalent thermal conductance $(k/l)_e$ for the combined wall or roof was determined by equation 2.5.

$$\left[\frac{k}{l}\right]_e = \frac{1}{\frac{l_1}{k_1} + \frac{l_2}{k_2} + \ldots + \frac{l_n}{k_n}} \tag{2.5}$$

Using the equivalent thermal conductance of the structure and the combined, time lag, the equivalent (kpc) or $(kpc)_e$ was determined from Figure 1 of Mackey and Wright (1944).

Using $(k/l)_e$ and $(kpc)_e$ the fundamental decrement factor L_1 was obtained from Figure 1 of Mackey and Wright (1944). The second harmonic decrement factor L_2 was then obtained from the Figure 1 using $(k/L)_e$ and 2 times $(kpc)_e$.

The equivalent decrement factor L_e could then be obtained by using the fundamental and second harmonic decrement factors and the following orientation equations;

<u>Orientation</u>	L_e
Horizontal Roof	$0.7L_1 + 0.3L_2$
North Wall	$0.7L_1 + 0.3L_2$
East Wall	$0.2L_1 + 0.8L_2$
South Wall	$0.6L_1 + 0.4L_2$
West Wall	$0.4L_1 + 0.6L_2$

With the material characteristics defined, the predicted surface temperature at a time $(\Theta+\phi/15)$ was given by equation 2.6.

$$t_o(\Theta+\phi/15) \;=\; t_m+L_e\cdot[t_e(\Theta) - t_m] \tag{2.6}$$

where;
- t_o = Surface temperature, (°C)
- Θ = Current time, (hours)
- ϕ = Time Lag, (degrees)
- t_m = Average indoor temperature, (°C)
- t_e = Sol-Air temperature, (°C)

The rate of heat flow to the room was then simply the heat transfer from the inside surface of the wall to the room, and was given by equation 2.7.

$$q = h_i\cdot A\cdot(t_o - t_m) \tag{2.7}$$

3.0 METHODS

3.1 Control System

3.1.1 Temperature Database

The basis of the model was a database of temperatures collected by Murray (1987), using a pilot soil-air tempering system at the University of Manitoba (Section 2.1). Each lateral had a 30 m section of pipe buried at a depth of 3.0 m. Temperature measurements of the airflow were made at stations 1 m, 5 m, 10 m, 15 m, 20 m, and 29 m from the inlet end of each lateral. Specific thermocouple data were selected from the database for two periods; winter and summer. These data were from the thermocouples measuring the temperatures at the 20 m and 29 m lengths along the axis of each pipe, and the ambient temperature thermocouples. Results for Pipe 2 (150 mm diameter, Airflow 0.10 m^3/s) were not available due to system malfunction. The remaining three pipes were modeled for; a winter period from January 2, 1985 to March ,1985; and a summer period from May 27, 1985 to September 3, 1985. The three modeled pipes had the following diameters and airflow rates:

Pipe 1 - 150 mm diameter, Airflow rate 0.05 m^3/s

Pipe 3 - 250 mm diameter, Airflow rate 0.10 m^3/s

Pipe 4 - 250 mm diameter, Airflow rate 0.05 m^3/s

3.1.2 Animal Parameters

The modeled system was a swine farrowing and nursery operation, because these operations typically have high energy demands due to ventilation and high room temperatures. A system such as this would have the greatest benefit from reduced heating and ventilating costs. The operation was sized for approximately 250 breeding sows with an average of 2.1 litters/sow/year. Average litter size was taken to be 8.5 piglets/litter. Table 3.1 shows the animal parameters used by the models.

Table 3.1 Animal parameters assumed for the model.

Parameter	Farrowing	Nursery
Temperature	21° C	27° C
Relative humidity	75%	75%
Ventilation Rate		
Winter	Variable	Variable
Summer	110 L/s/litter	15 L/s/pig
Heat Production		
Sensible	248 W/litter	41.7 W/pig
Latent	394 W/litter	54.3 W/pig

Note: Winter Ventilation Rates for moisture control only.

In the case of the farrowing rooms, all of the rooms will be maintained at a design temperature of 21° C. The nursery rooms required temperature control as a function of

piglet age. The recommended ranges were between 21° C and 28° C, with the required temperature decreasing with increased piglet age. It was assumed that the design temperature for the rooms was
27° C.

The relative humidity level was arbitrarily selected as 75%, because the European operating ranges of 30% to 60% (LeDividich and Aumaitre, 1977);(Sainsbury and Sainsbury, 1979) seemed low. Maintaining these levels during a cold Canadian winter would result in unnecessarily high ventilation and resultant heating costs.

The winter ventilation rate was not specified, because it was calculated by the programs as a function of ambient and outlet temperatures. The summer maximum ventilation rates were specified, because the tempering systems were inadequate for cooling. The maximum summer ventilation rate for the farrows was 110 L/s/litter (Agriculture Canada, 1981 as cited by VIDO, 1986). The maximum summer ventilation rate for the nursery animals was 15 L/s/pig (Kennedy, 1985).

The animal heat and moisture production rates represent values typical of the oldest animals within the rooms. The farrowing values were interpolated from values presented by Esmay and Dixon (1986) and the ASAE (1984) for 5 week old litters. Nursery production rates were for 22.73 kg animals raised on a partially slotted floor at 26.7°C (Esmay and Dixon, 1986).

Based on recommendations from VIDO (1986), the farrowing rooms were equipped with 1500 mm x 2100 mm, side creep pens. Each pen was equipped with a 250 W heat lamp for the creep area. For a 250 sow operation, approximately 60 farrowing pens were required. The nursery pens were sized to handle 2 litters/pen, with a space allowance of 0.2 to 0.3 m^2/pig (ASAE, 1984). Each nursery pen was 1200 mm x 3600 mm.

All parameters involving the animals environment were held constant. This meant that the heat and moisture production of the animals were viewed as constants. The animals heat and moisture production was defined for both environmental temperature and piglet age, but to simplify the calculations and to isolate variables they were held constant.

3.1.3 The Model Structure

The Structure was intended to house both the farrowing and the nursery operations, as recommended by VIDO (1986). The planned layout and dimensions of the structure were as shown in Figure 3.1. The interior space was laid out along the structure's centre hall with the farrowing operation along one side of the building, and the nursery operation on the other. There

was an unused space which was designated as storage/office space, and had no prescribed function. In further calculations it was considered a part of the hallway, that is no temperature control. The two animal areas were both subdivided into five rooms. Each

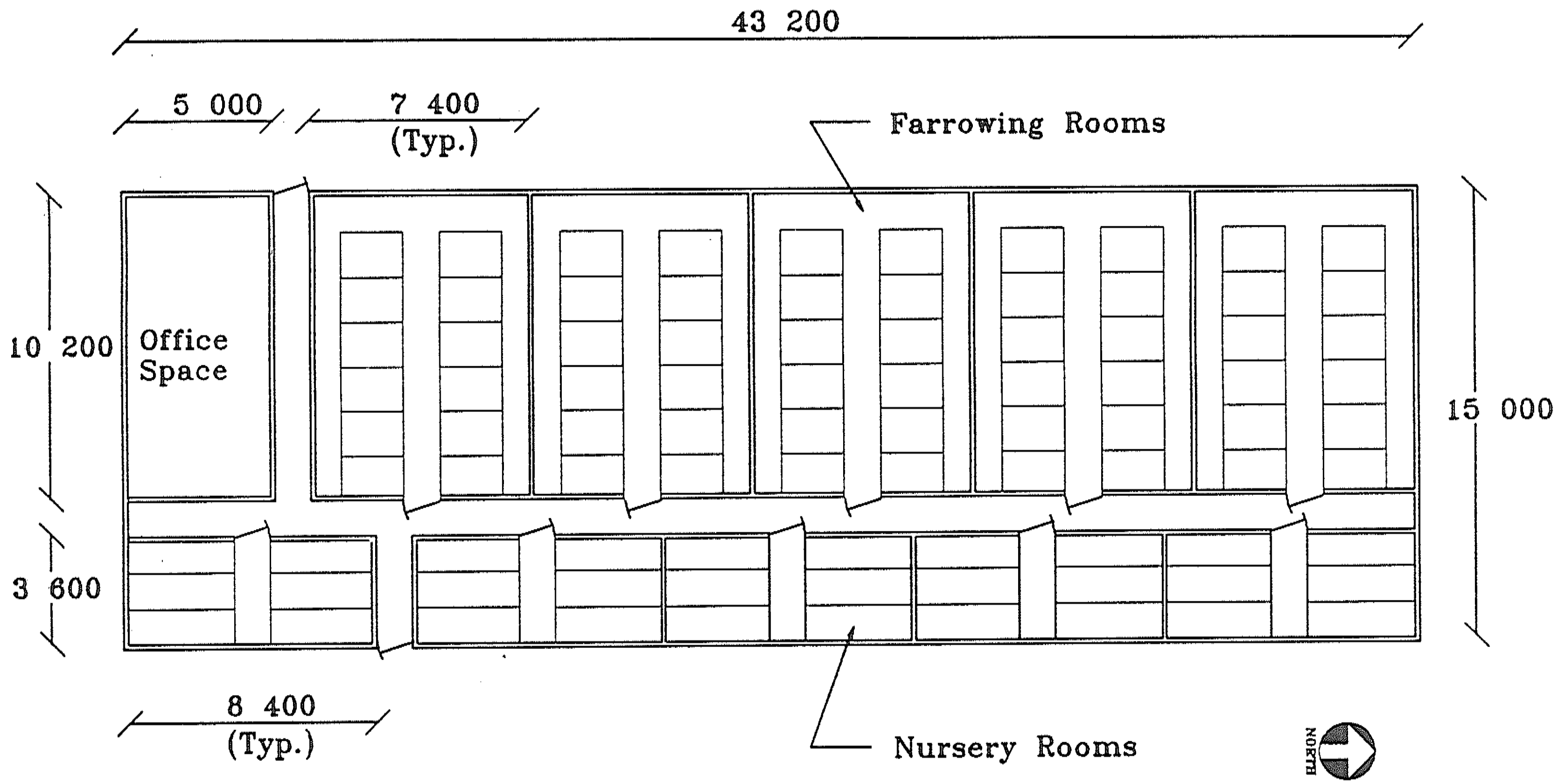


Figure 3.1 Farrowing–Nursery Structure Layout

of these rooms had the ability to modulate its environment within a limited range. The farrowing and nursery rooms were assumed to be equipped with supplemental heat.

The structure was assumed to be a typical stud wall construction, with a gabled roof and a slab on grade foundation. The roof slope was assumed to be 18.4°. The side wall height was assumed to be 2.5 m. The exterior walls and ceiling were assumed to have a design thermal resistance value (R_{SI}) of 3.5 m^2K/W, and the foundation an R_{SI} value of 1.4 m^2K/W. These values are within the insulation ranges recommended by VIDO (1986) and Kennedy (1985). The interior partitions were not provided with any insulation. Table 3.2 shows the thermal properties of the various building sections used within the program.

The building was oriented in a north-south direction along its longitudinal axis. It was further assumed that the farrowing rooms would be along the east wall of the structure. This orientation was chosen as it minimized the wall surface area facing due south and it minimized the calculations necessary for the roof.

The overall coefficients of heat transfer (U, $W/m^2\cdot K$) represent the thermal conductivity of the composite structure with surface effects taken into consideration. The U value which represents the structure's conductivity is approximately equal to the reciprocal of the R_{SI} value or thermal resistance of the structure. The actual values of U differ from the reciprocal of the design R_{SI} values because the actual construction material values

Table 3.2 Thermal properties of the model structure.

Section	U W/m_2K	$(k/l)_e$ Btu/hrft^{2o}F	$(kpc)_e$	Time Lag (h)	L_e
Roof	2.783	0.89	0.9	0.43	0.30
Gable Ends					
North	3.543	1.47	2.3	0.47	3.97
South	3.543	1.47	2.3	0.47	3.96
Ceiling	0.246	0.046	0.06	4.23	0.017
Exterior Walls					
North	0.255	0.44	3.0	4.43	0.14
East	0.255	0.44	3.0	4.43	0.12
South	0.255	0.44	3.0	4.43	0.13
West	0.255	0.44	3.0	4.43	0.13
Interior Walls	1.793	0.52	3.6	0.40	0.21
Foundation					
North	0.607	0.12	0.3	3.84	0.043
East	0.607	0.12	0.3	3.84	0.025
South	0.607	0.12	0.3	3.84	0.039
West	0.607	0.12	0.3	3.84	0.032

Note:
1. Units for $(k/l)_e$ are in Imperial units to maintain consistency with Stewart (1948).
2. Units for $(kpc)_e$ are Btu/hr·ft^2·°F * lb/ft^3 * Btu/lb·°F to maintain consistency with Stewart (1948).
3. The Equivalent Decrement Factor (L_e) is dimensionless.

were used, and the effects of convective heat transfer where not included in the R_{SI}. The U values are calculated using the following form as presented by ASHRAE (1985):

$$U = \frac{1}{1\backslash h_o + l_1/k_1 + l_2/k_2 + \ldots + 1/h_i}$$

where; U = Coefficient of Heat Transfer, (W/m^2K)
h_o = Outside Air Film Coefficient of Heat Transfer = 22.70 (W/m^2K)
l_n = Thickness of Material "n", (m)
k_n = Thermal Conductivity of Material "n", ($W/m^2K/mm$)
h_i = Inside Air Film Coefficient of Heat Transfer = 9.36 (W/m^2K)

Ventilation for the building was provided by a centrifugal fan connected to the soil-air tempering system, and through axial flow exhaust fans mounted in the outer wall of each room. As a result, the system was to operate at negative pressures with air drawn through the air tempering system, and then drawn into each of the rooms as required. Air inlets into the rooms delivered air from the hallway. The hallway acted as the main air duct or plenum, in which further tempering of the air was to be accomplished through the addition of heat by forced air units. Air inlets were not specified, because they are dependent upon the actual ventilation requirements and equipment selection.

All of the parameters pertinent to the model have been defined, with the exception of ambient temperature and heat surplus/deficiency. Thus the necessity of either heating or cooling can be defined simply as a function of the outside temperature. Initially the heat balance neglected the effects of ventilation on the system. After the initial heat balance approximation was calculated it became necessary to define the required ventilation for all conditions, whether it be for temperature or moisture control.

3.1.4 The Model System

The modeled systems represented the available databases for each of the three pipes in the pilot system. Soil-air tempering systems were modeled for each of the pipes using both a 20 m and a 29 m lateral length. Pipes were modeled to determine which combination of airflow, lateral length and pipe diameter would yield the most cost effective system. The two layouts for the systems are shown in Figures 3.2 and 3.3.

The operating parameters of airflow rate, lateral length and pipe diameter were kept the same as the pilot system, so as not to invalidate the basis of the data. It was assumed that the airflow through the pipes was constant at all times. The system was to be constructed of polyvinyl chloride (PVC) pipe to maintain the same heat transfer characteristics between the soil and the pipes as the pilot system. No consideration was given for heat transfer as the air passed through the buried header or through the ducting to the structure.

The proposed soil-air tempering systems were laid out to minimize the costs of construction and materials, while maintaining the specified lengths and spacings indicated by Murray (1987). Two laterals were laid in each trench with a 2.0 m spacing between pipes. Trenches were located 7.0 m on centre, to maintain slope stability while excavating. The pipes were buried to a minimum depth of 3.0 m, with a -0.5% slope towards the header for drainage. Inlet risers consisted of a 4.2 m long piece of PVC pipe

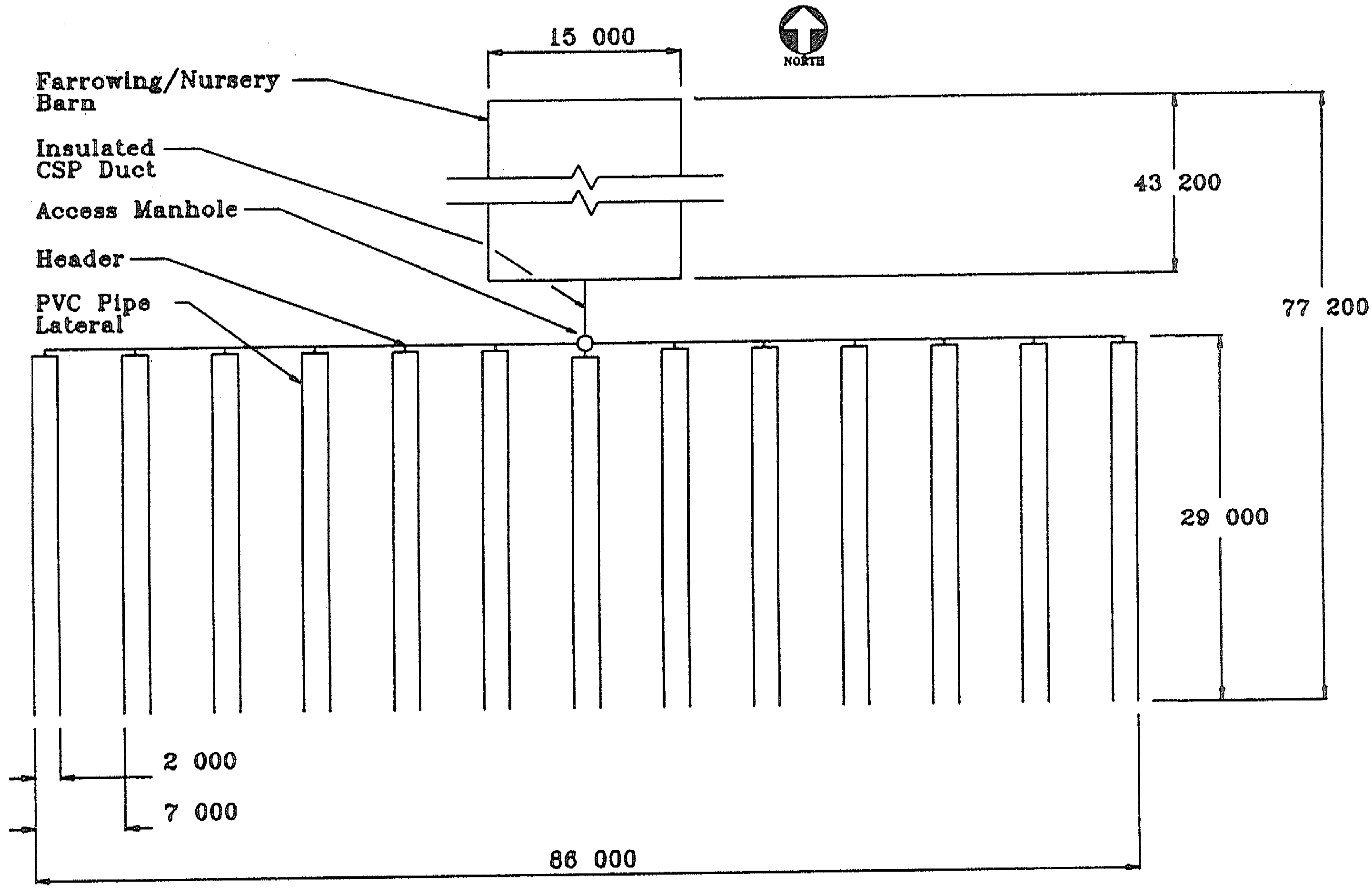


Figure 3.2 General layout of soil-air tempering for pipes 1 and 4. Low airflow rate (50 L/s).

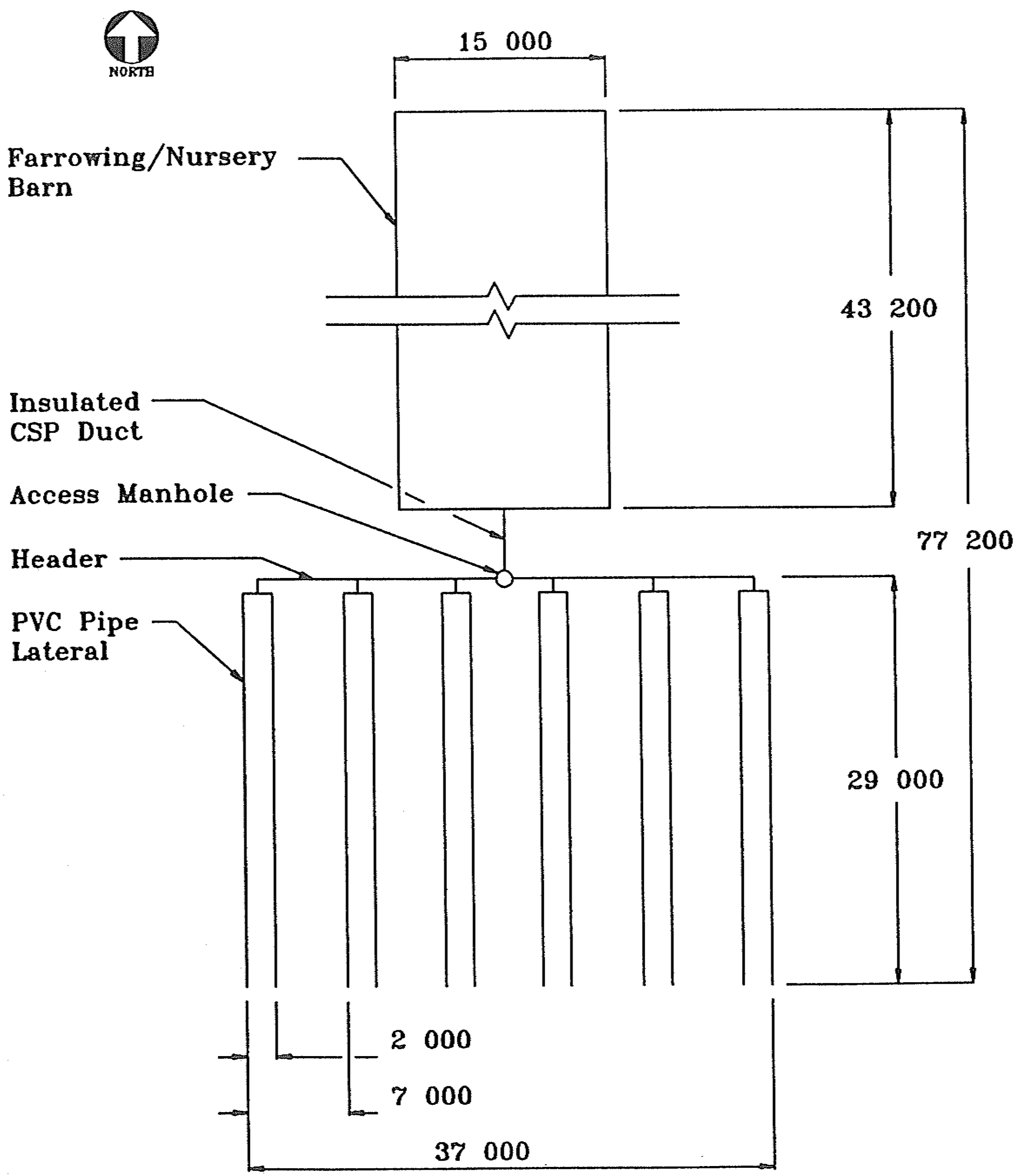
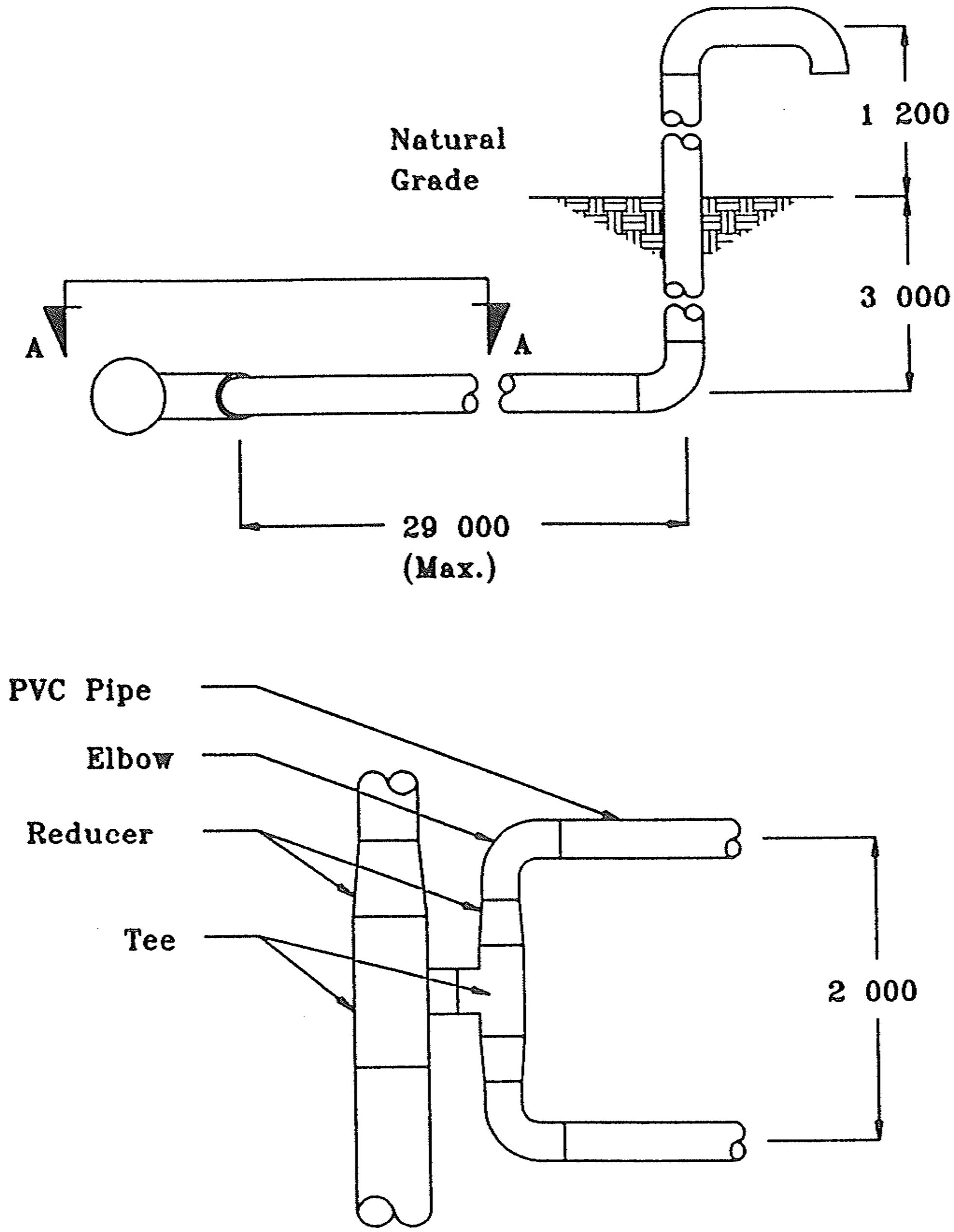


Figure 3.3 General layout of soil-air tempering system for pipe 3. High airflow rate (100 L/s).

with a gooseneck at the surface, to prevent entry of rainwater and debris. Figure 3.4 shows typical connection details for the PVC laterals.

The header and the duct work were included in the cost analysis of the systems, but were not considered in the heat transfer calculations. The header was constructed of PVC pipe, which increased in diameter as the airflow increased with the addition of laterals. All duct work consisted of corrugated steel pipe (CSP) for both the access manhole and the horizontal air duct. An access manhole was constructed of 1200 mm CSP pipe to permit access for service and the installation of a sump pump. The manhole extended 1.0 m deeper than the grade of the header to provide a sump. Horizontal ducting into the structure consisted of 750 mm CSP pipe, contained within an insulated enclosure. Figure 3.5 shows the proposed section of the header and duct work.



Section A–A

Figure 3.4 Soil–air tempering lateral and connection detail.

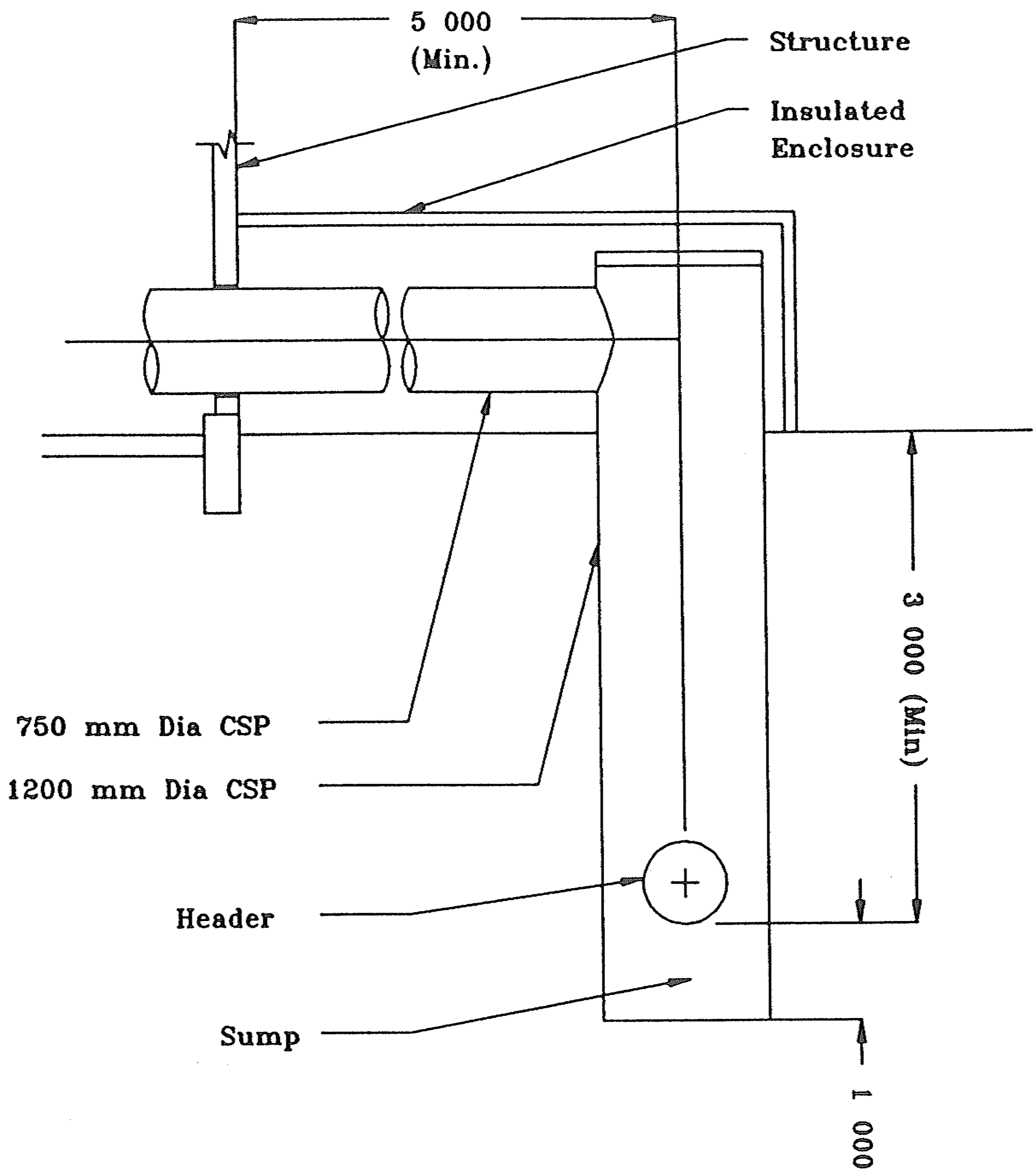


Figure 3.5 Collection header and duct work detail.

3.2 WINTER SIMULATION

The winter simulation was performed using a series of BASIC programs which used the modified temperature data from the data acquisition system employed by Murray (1987), to model against a simulated farrowing/nursery barn over the winter period. Program output was designed to yield the heat balance and required air flows for each temperature record. The simulation was performed using a number of routines which were developed to process the data and yield specific system parameters. The following paragraphs will describe the flow paths of the programs and the specific routines. Program listings for the winter simulations are given in Appendix A.

The winter simulation made use of 5 programs, which are listed in Table 3.3. The number of programs was necessary due to system constraints and a means of formatting the output data. The programs were written in BASIC which limited the amount of information that was stored for each record. In some instances, the programs were run to just recalculate and store a specific variable. General heat and mass transfer calculations were similar in each of the programs, the differences between the programs occurred due to specific tasks and output of each of the programs.

The purpose of the initial program, WINMOD1, was to determine the number of laterals necessary for each pipe configuration. The output yielded the number of laterals necessary to deliver sufficient airflow to the model structure for each temperature record.

The program assumed that the airflow rate through individual laterals was constant, and that there was always sufficient tempered airflow to meet the current ventilation demand. Results of this program were analyzed at the winter design temperature to determine the final size of the modeled system.

Table 3.3 Winter Simulation Programs

Program Name	Ventilation Model	Lateral Lengths	Output Data
WINMOD1	Tempered	29 m	Heat Balance, Number of Laterals
WINMOD2	Ambient	N/A	Heat Balance, Mass Flow Rate, Airflow Rate
WINMOD3	Tempered	29 m	Heat Balance, Mass Flow Rate, Airflow Rate, Number of Laterals
WINMOD4	Tempered	20 m	Heat Balance, Number of Laterals
WINMOD5	Tempered	20/29 m	Heat Balance, Total Airflow, Tempered Airflow, Ambient Airflow

WINMOD2 simulated the model structure with a conventional, ambient air ventilation system. The purpose of this program was to calculate the heat balance, mass flow and airflow rates of the conventional ventilation system. The output of this program was the basis for comparison with the soil-air tempering scenarios.

WINMOD3, a refinement of WINMOD1, was designed to yield the mass flow and airflow rates of the tempering systems. The program modelled each of the pipe configurations using 29 m lateral lengths. The purpose of the program was to yield the number of laterals necessary and the ventilation requirements of each of the tempering systems. The calculation of the air and mass flow rates were the principal objectives of this program.

WINMOD4 was identical to WINMOD3 with the exception that it modelled 20 m lateral lengths for each configuration. The output of the program yielded the number of laterals necessary and the ventilation and mass flow rates for each of the tempering systems.

WINMOD5 was the final refinement to the winter simulation. The purpose of this program was to determine the heat balance and the ventilation requirements for a given number of laterals. The program was designed to model any given pipe configuration, lateral length, and number of laterals. The program simulated an actual system with its physical limitations and the necessity of using ambient air to supplement the soil-air tempering systems. Using a limited number of laterals created occasions when the tempering system had insufficient capacity to meet the moisture removal needs.

3.2.1 Main Program

The main program was a simple routine which was designed to initiate variables, request operating selections, open data and output files, and call the subroutines. The principal operations of the program were performed within specific subroutines to simplify the structure of the program and reduce the computation time. Figure 3.6 presents a simple flow chart of the winter simulation programs. The program was not written to be user interactive, but rather as a bulk data processor. Proper operation of the program relied on the correct location and naming conventions for subdirectories and files. Further to this the model of the structure and the animal parameters were fixed in order to simplify the operation of the program.

Operator inputs were restricted to the pipe to be modelled and in the case of WINMOD5 the total number of laterals within the system. The program modeled a selected pipe configuration using its data base. The available data files were for pipes 1, 3, or 4 (Murray, 1987), and their respective pipe diameters and airflow rates. The choice of program, WINMOD3 or WINMOD4, allowed for the simulation of either 20 m or 29 m pipe lengths.

Input data files were derived directly from the original output files of the data acquisition system (Murray, 1987). The files had simply been condensed to include the date and time of record, thermocouple measurements along the longitudinal axis of the pipes and

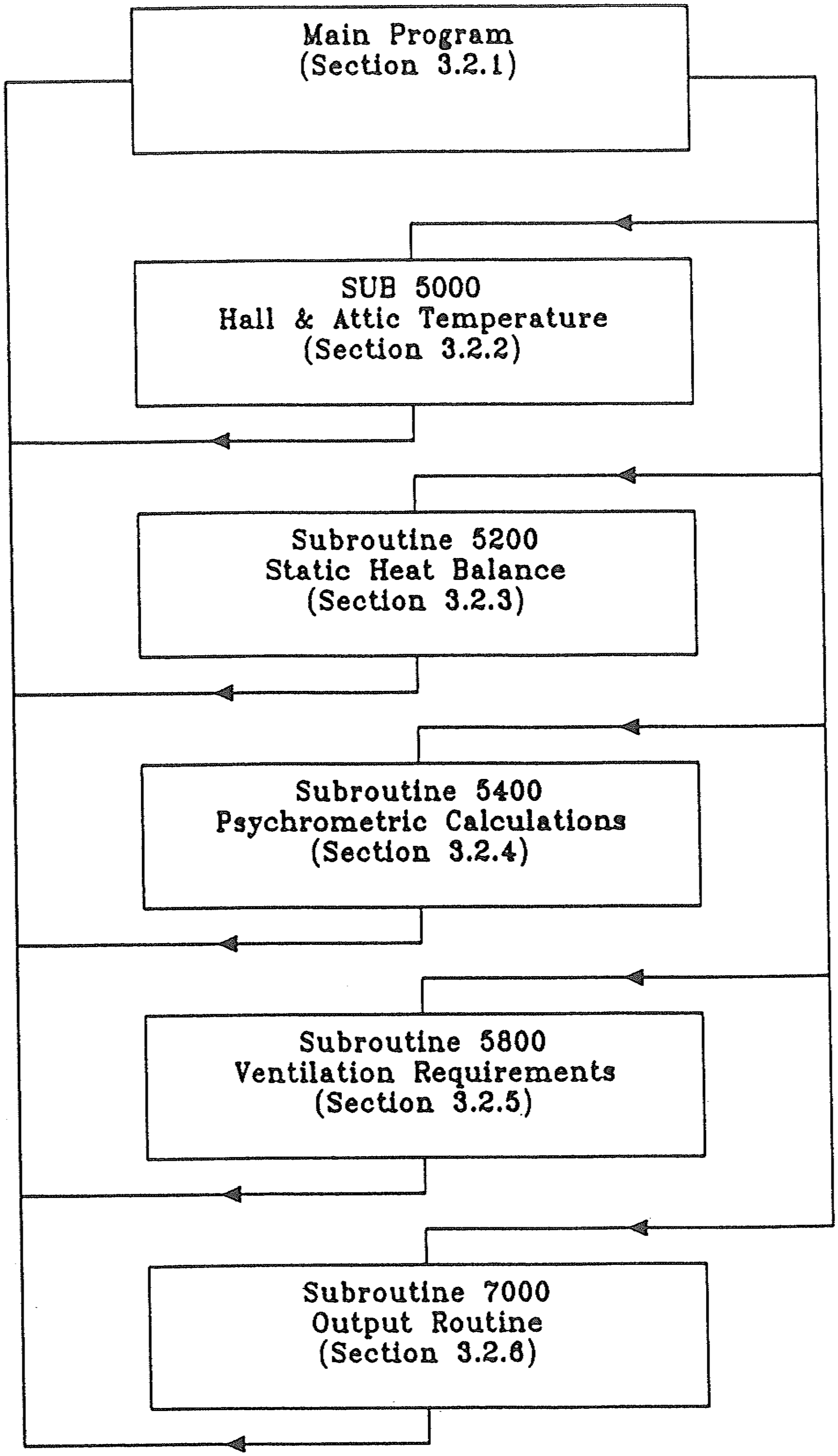


Figure 3.6 – Winter simulation program major subroutines and flowchart

the ambient temperature records. The data files first entry gave the total number of records contained within the file. The program retrieved this value and set its own limits of operation.

Output files were random ASCII text files and were named according to the user selections (modeled pipe and number of laterals) and the sequential number of the corresponding data file.

Output was formatted as text and was retrieved either with a BASIC routine or by using a standard spreadsheet program.

3.2.2 Calculation of Hall and Attic Temperatures

These calculations were contained within a subroutine which used an iterative procedure to calculate the hall and attic temperatures for the current data record. The calculations were based on the previous temperatures of the hall and attic, and the current temperature records. The process iterated until the difference in temperature between successive iterations was less than ±0.1° C.

The hall and attic temperatures were calculated by equating the heat flows between these areas and each of the other control volumes. The identified control volumes were the farrowing rooms, the nursery rooms, the hall, the attic, and the outside. For each temperature record, farrowing rooms, nursery rooms, and ambient conditions were all assumed to be constant. Since the hall and attic had a mutual interface, their temperatures have an implicit dependency on one another. Therefore an iterative process was required to obtain convergence of the temperature values for each area. The iterations continued until there was a net difference of less than ±0.1° C between successive iterations for both hall and attic temperatures. The form of the equations for estimating temperatures in adjacent unheated spaces was obtained from the ASHRAE Fundamentals (1985), and was as follows;

$$t_u = \frac{t_i\,(A_1U_1 + A_2U_2 + \ldots) + t_o\,(K \cdot V_o + A_aU_a + A_bU_b + \ldots)}{(A_1U_1 + A_2U_2 + \ldots) + (K \cdot V_o + A_aU_a + A_bU_b + \ldots)} \qquad (3.1)$$

where; t_u = Temperature in the unheated space, (° C)

t_i = Temperature in adjacent heated spaces, (° C)

t_o = Outdoor temperature, (° C)

$A_1, A_2, \ldots$ = Surface areas adjacent to heated space, (m^2)

$A_a, A_b, \ldots$ = Surface areas adjacent to outdoors, (m^2)

U_n = Heat transfer coefficients of surfaces, W/m_2K

K = 1200

V_o = Rate of airflow into the unheated space, (L/s)

WINMOD5 had an additional step in the calculation process due to the addition of ambient air for ventilation. After the ventilation quantities had been determined, the program returned to this subroutine to recalculate the hall and attic temperatures. The other programs did not mix ambient and tempered air, which made this step unnecessary.

3.2.3 Calculation of Static Heat Balance

This subroutine calculated the sensible heat balances between the various model control volumes, without ventilation. Control volumes for the static heat balances were the same as those for the previous subroutine. The following is a list of the heat flows calculated;

- Q1 Heat transfer between building and ambient conditions,
- Q2 Heat transfer between interior rooms and hall,
- Q3 Heat transfer between each room and the attic,
- Q4 Heat transfer between the floor and each of the rooms,
- Q5 Sensible heat given off by the animals,
- Q6 Total heat balance for each of the rooms.

The sensible heat value of Q5 was incremented by 250 W per sow for heat lamps in the farrowing area.

For the given temperature record, the program had now collected all of the temperature variables, or had an initial estimated value. This allowed the calculation of the simple, steady state heat flow between the control volumes. The heat transfer calculations were based upon Fourier's Equation, which was given as:

$$q = A \cdot U \cdot (t_1 - t_2) \tag{3.2}$$

where; q = Instantaneous heat flow, (W)

A = Surface area, (m^2)

U = Coefficient of heat transfer, (W/m_2K)

The difference in temperatures was the temperature gradient between any two control volumes.

3.2.4 Psychrometric Calculations

This subroutine calculated the psychrometric properties of the control volumes. The psychrometric properties of the air for each of the rooms, the hall, ambient conditions and the pipe outlet were calculated. The following properties were considered:

Saturation Vapour Pressure, P_S
Actual Vapour Pressure, P_W
Absolute Humidity, W
Sensible Heat, H_S
Latent Heat, H_L
Total Enthalpy, H_T
Specific Volume, v
Air Densities, p

These properties were stored for later use to calculate the required ventilation rates.

The first quantities calculated were the saturation vapour pressures (P_S). The choice of equation was dependent on the temperature of the air. Muir (1982) gave the following equations;

For $233\ K \leq T \leq 273\ K$ (3.3)

$$\ln(P_s) = 24.28 - 6238/T - .3444 \ln(T)$$

For $273\ K \leq T \leq 393\ K$ (3.4)

$$\ln(P_s) = 89.63 - 7512/T + .024T - 1.165E\text{-}5T^2 - 1.281E\text{-}8T^3 + 2.1E\text{-}11T^4$$

Where; T = Absolute Temperature, (K)

The program assumed that the relative humidities within each of the control volumes remained constant. In the cases of the farrowing and nursery rooms, the relative humidity was the controlling parameter of the ventilation system.

Vapour pressure and absolute humidity for each of the control volumes was then calculated. Vapour pressure was calculated as the product of the relative humidity and the saturation vapour pressure. Absolute humidities were calculated using the following equation from Muir (1982);

$$W = 0.622 \cdot \frac{P_w}{P_A - P_w} \tag{3.5}$$

where; W = Absolute humidity, (kg of H_2O/kg of dry air)
P_w = Vapour Pressure, (kPa)
P_A = Atmospheric Pressure, (kPa)

Two assumptions were made with respect to the calculation of absolute humidity and vapour pressure. For the purposes of these calculations, the atmospheric pressure was taken as the value at sea level (101.325 kPa), and held constant. The programs assumed that the absolute humidity and the vapour pressure of the tempered air and the hall were equivalent to ambient conditions.

Sensible and latent heat components of the control volumes were then calculated. The , which were valid for temperatures between -50°C and 110°C, were obtained from Muir (1982).

$$H_s = c_a \cdot T, \quad \text{and} \tag{3.6}$$

$$H_l = W\,(L + c_w \cdot T) \tag{3.7}$$

where;

H_s = Sensible heat, (kJ/kg)
c_a = Specific heat of dry air, (kJ/kg·K)
= 1.01 (kJ/kg·K)
H_l = Latent heat, (kJ/kg)
L = Latent heat of evaporation, (kJ/kg)
= 2501 kJ/kg
c_w = Specific heat of water vapour, (kJ/kg·K)
= 1.78 (kJ/kg·K)

The specific volumes and the density of the air volumes were then calculated using equations from Muir (1982).

$$v = \frac{R_a\,(t + 273)}{(P_A - P_w)} \tag{3.8}$$

where; v = specific volume, (m^3/kg)
R_a = Ideal gas constant for dry air, ($Pa{\cdot}m^3/kg{\cdot}K$)
= 287 ($Pa{\cdot}m^3/kg{\cdot}K$)

The atmospheric and vapour pressure were reduced to units of Pascals (Pa).

The density of the dry air was determined as;

$$p = \frac{1 + W}{v} \tag{3.9}$$

where; p = density, (kg/m^3)

3.2.5 Calculation of Ventilation Requirements

This subroutine calculated the required ventilation rates for each of the rooms within the building. The calculation was based upon the latent heat balances within the building. As an initial approximation it was assumed that ventilation was to be required for moisture control not temperature control. Esmay and Dixon (1986) suggest that the minimum continuous ventilation rate be designed to remove all of the water vapour produced during winter conditions. It was therefore assumed that the entire latent production of the animals was to be the basis for determining the ventilation requirement. It was also assumed that tempered air would be the primary source of ventilation, and ambient air would be used only as make up air. The following paragraphs describe the procedures used to determine the quantity and effects of the ventilation air.

The first step was to calculate the difference in latent energy between the hall and each of the rooms, to yield the moisture removal potential of the ventilating air. The volume of air required was then calculated based on the latent heat production rates of the animals (ASAE, 1984). Latent heat productions of the animals were assumed to be constant, and were; 394 W per litter for the farrowing rooms; and 54.3 W per animal in the nursery. The following equation was used to calculate the required airflows for moisture control;

$$V(n) = \frac{E_L(n) * N}{(H_L(n) - H_L(3)) * p} \tag{3.10}$$

where;
$V(n)$ = Required airflow for room, (m^3/s)
$E_L(n)$ = Latent Heat production of animals, (kW)
N = Number of animals
$H_L(n)$ = Latent heat content of room, (kJ/kg)
$H_L(3)$ = Latent heat content of hall, (kJ/kg)
p = Density of hall air, (kg/m^3)

Sensible heat loss due to the addition of this ventilation air was then calculated for each of the rooms. The sensible heat of each of the rooms, and the hall were calculated using equation 3.6. The sensible heat loss for each room was calculated as the difference in sensible heat between the hall and the room in question, multiplied by the mass flow rate of the required ventilation. Heat loss due to ventilation was then added to the instantaneous static heat loss Q6(n) to yield the total instantaneous heat loss Q7(n) for each of the rooms. Q6(n) is the sensible heat balance taking into consideration convective and conductive losses, and animal heat production. Q7(n) represented the total sensible heat loss (or gain) experienced by each of the rooms for the current temperature record. Q7 for the hall was assumed to be zero. Q7(n) was calculated as follows;

$$Q7(n) = Q6(n) - [H_s(n) - H_s(3)] * m(n) \qquad \text{for } n = 1 \text{ to } 3 \quad (3.11)$$

where;

$Q7(n)$ = Total instantaneous heat loss, (kW)
$Q6(n)$ = Instantaneous static heat loss, (kW)
$H_s(n)$ = Sensible heat content of room, (kJ/kg)
$H_s(3)$ = Sensible heat content of hall, (kJ/kg)
$m(n)$ = Required mass flow for each room, kg/s
n = Room number; 1 = Farrowing; 2 = Nursery; 3 = Hall

Mass flow rates were calculated so that direct comparisons could be made between the air volumes at differing temperatures.

The volume of tempered air required was then calculated based on the previously calculated mass flow rates. Total mass of air required in the rooms was equivalent to the mass of tempered and/or ambient air required. Once the volume of air had been determined, the number of laterals required was simply the air volume divided by the airflow rate through each pipe. It was assumed that the airflow rate within the pipes was constant, as a means of simplifying the procedure.

In the specific case of WINMOD5, the number of laterals had been specified. If the number of required laterals was less than or equivalent to the number of laterals initially specified in the model, the routine continued with the calculation of the sensible heat loss due to the addition of the ventilation air. If the number of required laterals was greater than that originally provided, the routine proceeded with calculation of the tempering system maximums. Maximum airflow and mass flow rates were recalculated for the current ambient conditions. In this case there would be insufficient airflow for proper moisture control within the structure. The difference was made up with ambient air. The routine then continued with the calculation of the sensible heat balance based on a mixture of ambient and tempered air.

The rate of heat transfer due to the addition of the tempered and ambient ventilation air (Q8) was the final component of sensible heat. The heat flow was calculated as the product of the differences in enthalpy multiplied by the mass flow rate of the ventilation air.

$$Q8 = m_T * (H_s(4)-H_s(3)) + m_A * (H_s(0)-H_s(3)) \qquad (3.12)$$

where;
$Q8$ = Sensible Heat flow due to ventilation, (W)
m_T = Mass flow tempered air, (kg/s)
m_A = Mass flow ambient air, (kg/s)
$H_s(4)$ = Sensible heat tempered air, (kJ/kg)
$H_s(0)$ = Sensible heat ambient air, (kJ/kg)
$H_s(3)$ = Sensible heat hall air, (kJ/kg)

The total heat balance of the structure (DQ) was calculated as the sum of the room and hall heat balances. The total represented the sensible heat loss (gain) for the farrowing/nursery system, and represented the total additional heat required by the system.

As a final check of the iterative process the temperature variation of the hall over the period was calculated. The object was to ensure that a representative hall temperature was used for the iteration. A new hall temperature was calculated based on the average quality of the hall air with the included ventilation for the three hour period. If the difference in temperature was less than ±1 °C, convergence was assumed, signalling the end of the iteration. If the difference was greater than ±1 °C, the program returned to the initial subroutine to recalculate the hall and attic temperatures. The entire process was repeated until the convergence test had been passed.

3.2.6 Output Routines

One of the objects of the programs was the creation of a database of simulated results which could be later used to calculate the economic viability of the soil-air tempering system. The final subroutines were included for control of the program output. The results were stored as random ASCII text files, which could be recovered by a number of software packages. The 7000 subroutine created the heading lines of the output files. This subroutine was called before the actual iterative process of calculations. The 8000 subroutine created the actual output of data for each iteration. This subroutine was called after the successful completion of each iteration.

3.3 Summer Simulation

3.3.1 Main Program

The summer simulation followed essentially the same logic as the winter simulations. The major difference between the summer and winter models was the inclusion of solar radiation effects upon the structure. This was omitted in the winter program, as it was viewed as a benefit, though an unreliable one. In the case of the summer model, solar effects could not be neglected as they contributed to the undesired effect of additional heat load.

The summer simulation made use of four programs, which have been listed in Table 3.4. SUMMOD was written to model any of the three pipe configurations with user specified pipe lengths and number of pipe laterals. SUMVENT modelled the ventilation requirements with no air tempering. The other two programs, SOL-AIR and MEANS, processed the raw temperature data, and calculated the sol-air and mean daily temperatures for use in the summer simulation models.

A further difference between the winter and summer models was that the number of pipes necessary for cooling was not calculated. Instead the optimum number of pipes for winter operation was used to calculate the potential benefit. This introduced the problem of variable inside temperatures within the structure. With limited total airflow into the

Table 3.4 Summer Simulation Programs

Program Name	Model	Output Data
SUMMOD	Tempered	Room Temperature, Relative Humidity, and Ventilation Rates
SUMVENT	Ambient	Room Temperature, Relative Humidity, and Ventilation Rates
SOL-AIR		Sol-Air, Ambient and Outlet Temperatures
MEANS		Hourly and Daily mean temperature values

building, temperature control was not always possible.

The program was operated as an unsteady state process taking into account the periodic effects of temperature and solar radiation, and the effects of the structural materials on the flow of heat. It follows the basic steps presented in the winter program for the ultimate conclusions, in this case the temperature within the structure, and cooling supplied. The effects of no tempering on the structure were also calculated.

In terms of the program, data were held within arrays for the necessary number of iterations. The number of iterations was defined as a function of the decrement factor and the fundamental time lag, as defined in Section 2.4.2. It was also necessary to include a method for calculating the properties of the mixed airflow, as a large amount of ambient air was necessary for ventilation purposes.

3.3.2 Sol-Air Temperature

The first step of the calculations was to define the effects of solar radiation and the sol-air temperature of the structure. This was accomplished by a BASIC routine (SOL-AIR), which can be found in Appendix A. The program calculated the sol-air temperatures, for each temperature record, based on Equation 2.4 (Mackey and Wright, 1944). Calculation of the incident solar radiation (I) involved a lengthy process, which is described in the following paragraphs. The program saved its output in a random text file for later retrieval.

Incident solar radiation (I) was calculated for the roof slopes and the vertical walls, based on the methods presented by Muir (1982) and Esmay and Dixon (1986). The north wall was considered shaded with no incident solar radiation. The incident solar radiation was calculated as a function of recorded climatic data, and the incidence angle for that particular surface. The equation given by Muir (1982) was:

$$I = \frac{H*K}{Z} \qquad (3.13)$$

where; I = Incident solar radiation, W/m^2
H = Solar radiation on a horizontal surface, W/m^2
K = Cosine of the incidence angle
Z = Cosine of the zenith angle

Mean hourly values for the solar radiation on a horizonal surface in Winnipeg on a monthly basis were obtained from the Canadian Climate Normals (1980). The values were given for the 21st day of each month. A linear interpolation was performed to account for the actual date. The program did not take into account the cloudiness index, and assumed that the sky was always clear. The values of mean hourly global solar radiation on a horizontal surface have been shown in Table 3.5.

Table 3.5 Mean hourly global solar radiation on a horizontal surface (MJ/m^2). Canadian Climate Normals, 1980.

LST	May	June	July	August	September
1	0	0	0	0	0
4	0	0.01	0	0	0
7	0.74	0.88	0.85	0.59	0.23
10	2.01	2.13	2.19	1.87	1.44
13	2.35	2.47	2.55	2.25	1.79
16	1.57	1.68	1.71	1.45	0.97
19	0.30	0.43	0.39	0.17	0.02
22	0	0	0	0	0

Note: LST refers to Local Standard Time

Calculation of the cosines of the incidence angle (K) and the zenith angle (Z) were straight forward, as they were functions of the structure's latitude, longitude, orientation, angle with the horizontal, time of day, and the time of year. All of these properties were readily available from the data or they were constants. The equations for these values were obtained from Muir (1982) and were as follows:

$$\begin{aligned} K = {} & \sin \delta \sin \phi \cos s - \sin \delta \cos \phi \sin s \cos g \\ & + \cos \delta \cos \phi \cos s \cos w \\ & + \cos \delta \sin \phi \sin s \cos w \cos g \\ & + \cos \delta \sin s \sin w \sin g \end{aligned} \tag{3.14}$$

$$Z = \sin \delta \sin \phi + \cos \delta \cos \phi \cos w \qquad (3.15)$$

where; δ = Solar declination
ϕ = latitude, (49.9°)
s = surface slope, (facing south positive)
g = surface azimuth angle
w = solar hour angle

The surface slopes (s) for the structure were taken as; 18.4° for the roof, and 90° for the walls.

The surface azimuth angles (g) (Muir, 1982) were given the following values for each exposure; roof, 0°; east, 90°; south, 0°; and west, -90°. A value for the north exposure was not required.

The Solar Declination angle (δ) was defined as:

$$\delta = 23.45° * \sin(0.9863 * (284 + n)) \qquad (3.16)$$

where; n = the numeric day of the year

The Solar Hour Angle (w) was defined as:

$$w = 15\ (°/h) * (12 - \text{Solar Time}) \qquad (3.17)$$

The solar time was calculated as:

$$\text{Solar Time} = \text{Std Time} + E + (4\ \text{min/deg})*(L_{st} - L_{loc}) \qquad (3.18)$$

where; E = Correction factor for variations in Earth's orbit, (min)
L_{st} = Standard meridian for local time zone, (90°)
L_{loc} = Longitude of location, (97°14')

Values of the correction factor (E) were obtained from Muir (1982) and were as follows; May, 4 min; June, 1 min; July, -5 min; August, -5 min; and September, 4 min.

Once the incidence and zenith angles had been determined the incidence on any of the surfaces was defined and thus the incident solar radiation. This permitted the calculation of the effective or sol-air temperature for each surface. In the case of the north wall the sol-air temperature was assumed equivalent to the current ambient temperature.

The simulation required the daily mean ambient and daily mean sol-air temperatures for its calculation of inside temperatures. Output of the SOL-AIR program was further processed by a second program (MEANS) which calculated the daily means of each temperature record. The program read temperature records and averaged eight consecutive, three hour periods to form the daily means. Mean values were based on the current temperature record and the previous seven records. Output was stored in a random file to be retrieved by the summer simulation program.

3.3.3 Program Assumptions

Once the sol-air and mean temperatures were defined, the heat flow rates and the temperatures within the structure could then be determined using the summer model. The summer simulation was similar to the winter model and was based on the following assumptions.

1. The initial room temperatures were the optimum temperature for each room, which was assumed to be 21 °C for the farrowing rooms, and 27 °C for the nursery rooms.

2. The number of pipes used for the summer model would be the same as determined for optimum winter heating.

3. The airflow rates through the pipes remained constant. The airflow rates were held constant so as not to invalidate the temperature data. Any extra airflow requirements would be made up with ambient air.

4. The latent and sensible heat components due to the animals would remain the same as the winter values.

5. The solar adsorptivities (α) of the roof and the walls were assumed to be 0.7 for the roof, and 0.4 for the walls.

6. The surface film coefficients of heat transfer for the inside (h_i) and outside (h_o) surfaces were assumed to be;
   h_i = 9.3 W/m^{2}K (1.65 Btu/hrft^{2}°F),
   h_o = 23.0 W/m^{2}K (4.0 Btu/hrft^{2}°F).

7. It was assumed that there was only a transfer of sensible heat to the airflow while in the pipe. This approximation would have a negative effect on the cooling process as it did not take into account the lost heat of vapourization due to condensation. Overall, it would tend to make the calculations more conservative.

8. All solar radiation values were calculated for clear days to simplify the calculation process. This resulted in an overall negative effect upon the cooling process, again producing conservative results.

9. On the basis of the time lag calculations, it was assumed that all insulated/exterior surfaces would have a time lag of 3.0 hours. All interior surfaces, with the exception of the ceiling, would have no time lag. This considerably eased the burden of calculations while having little effect on the overall results. This was especially true in the case of the interior walls where the time lag was very much less than one hour.

3.3.4 Equivalent Outside Temperatures

The equivalent outside temperatures were calculated for each of the building exposures and construction types. The exposures considered were the roof, east, south, west and north sides of the structure. The building construction types were the roof, exterior walls, floor, and gable ends. The program maintained the previous iteration values for the current calculations. Equation 3.19 was used to calculate the equivalent outside temperatures (Stewart, 1948).

$$t_b = t_m + \frac{h_i * L_e}{U} (t_e' - t_m) \tag{3.19}$$

where;

t_b = Equivalent outside temperature,(°C)
t_m = Daily average Sol-Air temperature, (°C)
h_i = Inside air film coefficient of heat transfer, ($W/m^2 \cdot K$)
L_e = Equivalent decrement factor, dimensionless
t_e'= Sol-Air temperature for the previous period, (°C)

Sol-air and ambient temperature values were calculated for each exposure by the MEANS program (Appendix A). Mean values were calculated as the current temperature record plus the seven previous temperature records. The results were stored in a data file for later retrieval by the summer programs.

3.3.5 Calculation of Hall and Attic Temperatures

Calculation of the hall and attic temperatures was similar to that described for the winter simulation (Section 3.2.2). The principle difference between the two models was the treatment of ambient temperatures. The summer model replaced the ambient temperature with the equivalent outside temperatures for each building exposure. The theory remained the same, but the number of heat flow paths was increased.

3.3.6 Calculation of Static Heat Balance

Calculation of the static heat balances was similar to that described for the winter simulation (Section 3.2.3). The summer simulation calculated all of the heat transfer rates calculated by the winter simulation, however, the addition of the sol-air temperatures made it necessary to calculate heat transfer rates as a function of exposure. This affected the calculation of the gable ends, the exterior walls and the foundation. Calculations of heat transfer through the roof, ceiling, and interior partitions remained essentially the same, with the only difference being the addition of a time lag factor for the ceiling calculations.

3.3.7 Psychrometric Calculations

Calculation of the psychrometric properties of the control volumes was similar to that described in Section 3.2.4 for the winter simulation, except the summer simulation did not assume a constant ambient relative humidity. Mean monthly values of the hourly relative humidities for Winnipeg were obtained from the Canadian Climate Normals (1980) and substituted into the program. The relative humidity values used by the program have been shown in Table 3.6. As a result, relative humidity was allowed to vary throughout the structure. The quality of the air leaving the soil-air tempering system was checked for condensation. If the relative humidity was over 100%, the absolute humidity was recalculated based on the saturation vapour pressure of the pipe outlet air.

Table 3.6 Mean hourly relative humidity at Winnipeg International Airport (%). Canadian Climate Normals, 1980

LST	May	June	July	August	September
1	72	77	81	81	80
4	72	76	80	82	83
7	72	75	79	83	85
10	60	64	67	68	70
13	48	52	54	53	55
16	50	54	57	57	61
19	51	55	59	60	67
22	62	66	70	71	74

3.3.8 Calculation of Ventilation Requirements

The ventilation routine for the modeled structure differed greatly from the winter simulation, because the summer model was based on temperature control as opposed to moisture control. It was assumed that the ventilation rates required for temperature control would be more than adequate for moisture control. Another key difference between the simulations was that the summer simulation had a fixed number of laterals based on optimum winter conditions. This implied that temperature control would not be

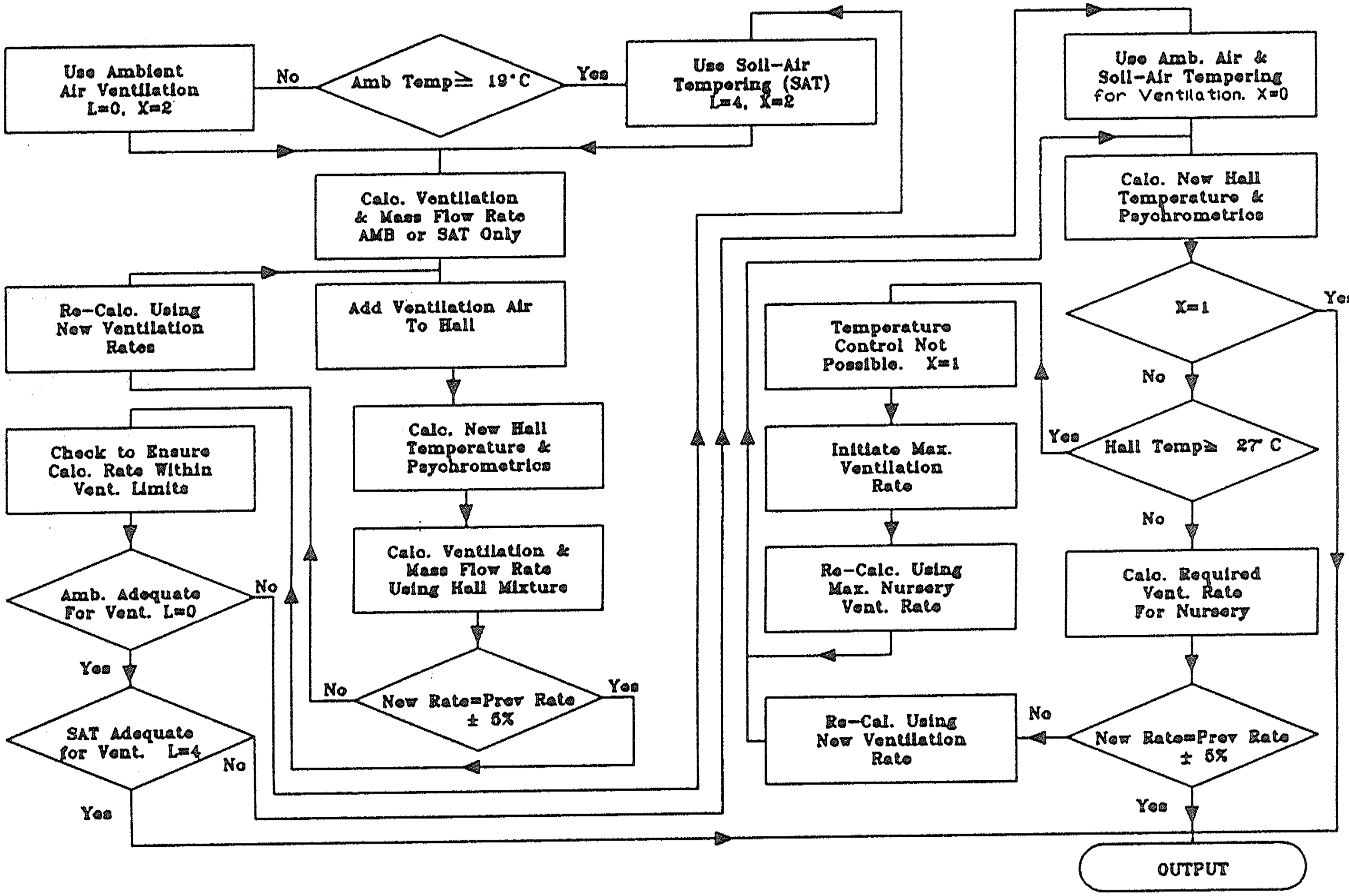


Figure 3.7 – Flowchart of the ventilation logic for the summer simulation programs.

possible at all times, and large amounts of ambient air would be necessary to make up the deficiencies. The following paragraphs described the logic and calculations of determining the ventilation requirements. The logic of the process is shown in Figure 3.7.

It was assumed that the soil-air tempering system would be used primarily for winter and summer conditions. Operation during the spring and fall seasons was seen to have little benefit, so ambient air would be used as the primary ventilation source during moderate weather. A ventilation set point was determined as the ambient temperature at which temperature control within the farrowing rooms was no longer possible at the recommended airflow rates (110 L/s/litter). The set point was determined to be 19° C. For temperatures greater than 19° C, the program would initiate tempering.

The next step was to calculate an estimate of the required ventilation rates based on the sensible heat balances of the rooms. Ventilation potential was determined as the difference between the rooms and the ambient or tempered air sensible heat contents. Initial estimates of the mass and airflow rates were then calculated.

The initial ventilation estimates enabled calculation of the quality and condition of the hall air after the addition of the ventilation air. An iterative procedure was used to test successive calculations of the ventilation rate until convergence of ±5% between successive calculations was achieved. The equation was as follows:

$$V(n) = \frac{Q6(n)}{(H_s(n) - H_s(3)) * 1000J/kJ * 1000L/m^3 * v(3)} \tag{3.20}$$

where; $V(n)$ = Required airflow for each room, (L/s)
$Q6(n)$ = Static heat balance for each room, (W)
$H_s(n)$ = Sensible heat content of each room, (kJ/kg)
$H_s(3)$ = Sensible heat content of hall air, (kJ/kg)
$v(3)$ = Specific volume of hall air, (m^3/kg)

Based upon the ambient temperature and the volumes of ventilation air required by each of the rooms, the program began a decision process to determine the composition of the ventilation air.

The program first tested to see if ambient air would be sufficient for temperature control in each of the rooms. This assumed that the ambient temperature was below the ventilation set point for air tempering (19°C). If both conditions are met, the program continued with ambient air as the sole source of ventilation air. If the maximum ventilation rate for either of the rooms was exceeded even though the ambient temperature was below the set point, air tempering was initiated and the program recalculated the flow rates based on tempered air.

The next test was to determine if there was sufficient tempered air to meet the requirements. This assumed that the ambient temperature was above the set point or that air tempering had been initiated. If the air tempering system could meet the demand, the program continued with tempered air as the sole source of ventilation air. If there was insufficient tempered air to meet the current demand, the program began calculations for a mixture of the two sources.

Ventilation balancing consisted of two cases; i) when the hall temperature was less than the nursery rooms; and ii) when the hall temperature was greater than the nursery. If the temperature in the hall was greater than or equal to 27°C, it was assumed that temperature control within the nursery was not possible. The program began its calculations by isolating the two areas and deriving an initial estimate based on the farrowing rooms. Once the estimate had been established, the airflow rates and psychrometrics of the system were compared against the nursery requirements and adjusted accordingly. The differing criteria of the two areas required an iterative process to determine the balance.

The programs initial estimate was based on the maximum soil-air tempering available plus enough ambient air to reach the maximum ventilation rate for the farrowing rooms. The tempered air was partitioned between the farrowing (46.3%) and the nursery rooms (53.7%). The mixture was then added to the hall, and the new condition was calculated. At this point it was assumed that the farrowing rooms were at the maximum ventilation, so all calculations depended on the state of the hall and the nursery rooms. The new hall temperature was checked to ensure it had not exceeded the nursery temperature of 27°C. If this temperature was exceeded, the maximum ventilation rate was initiated and the calculations were ended. If the hall temperature was less than 27°C, new nursery ventilation rates were calculated based on the new hall conditions. The total ventilation was then recalculated based on the needs of the nursery, and if there was less than ± 5% difference between successive calculations the iteration was ended.

3.3.9 Output Routines

The output routines of the summer simulation had the same objectives and were similar to the winter simulation (Section 3.2.6.) The output variables of the summer simulation program were as listed below. SUMVENT had similar output with the exception of the tempered results.

- Date,
- Time,
- Farrowing room
    - Temperature,
    - Relative Humidity,
    - Total Ventilation,
    - Ambient Ventilation,
    - Tempered Ventilation,
- Nursery Rooms
    - Temperature,
    - Relative Humidity,
    - Total Ventilation,
    - Ambient Ventilation,
    - Tempered Ventilation,
- Total Ventilation,
- Total Ambient Ventilation,
- Total Tempered Ventilation,
- Ambient Temperature, and
- Outlet Temperature

3.4 Cost Analysis

3.4.1 NPV and EAW Calculations

Feasibility of any system can not be judged solely on the basis of technical merit, the economics of the system has a great influence on the ultimate decision to proceed. To provide an unbiased approach, it was decided to perform any financial analyses against a typical conventional system using current market prices. Net present value (NPV) and equivalent annual worth (EAW) calculations were preformed on each of the modeled systems. Sensitivity analyses were performed on the NPV calculations to determine if there were any underlying trends.

In total, there were six soil-air tempering systems and one conventional system studied. The six tempering systems were derived from the 20 m and 29 m lateral length results for each of the 3 pipes. The cost estimates for each of the systems are presented in Section 3.4.2.

Net present value calculations were performed to illustrate the benefits and costs over the life of each system. The NPV's were calculated as a series of cashflows taken over the life of the system. It was assumed that the entire capital cost of the system was paid at year zero. The real rate of interest rate was assumed to be 5%, and the system life was assumed to be 20 years. The real rate of interest represents the lending rate less the annual rate of inflation. The cost of electricity was assumed to be \$0.03/kW·h. The calculations were performed using a PC based spreadsheet.

EAW calculations were performed to substantiate the NPV results, and to illustrate the costs from an annual perspective. The EAW's were calculated using a capital recovery factor, which was determined by Equation 3.21. The same variables of interest rate, rate of inflation, and system life were used.

$$\text{Annuity} = \text{Principal} * \frac{i\cdot(1+i)^n}{(1+i)^n - 1} \tag{3.21}$$

Where; i = Real rate of interest, (%)
n = Number of periods.

Sensitivity analyses were done for each of the financial variables to determine their effect if any on the system NPV. The calculations were performed by varying a single variable while holding the others constant. All calculations were made and tabulated within the spreadsheet.

3.4.2 Cost Estimates for Soil-Air Tempering Systems

Cost estimates were prepared for each of the soil-air tempering scenarios, and a conventional ventilation system. These cost estimates were the basis for the economic evaluation of the systems. The estimates were prepared with the intent that they represented a true financial picture of each system. Costs were based on standard construction values, and personal experience with PVC pipe. Tables were prepared for each of the systems, which itemized the principal cost elements of each system. The bottom line figure represented the total cost of the system including a 10% contingency. The costs did not include either federal or provincial sales taxes.

It was assumed that the PVC pipe was Series 100, bell and spigot, pressure pipe. This pipe has a 685 kPa (100 psi) pressure rating and is capable of withstanding the bearing load of the 3.0 m soil column. The basis of the prices shown in the tables was a cost of $1.43/kg for PVC. This price was obtained from Canron West in July, 1991. Unit weights of the various pipe diameters were representative of industry standards. The quantities of PVC pipe were determined by the number of laterals for each system. It was assumed that two pipes were laid in each trench, as a result there was always an even number of pipes.

Fabricated PVC fittings were used to ensure close, water tight connections. This prevents the possible flooding of pipes and the subsequent loss of capacity. Some of the bends can be fabricated on site for a slight cost savings. The PVC fittings were also quoted by Canron West though personal correspondence of July, 1991.

The cost of the corrugated steel pipe (CSP) was obtained from the ARMTEC Price Book for the Regina, Saskatchewan sales office. These values represented typical market prices for galvanized corrugated lock seam pipe and were given on a linear metre basis. It was assumed that the 800 mm and the 1200 mm diameter CSP had a 2.8 mm thickness.

Metal fabrication consisted of welding and cutting of the CSP pipes to allow connections to the PVC header and a centrifugal blower within the animal structure. The price was quoted in a personal conversation with LB Welding of Outlook, Saskatchewan in July, 1991.

Table 3.7 Soil-air tempering system capital cost estimate for Pipe No. 1 simulation using 29 m lateral lengths. 26 laterals within the system.

Item No.	Description	Unit	Unit Price	Qty	Extension
1.0	Series 100 PVC Pipe				
1.1	250 mm diameter pipe	m	12.42	952	$11,823.84
1.2	350 mm diameter pipe	m	21.19	14	296.66
1.3	500 mm diameter pipe	m	44.09	14	617.26
1.4	600 mm diameter pipe	m	63.45	56	3,553.20
2.0	PVC Fittings				
2.1	250 mm diameter elbow	m	251.18	104	26,122.72
2.2	350 mm diameter elbow	m	494.07	2	808.14
2.3	350 X 250 reducers	each	310.39	26	8,070.14
2.4	500 X 350 reducers	each	594.11	2	1,188.22
2.5	600 X 500 reducers	each	688.17	2	1,376.34
2.6	350 X 350 X 350 tees	each	507.90	13	6,602.70
2.7	500 X 500 X 350 tees	each	823.76	2	1,647.52
2.8	600 X 600 X 350 tees	each	1,049.52	8	8,396.16
3.0	CSP Culvert				
3.1	1200 mm diameter pipe	m	170.46	5	852.30
3.2	800 mm diameter pipe	m	119.59	6	717.54
4.0	Installation				
4.1	PVC Pipe	m	26.00	1,036	26,936.00
4.2	CSP Culvert	L.S.	2,500.00	1	2,500.00
5.0	Metal Fabrication	L.S.	1,000.00	1	1,000.00
6.0	Blower c/w Motor and Installation	L.S.	1,460.00	1	1,460.00
7.0	Insulated enclosure	L.S.	2,000.00	1	2,000.00
8.0	Electric Heaters	kW	100.00	30	3,000.00
	Contingency (10%)				10,896.87
	Total				$119,865.91

Table 3.8 Soil-air tempering system capital cost estimate for Pipe No. 1 simulation using 20 m lateral lengths. 24 laterals within the system.

Item No.	Description	Unit	Unit Price	Qty	Extension
1.0	Series 100 PVC Pipe				
1.1	250 mm diameter pipe	m	12.42	586	$7,278.12
1.2	350 mm diameter pipe	m	21.19	14	296.66
1.3	500 mm diameter pipe	m	44.09	14	617.26
1.4	600 mm diameter pipe	m	63.45	49	3,109.05
2.0	PVC Fittings				
2.1	250 mm diameter elbow	m	251.18	96	24,113.28
2.2	350 mm diameter elbow	m	494.07	2	808.14
2.3	350 X 250 reducers	each	310.39	24	7,449.36
2.4	500 X 350 reducers	each	594.11	2	1,188.22
2.5	600 X 500 reducers	each	688.17	2	1,376.34
2.6	350 X 350 X 350 tees	each	507.90	12	6,094.80
2.7	500 X 500 X 350 tees	each	823.76	2	1,647.52
2.8	600 X 600 X 350 tees	each	1,049.52	8	8,396.16
3.0	CSP Culvert				
3.1	1200 mm diameter pipe	m	170.46	5	852.30
3.2	800 mm diameter pipe	m	119.59	6	717.54
4.0	Installation				
4.1	PVC Pipe	m	26.00	663	17,238.00
4.2	CSP Culvert	L.S.	2,500.00	1	2,500.00
5.0	Metal Fabrication	L.S.	1,000.00	1	1,000.00
6.0	Blower c/w Motor and Installation	L.S.	1,460.00	1	1,460.00
7.0	Insulated enclosure	L.S.	2,000.00	1	2,000.00
8.0	Electric Heaters	kW	100.00	35	3,500.00
	Contingency (10%)				9,164.28
	Total				$100,807.03

Table 3.9 Soil-air tempering system capital cost estimate for Pipe No. 3 simulation using 29 m lateral lengths. 12 laterals within the system.

Item No.	Description	Unit	Unit Price	Qty	Extension
1.0	Series 100 PVC Pipe				
1.1	250 mm diameter pipe	m	12.42	440	$5,464.80
1.2	350 mm diameter pipe	m	21.19	14	296.66
1.3	500 mm diameter pipe	m	44.09	21	925.89
2.0	PVC Fittings				
2.1	250 mm diameter elbow	m	251.18	48	12,056.64
2.2	350 mm diameter elbow	m	494.07	2	808.14
2.3	350 X 250 reducers	each	310.39	12	3,724.68
2.4	500 X 350 reducers	each	594.11	2	1,188.22
2.5	350 X 350 X 350 tees	each	507.90	6	3,047.40
2.6	500 X 500 X 350 tees	each	823.76	4	3,295.04
3.0	CSP Culvert				
3.1	1200 mm diameter pipe	m	170.46	5	852.30
3.2	800 mm diameter pipe	m	119.59	6	717.54
4.0	Installation				
4.1	PVC Pipe	m	26.00	475	12,350.00
4.2	CSP Culvert	L.S.	2,500.00	1	2,500.00
5.0	Metal Fabrication	L.S.	1,000.00	1	1,000.00
6.0	Blower c/w Motor and Installation	L.S.	1,460.00	1	1,460.00
7.0	Insulated enclosure	L.S.	2,000.00	1	2,000.00
8.0	Electric Heaters	kW	100.00	40	4,000.00
	Contingency (10%)				5,568.73
	Total				$61,256.04

Table 3.10 Soil-air tempering system capital cost estimate for Pipe No. 3 simulation using 20 m lateral lengths. 12 laterals within the system.

Item No.	Description	Unit	Unit Price	Qty	Extension
1.0	Series 100 PVC Pipe				
1.1	250 mm diameter pipe	m	12.42	293	$3,639.06
1.2	350 mm diameter pipe	m	21.19	14	296.66
1.3	500 mm diameter pipe	m	44.09	21	925.89
2.0	PVC Fittings				
2.1	250 mm diameter elbow	m	251.18	48	12,056.64
2.2	350 mm diameter elbow	m	494.07	2	808.14
2.3	350 X 250 reducers	each	310.39	12	3,724.68
2.4	500 X 350 reducers	each	594.11	2	1,188.22
2.5	350 X 350 X 350 tees	each	507.90	6	3,047.40
2.6	500 X 500 X 350 tees	each	823.76	4	3,295.04
3.0	CSP Culvert				
3.1	1200 mm diameter pipe	m	170.46	5	852.30
3.2	800 mm diameter pipe	m	119.59	6	717.54
4.0	Installation				
4.1	PVC Pipe	m	26.00	328	8,528.00
4.2	CSP Culvert	L.S.	2,500.00	1	2,500.00
5.0	Metal Fabrication	L.S.	1,000.00	1	1,000.00
6.0	Blower c/w Motor and Installation	L.S.	1,460.00	1	1,460.00
7.0	Insulated enclosure	L.S.	2,000.00	1	2,000.00
8.0	Electric Heaters	kW	100.00	45	4,500.00
	Contingency (10%)				5,053.96
	Total				55,593.53

Table 3.11 Soil-air tempering system capital cost estimate for Pipe No. 4 simulation using 29 m lateral lengths. 26 laterals within the system.

Item No.	Description	Unit	Unit Price	Qty	Extension
1.0	Series 100 PVC Pipe				
1.1	150 mm diameter pipe	m	4.73	952	$4,502.96
1.2	200 mm diameter pipe	m	7.97	14	111.58
1.3	300 mm diameter pipe	m	17.53	14	245.42
1.4	400 mm diameter pipe	m	27.91	14	390.74
1.5	500 mm diameter pipe	m	44.09	42	1,851.78
2.0	PVC Fittings				
2.1	150 mm diameter elbow	m	251.18	104	4,324.32
2.2	200 mm diameter elbow	m	494.07	2	160.76
2.3	200 X 150 reducers	each	310.39	26	2,709.98
2.4	300 X 200 reducers	each	594.11	2	485.94
2.5	400 X 300 reducers	each	688.17	2	792.76
2.6	500 X 400 reducers	each	688.17	2	1,139.62
2.7	200 X 200 X 200 tees	each	507.90	13	1,654.12
2.8	300 X 300 X 200 tees	each	823.76	2	796.34
2.9	400 X 400 X 200 tees	each	1,049.52	2	932.04
2.10	500 X 500 X 200 tees	each	740.17	6	4,441.02
3.0	CSP Culvert				
3.1	1200 mm diameter pipe	m	170.46	5	852.30
3.2	800 mm diameter pipe	m	119.59	6	717.54
4.0	Installation				
4.1	PVC Pipe	m	26.00	1,036	26,936.00
4.2	CSP Culvert	L.S.	2,500.00	1	2,500.00
5.0	Metal Fabrication	L.S.	1,000.00	1	1,000.00
6.0	Blower c/w Motor and Installation	L.S.	1,460.00	1	1,460.00
7.0	Insulated enclosure	L.S.	2,000.00	1	2,000.00
8.0	Electric Heaters	kW	100.00	25	2,500.00
	Contingency (10%)				5,764.53
	Total				$63,409.79

Table 3.12 Soil-air tempering system capital cost estimate for Pipe No. 4 simulation using 20 m lateral lengths. 24 laterals within the system.

Item No.	Description	Unit	Unit Price	Qty	Extension
1.0	Series 100 PVC Pipe				
1.1	150 mm diameter pipe	m	4.73	586	$2,771.78
1.2	200 mm diameter pipe	m	7.97	14	111.58
1.3	300 mm diameter pipe	m	17.53	14	245.42
1.4	400 mm diameter pipe	m	27.91	14	390.74
1.5	500 mm diameter pipe	m	44.09	35	1,543.15
2.0	PVC Fittings				
2.1	150 mm diameter elbow	m	251.18	96	3,991.68
2.2	200 mm diameter elbow	m	494.07	2	160.76
2.3	200 X 150 reducers	each	310.39	24	2,501.52
2.4	300 X 200 reducers	each	594.11	2	485.94
2.5	400 X 300 reducers	each	688.17	2	792.76
2.6	500 X 400 reducers	each	688.17	2	1,139.62
2.7	200 X 200 X 200 tees	each	507.90	12	1,526.88
2.8	300 X 300 X 200 tees	each	823.76	2	796.34
2.9	400 X 400 X 200 tees	each	1,049.52	2	932.04
2.10	500 X 500 X 200 tees	each	740.17	6	4,441.02
3.0	CSP Culvert				
3.1	1200 mm diameter pipe	m	170.46	5	852.30
3.2	800 mm diameter pipe	m	119.59	6	717.54
4.0	Installation				
4.1	PVC Pipe	m	26.00	663	17,238.00
4.2	CSP Culvert	L.S.	2,500.00	1	2,500.00
5.0	Metal Fabrication	L.S.	1,000.00	1	1,000.00
6.0	Blower c/w Motor and Installation	L.S.	1,460.00	1	1,460.00
7.0	Insulated enclosure	L.S.	2,000.00	1	2,000.00
8.0	Electric Heaters	kW	100.00	30	3,000.00
	Contingency (10%)				4,747.03
	Total				$52,217.32

The centrifugal blower was sized to accommodate a system requirement of 1,300 L/s (2200 CFM). The price was obtained from the Hanscomb's Yardsticks for Costing (1991). The price included the fan, airfoil, motor and drive, and installation. The indicated price for Winnipeg was $1,460.

The cost of the electric heaters was based on using wall mounted, forced air units. A price of $100 per installed kW was suggested by Dennis Hodgkinson during a conversation in 1988. This price was further supported by Hanscomb's (1991). The heater requirements for each system were determined as the required heat at the winter design temperature of -34°C.

The conventional system required no special costing beyond that for electric heaters. Based on a system requirement of 75 kW and a 10% contingency, the cost of heaters was $8,250 for the conventional system. All other equipment, such as exhaust fans, was common between the systems and was not considered in the cost estimates.

4.0 DISCUSSION AND RESULTS

4.1 Model Assumptions

The calculation procedure for the simulations was based on the Equivalent Temperature Differential Method recommended by ASHRAE (1985), a finite difference model. The procedure was well suited to this type of approach as the results were based on current and previously defined quantities. Implicit relationships between the various control volumes within the model made it necessary to make use of an iterative procedure.

The winter simulations assumed a simple one dimensional heat transfer through the structure. For the purposes of energy estimations this was an acceptable means, as outlined by ASHRAE (1985) and Esmay and Dixon (1986).

The effects of convective heat transfer were simply handled within the overall coefficient of heat transfer (U). The air film heat transfer coefficients were obtained from Mackey and Wright (1944) and ASHRAE (1985). The outside coefficient was assumed to be 22.7 W/m^2K, and the inside coefficient was 9.3 W/m^2K.

The effects of solar radiation were not included in the winter simulation. The net effect of solar radiation would have been positive, however it was felt that the design should reflect the worst condition which excludes solar gain. This assumption had a conservative effect on the results.

The effects of solar radiation were included in the summer model. The sol-air temperature as outlined by Mackey and Wright (1944) and ASHRAE (1985) effectively incorporated the solar radiation component into an equivalent dry bulb temperature. Sol-air temperatures were calculated for each exposure of the building based on the time of day and year. It was assumed that the cloudiness index was unity, that is a clear day. This assumption overestimated the solar intensity, which added a safety factor to the summer simulation.

The purpose of the simulations was to determine if the soil-air tempering systems were capable of providing ventilation requirements in a harsh environment. The simplifying assumptions of the simulations tended to be conservative, but they provided the worst case scenarios for both winter and summer.

The winter simulation assumed that moisture control was possible at all times therefore the relative humidity within each rooms was constant. It was also assumed that the absolute humidity of the soil-air tempering system was equal to the absolute humidity of the ambient air, which was assumed to have a relative humidity of 95%. An assumption was made that the system laterals were dry and that there was no moisture picked up by the airflow. These were not unreasonable assumptions, and the resulting airflow rates are within the recommended ranges of Agriculture Canada (1981, as cited by VIDO, 1986) and Esmay and Dixon (1986).

The relative humidity of the ambient air was allowed to vary in the summer simulation, because temperature control was the governing criterion. It was assumed that the absolute humidity of the outlet air was equivalent to the ambient air, unless the dew point was encountered at which time the absolute humidity of the outlet was recalculated.

4.2 Model Results

The temperature database for ambient temperatures and outlet temperatures for each of the 29 m long pipes are shown in Figures 4.1 to 4.3. The figures show that the soil-air tempering systems provided a substantial heat gain to the ambient air and greatly reduced the amplitude of the daily variation in temperatures. As ambient air began to warm the effects become less pronounced. This heat gain and reduction in daily variation were the factors which prompted further investigations into the feasibility of using a soil-air tempering system.

The outlet temperatures for the 20 m lengths are not shown to simplify the graphs. A comparison of the 29 m outlet temperature and the 20 m outlet temperature for Pipe 1 (250 mm Diameter, 50 L/s airflow) is given in Figure 4.4.Over the winter period the 20 m pipe lengths experienced a wider range of temperature swings than the longer pipes. The average temperature of the two lengths for the period were within 2°C. Murray (1987) found that on average 80% of the temperature tempering occurred within the first

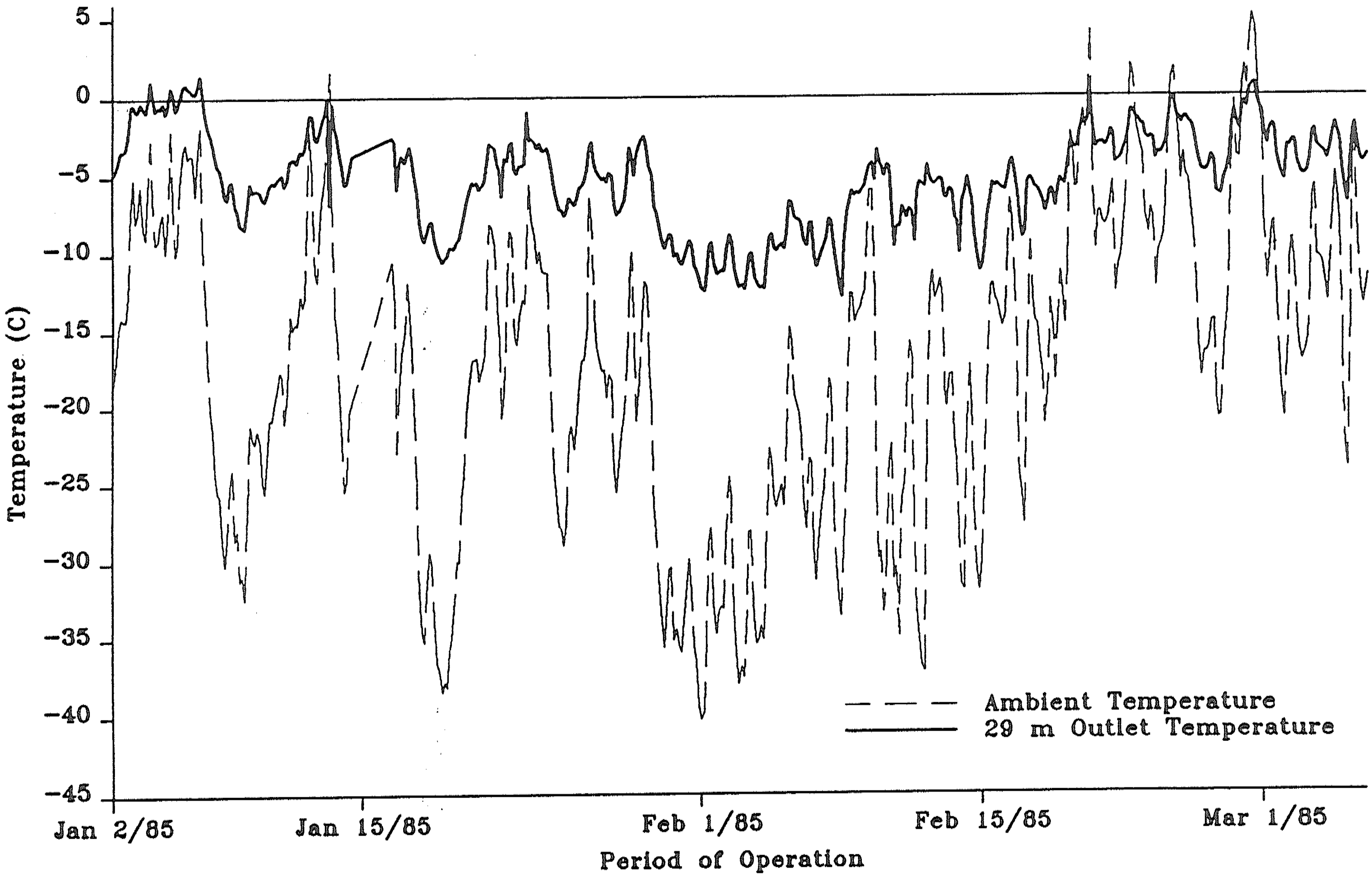


Figure 4.1 Pipe 1 temperature database for winter period. Jan. 2 to Mar. 6, 1985

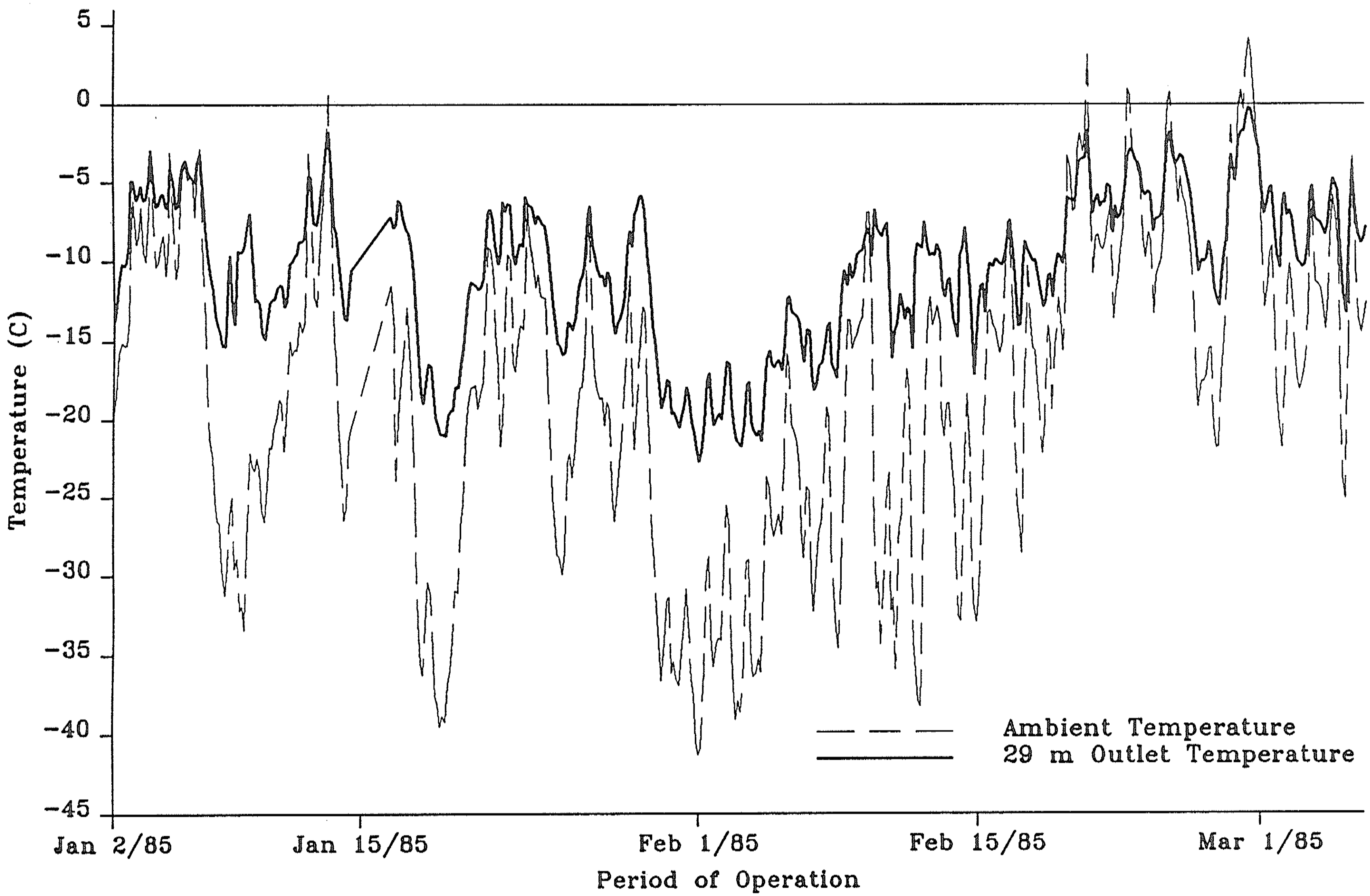


Figure 4.2 Pipe 3 temperature database for winter period. Jan. 2 to Mar. 6, 1985.

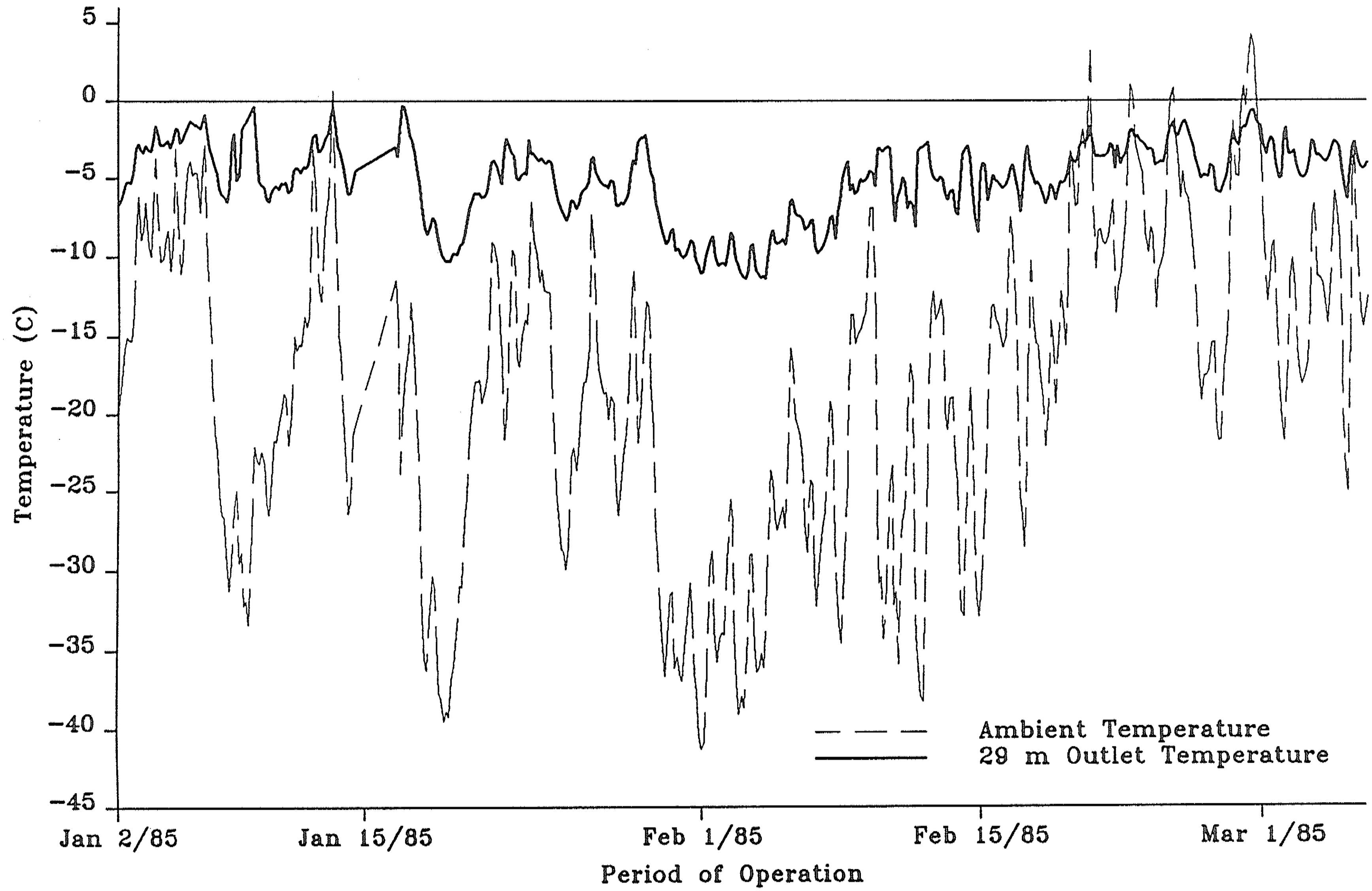


Figure 4.3 Pipe 4 temperature database for winter period. Jan. 2 to Mar. 6, 1985.

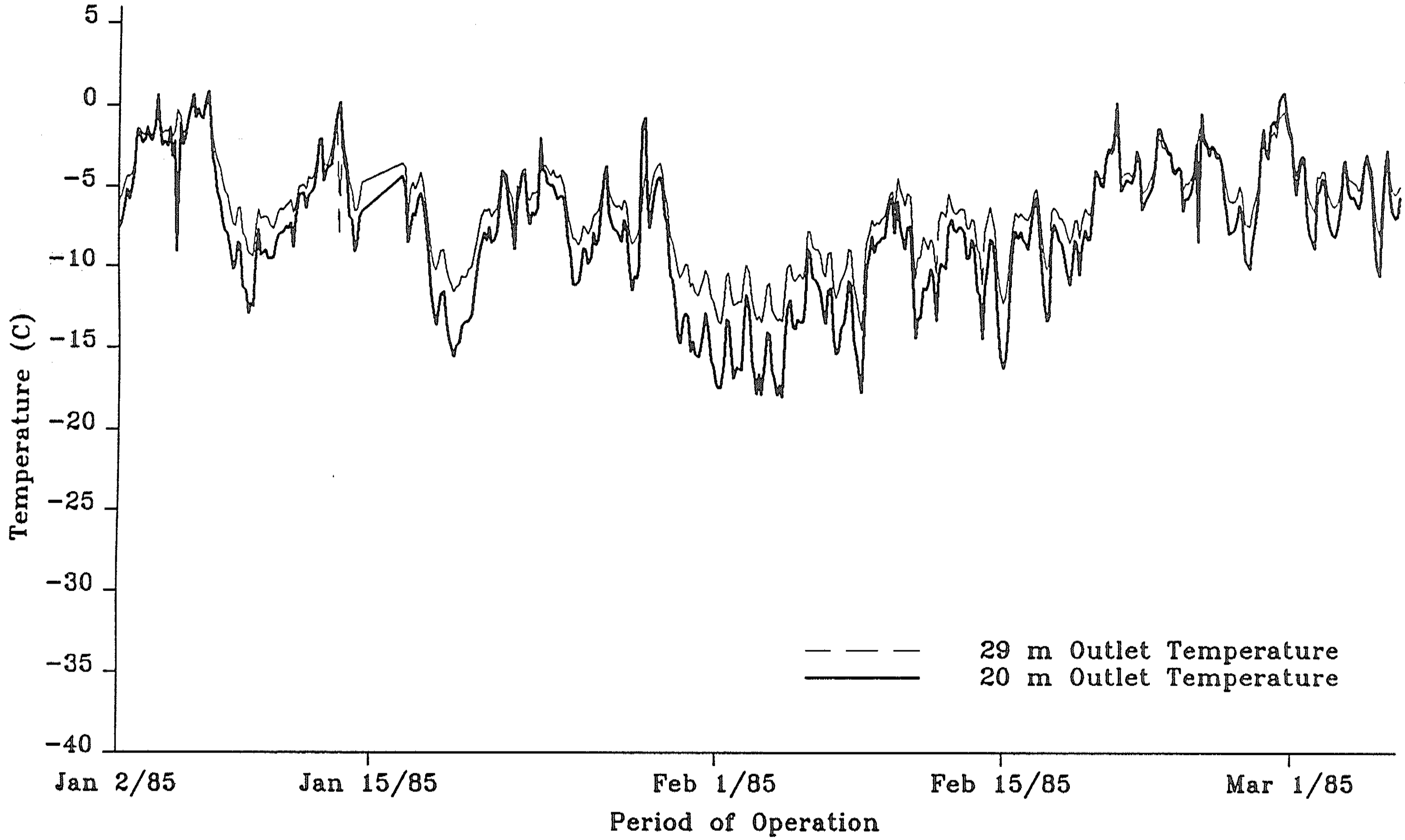


Figure 4.4 Comparison of pipe 1, 20 m and 29 m outlet temperatures for the winter period. Jan. 2 to Mar. 6, 1985.

20 m of the lateral or 67% of the length. The effects of differing the length were to be investigated to determine the cost-benefit effect. The other two pipes performed in a similar manner with comparable results.

Summer databases are shown in Figures 4.5 to 4.7, and as was the case with the winter simulation only the ambient and 29 m length results have been shown. The outlet air of the tempering systems showed significant differences from the ambient air for each of the pipes. A noticeable effect of the soil-air tempering was the reduced amplitudes of the daily temperature variations. It was thought that this effect and the reduced temperatures might provide a benefit for summer cooling.

The 20 m lateral lengths performed with slightly warmer temperatures than the 29 m lengths. The 20 m lengths were not modeled for summer operation, because the preliminary results of the 29 m lengths showed little or no summer cooling benefit. It was felt that if only marginal benefits existed with the 29 m lengths, there was nothing to prove by running the 20 m pipe length model. Benefits were gauged by comparing the resultant temperatures within the structure, while using or not using soil-air tempering.

The design criteria of the physical systems were based on winter operation. Typical systems in Manitoba do not make use of summer cooling beyond the use of increased ventilation rates. To make a valid comparison with conventional systems, the soil-air tempering system was sized for winter moisture control not summer cooling. The size

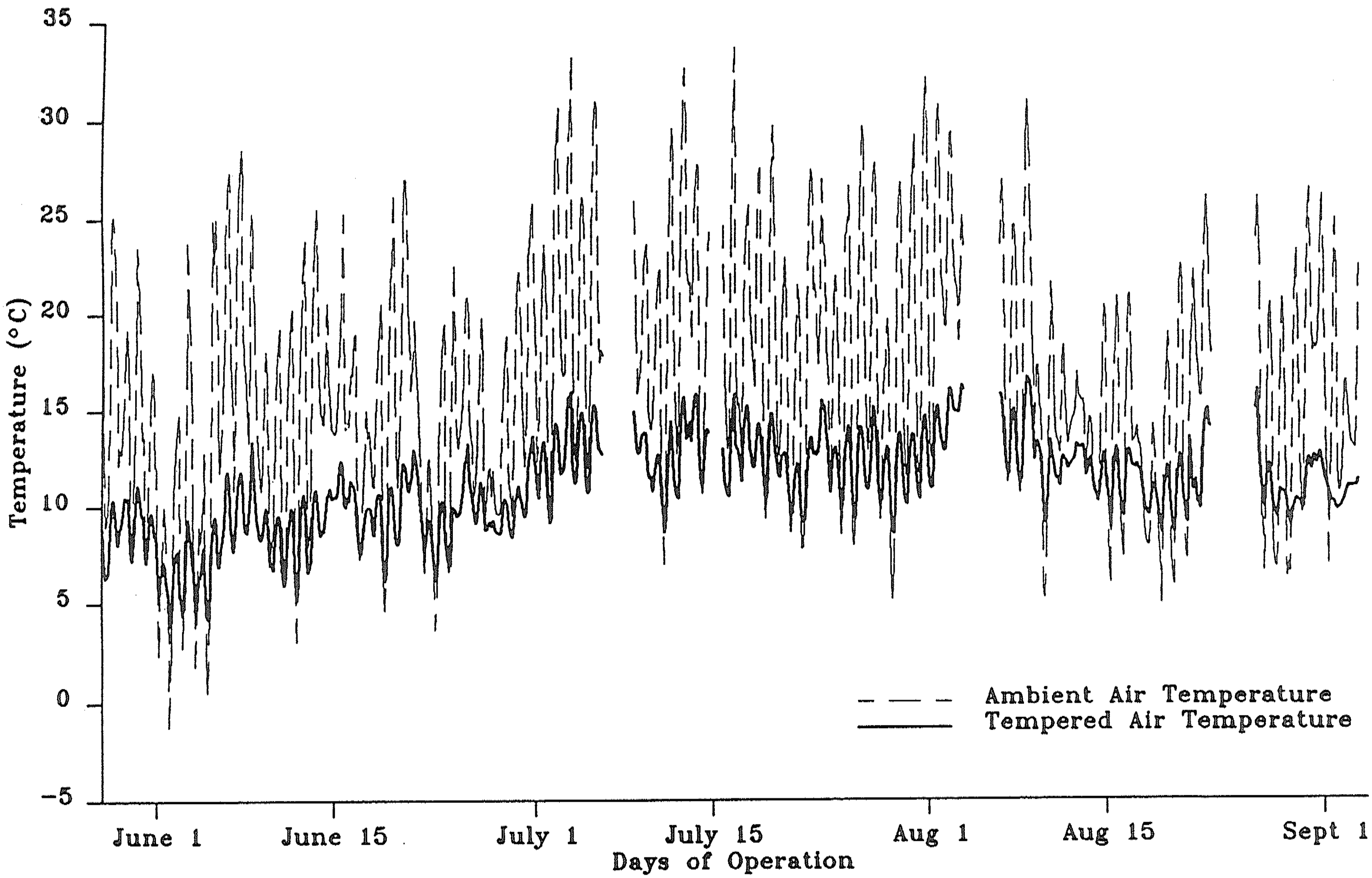


Figure 4.5 Pipe 1, 29 m outlet and ambient temperatures for the summer period. May 27, 1985 to September 3, 1985.

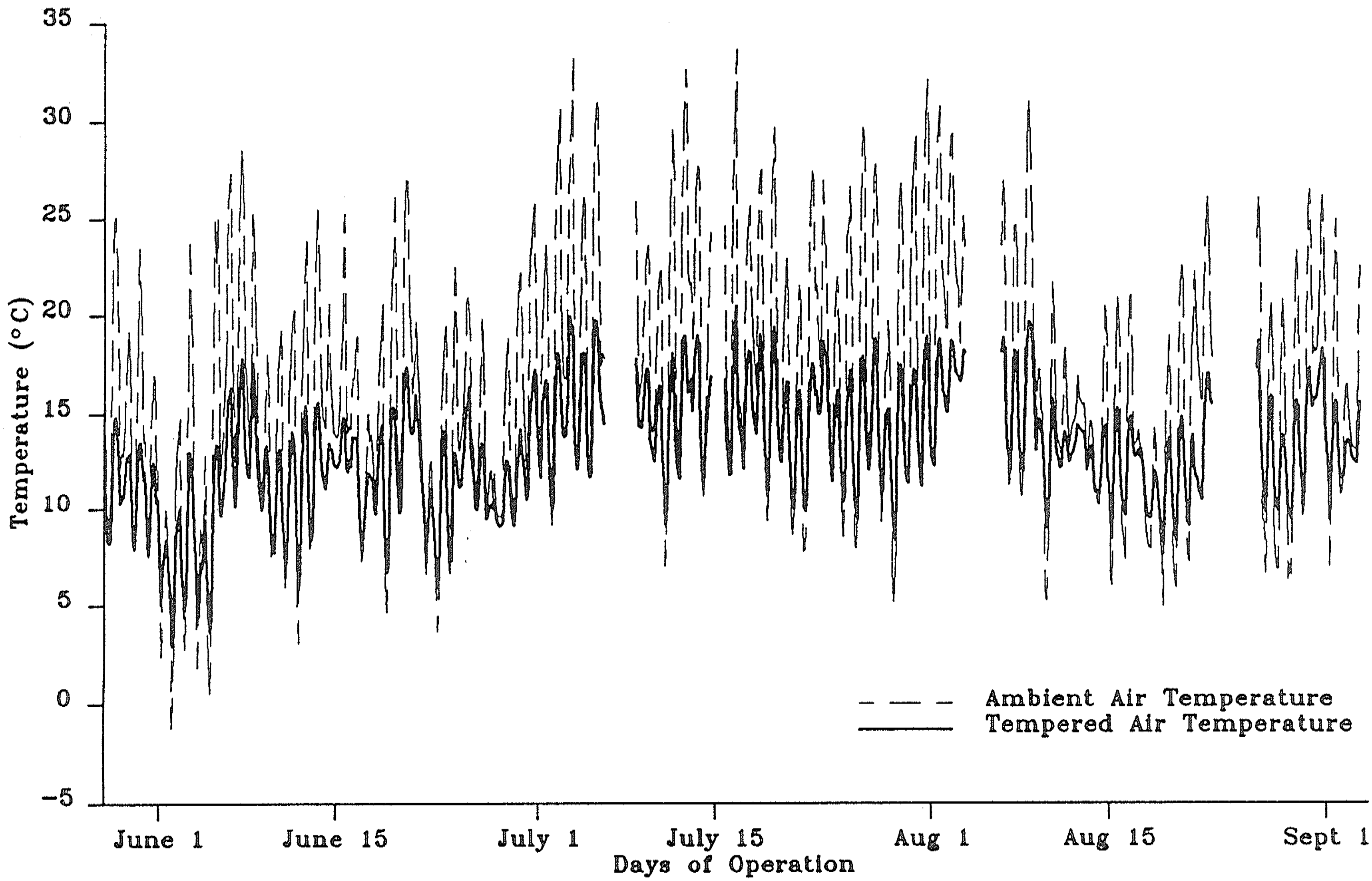


Figure 4.6 Pipe 3, 29 m outlet and ambient temperartures for the summer period. May 27, 1985 to September 3, 1985.

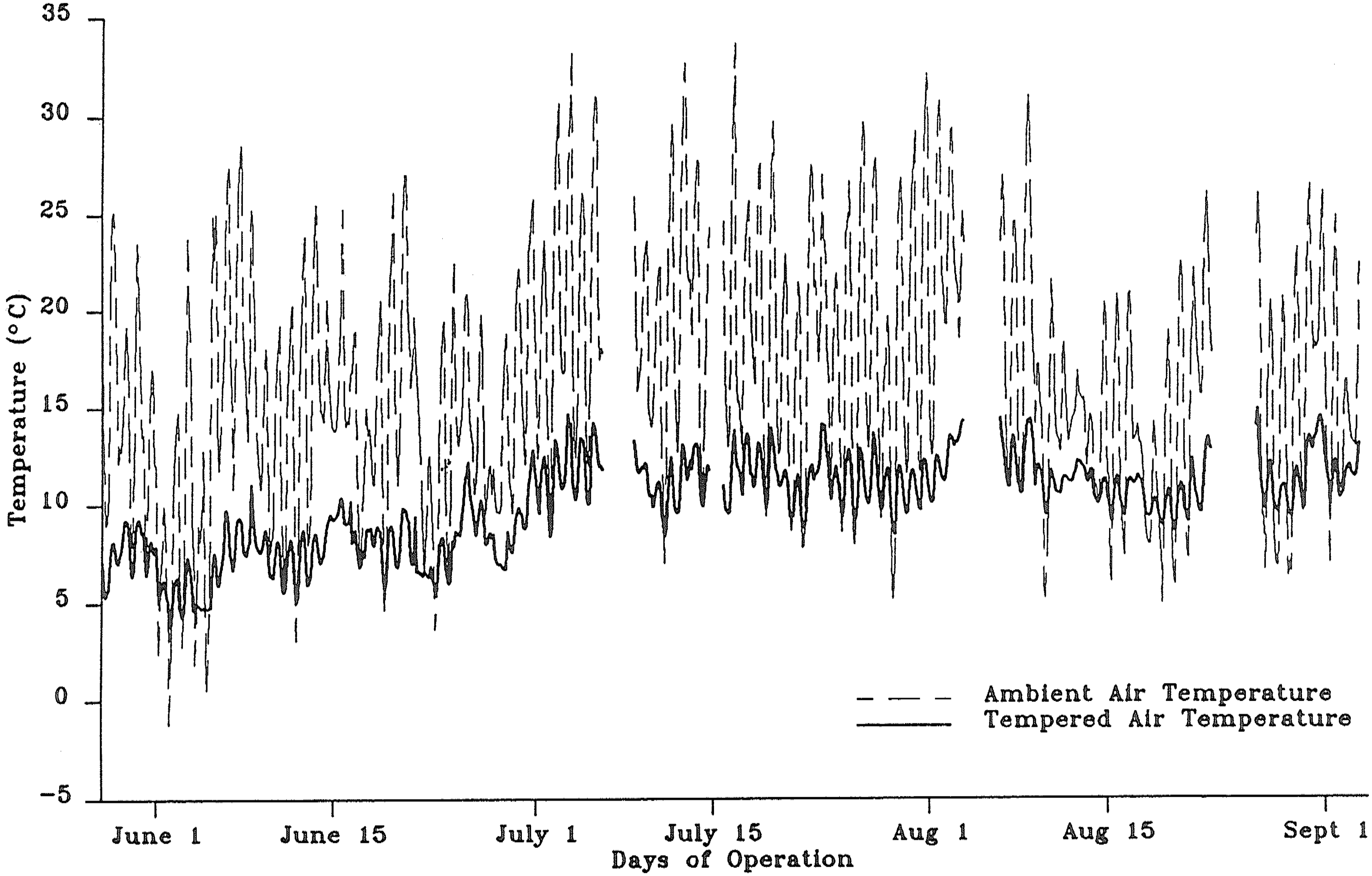


Figure 4.7 Pipe 4, 29 m outlet and ambient temperature for the summer period. May 27, 1985 to September 3, 1985.

or number of laterals within the system were determined as a function of the winter design temperature for Winnipeg and the resulting design heat load. After the systems were selected for winter conditions, they were run under summer conditions to determine if any benefits were derived.

To determine the design heat load, the temperature database was searched for occurrences of the ambient temperature at the winter design temperature of -34° C. A range of ± 2° C from the design temperature was used for the search. There were 34 occurrences of the design temperature within the database. The average temperature of these occurrences was -34° C. The required heat (kW) values of all occurrences were averaged to yield the winter design load for each of the pipe configurations. The design heat load for the conventional system was determined to be 71.8 kW. The design heat loads for the soil-air tempering systems have been shown in Table 4.1.

Table 4.1 Design heat loads and number of pipes for each soil-air tempering system configuration.

System Number No.	Pipe No.	Pipe Dia. (mm)	Airflow Rate (L/s)	Lateral Length (m)	Design Heat Load (kW)	Required of Pipes
1	1	250	50	29	28.1	25
2	1	250	50	20	34.4	24
3	3	250	100	29	39.3	11
4	3	250	100	20	45.1	11
5	4	150	50	29	22.8	26
6	4	150	50	20	31.7	24

The simulation results of each pipe were then searched for all occurrences of the design heat load, for that specific pipe and length. The simulation had calculated the instantaneous heat production of the soil-air tempering systems using a variable number of pipes. A search for a specific range of heat values (± 1 kW) yielded the number of pipes required to supply that quantity of heat. The number of pipes required by each pipe configuration has been given in Table 4.1.

Tables 4.2 to 4.4 on the following pages present the results of the winter simulations for Pipes 1, 3, and 4. The maximum, minimum and average values of temperature, required heat, and ventilation rates are given for ambient conditions and each pipe. The results were separated into the required heat loads for the conventional and tempered systems, and a summary of the ventilation rates for each system. The required heat loads for the period of study are given as a total sum in units of kW·h, and the instantaneous maximums, minimums, and average values (kW). The system gain was the difference between the conventional and the tempered system, and represented the derived energy savings.

The ventilation results were presented to determine the effect of tempering on ventilation rate. The dry air mass flow rates were constant for all systems, since it was assumed that the absolute humidities of the ambient and tempered air were equivalent. The airflow for the conventional system was presented in terms of m^3/h and L/s. Results for the tempering systems were presented for the total ventilation rate, and the components of tempered and ambient air required. All airflows were measured at the inlet of the building.

Table 4.2 Winter simulation results for Pipe 1 (250 mm diameter, 50 L/s airflow), including temperature, system heat, and ventilation rates for ambient, 29 m and 20 m pipe length systems. January 2 to March 6, 1985

	Sum (kW·h)	Minimum	Maximum	Average
System Heat Loads				
Ambient Ventilation				
Ambient Temp (C)		-41.3	4.2	-18.4
Req'd Heat (kW)	-20321	-86.8	-5.5	-41.8
Tempered Airflow - 29 m Pipe Length				
Outlet Temp (C)		-13.9	0.5	-6.2
Req'd Heat (kW)	-9046	-34.7	-7.4	-18.6
System Gain (kW)	11275	-16.8	59.0	23.2
Tempered Airflow - 20 m Pipe Length				
Outlet Temp (C)		-18.0	1.0	-7.8
Req'd Heat (kW)	-10542	-42.3	-7.3	-21.7
System Gain (kW)	9779	-16.1	52.9	20.1
Ventilation Rates				
Ambient Ventilation				
Mass Flow (kg/s)		1.5	2.3	1.6
Airflow (m^3/h)		3868.1	6313.6	4300.0
Airflow (L/s)		1074.5	1753.8	1194.4
29 m Pipe Length				
Total Airflow (L/s)		1073.0	1763.0	1193.4
Tempered Airflow (L/s)		1073.0	1273.0	1171.1
Ambient Airflow (L/s)		0.0	513.3	22.3
20 m Pipe Length				
Total Airflow (L/s)		1056.0	1769.0	1186.4
Tempered Airflow (L/s)		1056.0	1224.0	1150.8
Ambient Airflow (L/s)		0.0	569.9	35.7

Table 4.3 Winter simulation results for Pipe 3 (250 mm diameter, 100 L/s airflow), including temperature, system heat, and ventilation rates for ambient, 29 m and 20 m pipe length systems. January 2 to March 6, 1985.

	Sum (kw·h)	Minimum	Maximum	Average
System Heat Loads				
Ambient Ventilation				
Ambient Temp (C)		-41.3	4.2	-18.4
Req'd Heat (kW)	-20364	-0.1	-0.0	-0.0
Tempered Airflow - 29 m Pipe Length				
Outlet Temp (C)		-22.8	-0.2	-10.5
Req'd Heat (kW)	-13147	-0.1	-0.0	-0.0
System Gain (kW)	7217	-0.0	0.1	0.0
Tempered Airflow - 20 m Pipe Length				
Outlet Temp (C)		-27.4	2.2	-12.1
Req'd Heat (kW)	-14597	-0.1	-0.0	-0.0
System Gain (kW)	5767	-0.0	0.1	0.0
Ventilation Rates				
Ambient Ventilation				
Mass Flow (kg/s)		1.5	2.3	1.6
Airflow (m^3/h)		3868.1	6313.6	4300.0
Airflow (L/s)		1074.5	1753.8	1194.4
29 m Pipe Length				
Total Airflow (L/s)		1034.0	1766.0	1174.2
Tempered Airflow (L/s)		1034.0	1149.0	1097.9
Ambient Airflow (L/s)		0.0	666.6	76.6
20 m Pipe Length				
Total Airflow (L/s)		1015.0	1770.0	1167.5
Tempered Airflow (L/s)		1015.0	1148.0	1093.4
Ambient Airflow (L/s)		0.0	670.1	74.4

Table 4.4 Winter simulation results for Pipe 4 (150 mm diameter, 50 L/s airflow), including temperature, system heat, and ventilation rates for ambient, 29 m and 20 m pipe length systems. January 2 to March 6, 1985.

	Sum (kW·h)	Minimum	Maximum	Average
System Heat Loads				
Ambient Ventilation				
Ambient Temp (C)		-41.3	4.2	-18.4
Req'd Heat (kW)	-20364	-86.8	-5.5	-41.8
Tempered Airflow - 29 m Pipe Length				
Outlet Temp (C)		-11.4	-0.3	-5.1
Req'd Heat (kW)	-7990	-30.4	-6.3	-16.4
System Gain (kW)	12374	-9.6	66.2	25.4
Tempered Airflow - 20 m Pipe Length				
Outlet Temp (C)		-17.3	-1.0	-8.3
Req'd Heat (kW)	-11001	-41.9	-10.8	-22.6
System Gain (kW)	9363	-9.5	61.7	19.2
Ventilation Rates				
Ambient Ventilation				
Mass Flow (kg/s)		1.5	2.3	1.6
Airflow (m^3/h)		3449.1	6437.3	4118.3
Airflow (L/s)		958.1	1788.1	1144.0
29 m Pipe Length				
Total Airflow (L/s)		1083.0	1760.0	1198.4
Tempered Airflow (L/s)		1083.0	1324.0	1184.4
Ambient Airflow (L/s)		0.0	460.6	14.1
20 m Pipe Length				
Total Airflow (L/s)		1057.0	1760.0	1184.3
Tempered Airflow (L/s)		1057.0	1223.0	1150.8
Ambient Airflow (L/s)		0.0	560.9	33.6

The conventional ventilation system made use of ambient air as the sole source of ventilation air. For the period in question temperatures varied from a minimum of -41° C to a maximum of 4.2°C (Refer to Table 4.2). This range encompassed the design value of -33° C for Winnipeg suggested by ASHRAE (1985). The simulation calculated that for the period in question 20,321 kWh of energy would be required to warm the incoming ventilation air and maintain temperature control within the animal rooms. This value was the basis of all further comparisons with the soil-air tempering systems.

Results for Pipe 1 (Table 4.2) are shown for the winter period from January 2 to March 6, 1985. During this period the required heat of the 29 m Laterals was 9,046 kWh, and the 20 m laterals required 10,542 kWh. The required heat was the supplemental energy required by the tempering systems to maintain the optimum room temperatures. In both cases this was a reduction of approximately 50% compared to the conventional system. The 29 m lateral system required 14.2% less heat than the shorter system. Outlet temperatures varied by 14.4° C for the 29 m system, and 19° C for the 20 m system.

Results for Pipe 3 (Table 4.2) are shown for the same winter. During this period the supplemental heat required by the 29 m Laterals was 13,147 kWh, and the 20 m laterals required 14,597 kWh. In the case of Pipe 3, the reduction was between 28.5% and 35.4% of the conventional system. The 29 m lateral system required 9.9% less heat than the shorter system. Outlet temperatures varied by 23.0° C for the 29 m system, and 29.6° C for the 20 m system.

Results for Pipe 4 (Table 4.3) are shown for the same winter period. During this period the supplemental heat required by the 29 m Laterals was 7,990 kWh, and the 20 m laterals required 11,001 kWh. In the case of Pipe 4, the reduction was between 45.8% and 60.8% of the conventional system. The 29 m lateral system required 27.4% less heat than the shorter system. Outlet temperatures varied by 23.0° C for the 29 m system, and 29.6° C for the 20 m system.

The results suggest that system heat gain was inversely proportional to pipe diameter and to airflow rate. Pipe 4, which had the smaller pipe diameter (152 mm) and the lower airflow rate (50 L/s), had the highest heat recovery, followed by Pipe 1 (254 mm diameter, 50 L/s airflow rate). Pipe 3, which had a pipe diameter of 254 mm and an airflow rate of 100 L/s, showed the least amount of difference between pipe lateral length. This would imply that the slower airflow rates permitted greater heat transfer between the pipe and the moving airstream. The smaller diameters would also have a better surface area to volume ratio.

The ventilation rates of the three pipes varied little from that required by the conventional system. This was as expected since the basis of ventilation was moisture movement and absolute humidities were assumed constant. The variation shown is due to the differences in density because of temperature. Since the 29 m systems had the warmest temperatures they also had the higher volumetric airflow rates.

For the most part, there was relatively little difference between the range of ventilation rates necessary for each of the systems. This difference became even more negligible when one considered that these rates were further partitioned for each of the rooms. From the summer calculations it was determined that 46.3% of the ventilation requirement was necessary for the farrowing rooms, the remainder went to the nursery rooms.

From Table 4.5, the model indicated a need for maintaining a ventilation rate slightly higher than the recommended continuous ventilation rate recommended by VIDO (1986), but still in the correct range. This agreement between the values would tend to validate the results obtained by the model.

Table 4.5 Comparison of predicted ventilation rate to recommendations of VIDO (1986).

Ventilation Type	Model	VIDO
Winter Minimum		
Farrowing (L/s/litter)	8.3	7
Nursery (L/s/pig)	1.13	0.7 - 0.9
Winter Maximum		
Farrowing (L/s/litter)	13.5	14
Nursery (L/s/pig)	1.85	1.4-1.9

A comparison of the room temperatures during the summer period is shown in Table 4.6. These results are for the conventional and soil-air tempering systems for the period of May 27 to September 3, 1985. The table presents the maximum, minimum, and average values for the ambient and 29 m outlet temperatures, the farrowing rooms temperature, and the nursery rooms temperature. Pipe 4, which had the smallest diameter and slowest airflow rate, seemed to be the most efficient in terms of cooling, although the difference in cooling results were not as pronounced as those for heating. All of the soil-air tempering systems and the conventional system operated within the same range of temperatures with little perceived differences or benefits.

The results of Table 4.6 are supported by Figures 4.8 and 4.9 which give the temperature differentials between ambient and pipe 4 (150 mm diameter, 50 L/s airflow) for the resultant farrowing and nursery rooms temperatures, respectively. These graphs represent values calculated for the period between July 15 and July 31, 1985. This period was selected because it had relatively high ambient temperatures. In both rooms, the maximum temperature difference was approximately 2°C, between a conventional and the tempering system. These graphs just show the relative difference between the two systems, they do not imply that the optimum temperature of the rooms were maintained.

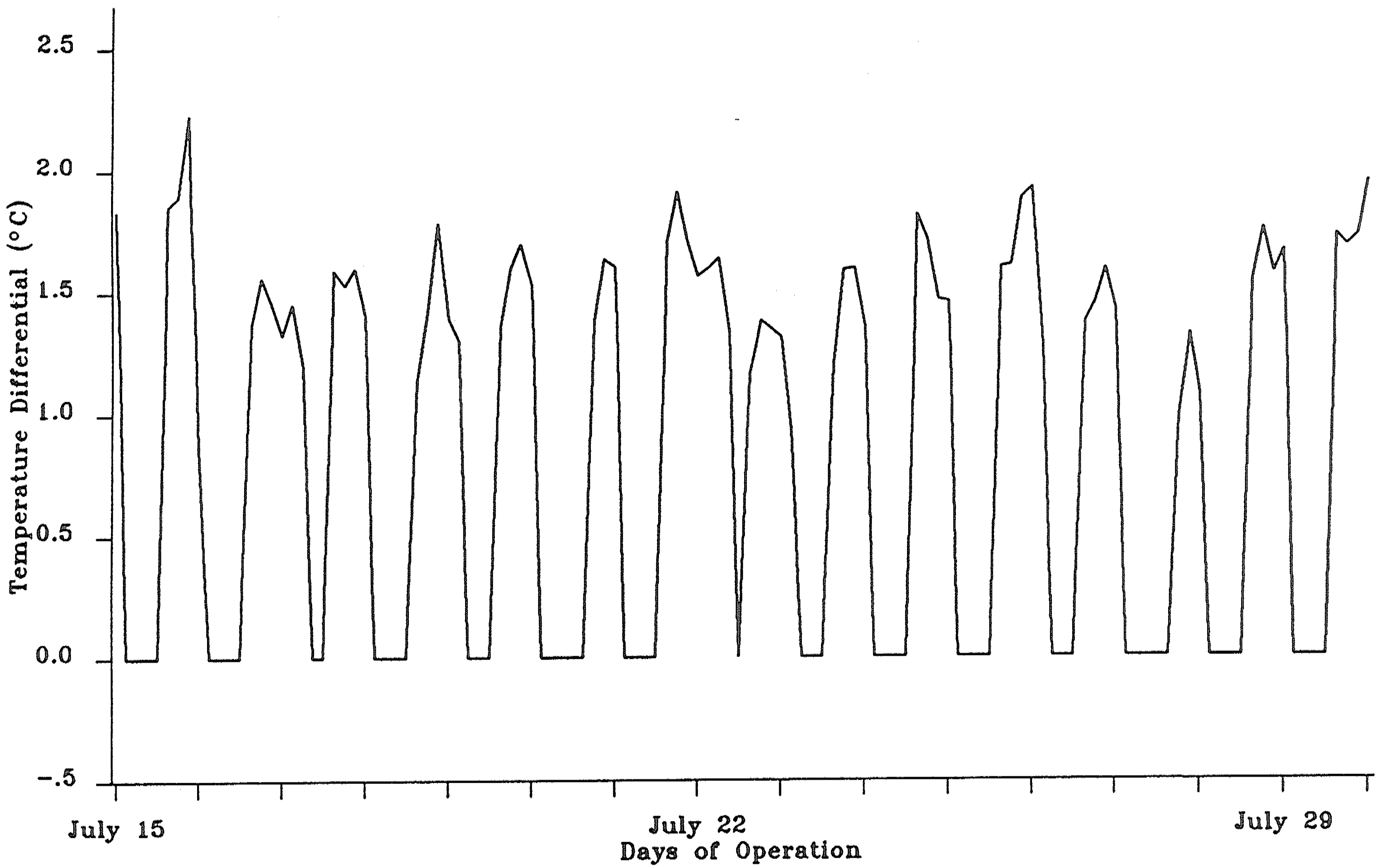


Figure 4.8 Summer simulation of pipe 4. Variation of farrowing room temperature when using ambient air for ventilation as compared to soil-air tempering.

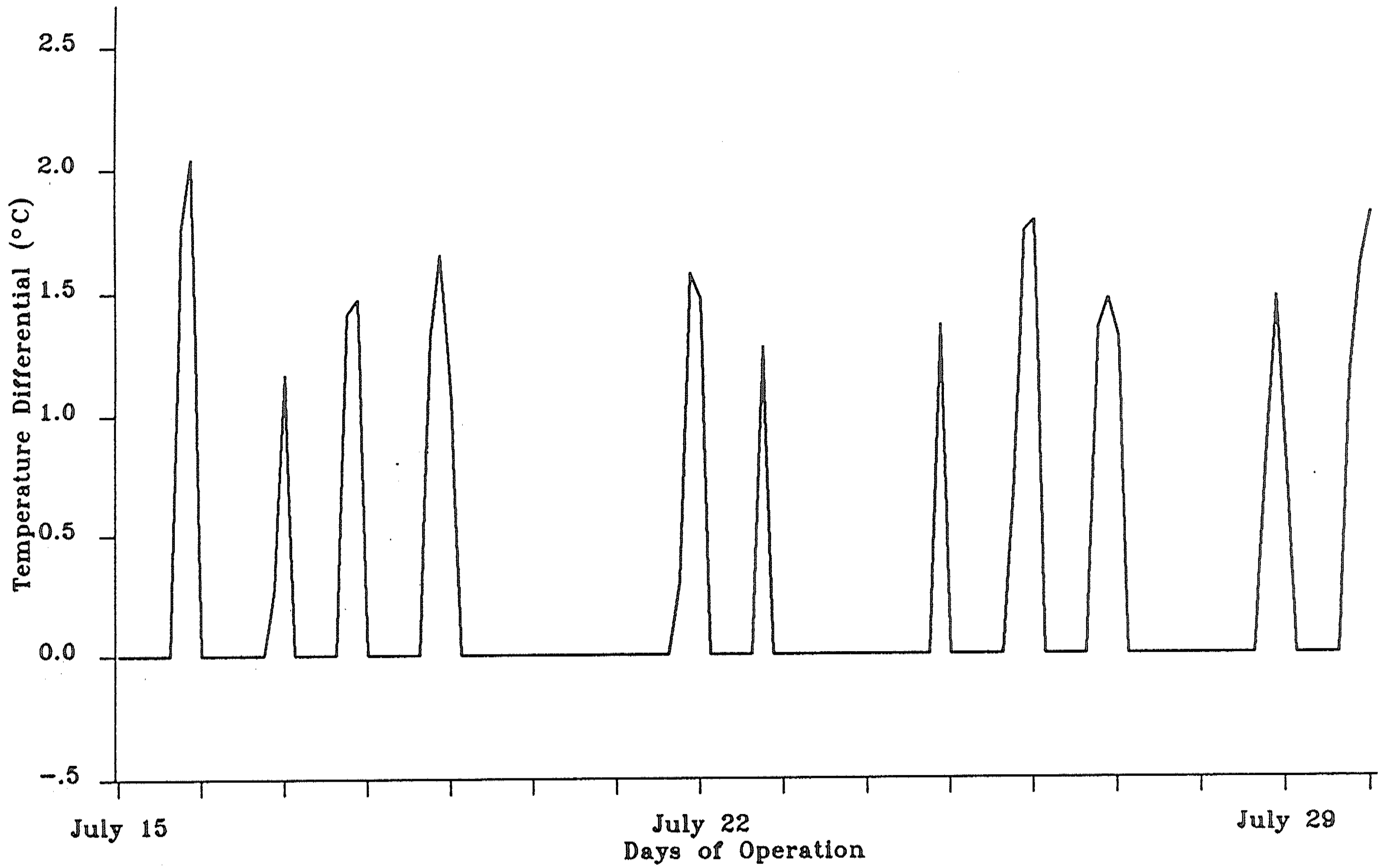


Figure 4.9 Summer simulation of pipe 4. Variation of nursery room temperature when using ambient air for ventilation as compared to soil-air tempering.

Table 4.6 Summer simulation room temperature results for the 29 m lateral length, soil-air tempering systems and the conventional system. May 27 to September 3, 1985.

	Conventional	Pipe 1	Pipe 3	Pipe 4
Ambient or Outlet Temperature (°C)				
Maximum	33.8	16.7	20.5	14.9
Minimum	-1.2	3.9	2.9	3.8
Average	16.5	11.0	13.0	9.9
Farrowing Room Temperature (°C)				
Maximum	39.1	37.1	37.8	36.8
Minimum	21.0	19.8	20.5	19.5
Average	23.4	22.8	23.1	22.7
Nursery Room Temperature (°C)				
Maximum	36.5	34.7	35.3	34.4
Minimum	27.0	26.9	26.9	26.9
Average	27.4	27.2	27.3	27.2

4.3 Economic Analysis

The analysis of the systems made a number of assumptions with respect to financial variables. Values were arbitrarily selected as being representative of the long term financial expectations. Sensitivity analyses of each of the variables were performed to ensure there were no material errors. The assumptions were as follows;

1. The entire capital cost of each system was paid by a loan amortized over the life of the system;

2. The real rate of interest was assumed to be 12%;

3. The cost of electricity was assumed to be \$0.03 /kW·h; and

4. The service life of the system was taken as 20 years.

The net present value (NPV) and the equivalent annual worth (EAW) were calculated for each of the soil-air tempering and conventional systems. The basis of the comparisons was the net difference in cost of capital and operation between the soil-air systems and a conventional system. Based upon the previous assumptions, the values for the systems have been presented in Table 4.7. Annual energy costs were extrapolated to a three month heating season. 150% of the calculated heat requirements for each system (kW·h) (Tables 4.2 to 4.4) were taken as the annual heating requirements. The cash flows used for the net present value calculations have been presented in Appendix C.

From Table 4.7, it was evident that none of the systems were competitive with a conventional system. The principle reason was the high capital cost of each of the soil-air tempering systems. Sensitivity analyses of the cost of electricity, and the real rate of interest are presented in Tables 4.8, and 4.9. These tables show that the above variables have little effect on the NPV's of the systems.

Some would argue that the capital costs of the soil-air tempering systems were too high, and that cost savings could have been found through the use of alternative materials and "Do-it-yourself" labour. The cost estimates were prepared assuming a turnkey philosophy for comparable new structures. Specific scenarios may have an inherent economic advantage, but for the majority of cases these estimates should be representative of most operations.

Table 4.7 Net present value and equivalent annual worth of soil-air tempering and conventional systems.

System No.	Pipe No.	Lateral Length (m)	Capital Cost ($)	Annual Energy Costs	Net Present Value	Equivalent Annual Worth
1	1	29	119,866	407	(124,785)	(10,025)
2	1	20	100,807	474	(106,540)	(8,563)
3	3	29	61,256	592	(68,406)	(5,507)
4	3	20	55,594	657	(68,532)	(5,118)
5	4	29	63,410	360	(67,755)	(5,448)
6	4	29	52,217	495	(58,200)	(4,685)
Conv.	_	—	8,250	916	(19,325)	(1,578)

Note: Parentheses () indicate negative cash values.

The use of PVC pipe and fittings contributed the largest cost factor to the systems. The cost estimates were based on the use of Series 10, gasketed joint, pressure pipe. This is not an unreasonable assumption, if the depth of bury and the use of smooth wall were to be maintained. The depth of bury dictated that a semi-rigid pipe would be necessary to prevent buckling. There would be little difference in the cost of using either sewer or pressure pipe, made from PVC or high density polyethylene (HDPE) pipe.

Table 4.8 Sensitivity of Net Present Value to cost of electricity.

Real Interest Rate 5.0%
Number of Years 20

Cost of Electricity ($/kW·h) System No.	0.015	0.03	0.035	0.04	0.05	0.10
1	(122,325)	(124,785)	(125,605)	(126,425)	(128,065)	(136,264)
2	(103,674)	(106,540)	(107,496)	(108,451)	(110,362)	(119,918)
3	(64,831)	(68,406)	(69,598)	(70,789)	(73,172)	(85,089)
4	(59,563)	(63,532)	(64,855)	(66,178)	(68,824)	(82,055)
5	(65,582)	(67,755)	(68,479)	(69,203)	(70,652)	(77,894)
6	(55,209)	(58,200)	(59,197)	(60,194)	(62,189)	(72,160)
Conv.	(13,787)	(19,325)	(21,171)	(23,016)	(26,708)	(45,166)

Table 4.9 Sensitivity of Net Present Value to real rate of interest.

Cost Of Electricity 0.03 $/kWh
Rate of Inflation 5.0%
Number of Years 20

Interest Rate (%) System No.	3	3.5	4	5	5.5	6
1	(125,696)	(125,446)	(125,212)	(124,785)	(124,591)	(124,408)
2	(107,602)	(107,311)	(107,038)	(106,540)	(106,314)	(106,100)
3	(69,730)	(69,367)	(69,026)	(68,406)	(68,123)	(67,857)
4	(65,002)	(64,599)	(64,221)	(63,532)	(63,218)	(62,923)
5	(68,560)	(68,339)	(68,132)	(67,755)	(67,583)	(67,422)
6	(59,308)	(59,004)	(58,719)	(58,200)	(57,964)	(57,741)
Conv.	(21,376)	(20,813)	(20,286)	(19,325)	(18,887)	(18,475)

The use of corrugated polyethylene drainage pipe was discounted because of the corrugated wall profile of the pipe. The corrugations would have different heat transfer characteristics due to greater air turbulence. This would have invalidated the temperature database, and hence the cost projections. It was thought that corrugations might lead to the accumulation of debris within the pipes. This could cause flow restrictions or the creation of pathogens within the pipes.

The cost of high density polyethylene (HDPE) corrugated drain tube is more attractive than PVC pipe, though the larger diameters loose some of their cost advantage. HDPE fittings are also cheaper than PVC, but they will not be as watertight. A typical cost for 150 mm diameter corrugated drain tube is approximately \$3.00/m from past experience. This represents a savings of 37.5% over the cost of PVC. If the system described in Table 3.12 is used as an example, and it is assumed that the same discount would apply for all materials, a savings of \$8,186.70 is realized. The cost of installation is not discounted, because at a 3.0 m depth the HDPE tube can not be plowed, and more care is necessary when placing the tube to maintain grade and compaction.

The other key element of the capital cost was the installation of the pipe. Arguments have been presented in favour of lowering the installation costs on the basis that many operators have both the time and the equipment to perform this type of work. This argument is not true in all cases, and thus was not used in the analysis of the model. Further to this end, a backhoe capable of excavating a 3.0 m trench would be far to valuable to have sitting idle. Operators with this type of equipment should look for more economic means of utilizing their equipment.

With respect to having ample free time, most farmers and especially livestock operators would disagree quite strongly with that assumption. Typical pipeline crews laying PVC pipe would have employed 4 to 5 crew members to safely perform the job. Using a typical farm backhoe, it would take almost a full month to install a system using either Pipe 1 or Pipe 4. This time estimate assumed that the excavation was opened, pipe laid, and the excavation closed within one day for each lateral. The pipe should also be laid during the warmer months, so that frost and snow would not impede the excavation. Based on the above arguments, the laying of the pipe would be beyond the abilities of the average farm operation.

Table 4.10 presents the breakeven capital costs for each of the systems considered, based on the NPV of the conventional system. Each of the system's capital costs were calculated assuming a NPV of ($19,325). The values in the table indicate that the supply and installation cost of the pipe has to be around $10/m to $15/m to justify a system.

Table 4.10 Breakeven capital costs for soil-air tempering systems based on the NPV of a conventional system.

System No.	Pipe No.	Pipe Dia. (mm)	Lateral Length (m)	Total Pipe Installed (m)	Breakeven Capital Cost
1	1	250	29	1,036	14,405
2	1	250	20	663	13,592
3	3	250	29	475	12,175
4	3	250	20	328	11,387
5	4	150	29	1,036	14,980
6	4	150	20	663	13,342

4.4 Subjects of Future Study

The thesis assumptions have created a number of shortcomings in the model which affect the overall system costs and economics. It may be possible to reduce the costs or improve the economics of the tempering systems by reducing some of the limitations placed on them by the design basis. The following paragraphs describe some of the potential areas of study that may individually or in concert make soil-air tempering a feasible solution.

A longer heating season, which included intermittent operation through the spring and fall, might increase the benefit of the tempering system. The tempering system was not modeled through the transition seasons because Murray (1987) found that continuous operation at these times tended to lessen the benefits later in the heating/cooling seasons. As an example, operating during warm spring days warms the soil temperature before the heat of summer when the maximum benefit occurs. The same would hold true in the fall when operation tends to cool the soil. Intermittent operation through the spring and fall might provide an optimum solution whereby the maximum heating and cooling values are achieved.

The potential benefits of summer cooling should be investigated. Currently air conditioning is not used for a typical Western Canadian operation, though there may be some benefit to maintaining the room temperatures through the summer months. By increasing the airflow of the tempering systems through the summer months, the systems contribute a larger proportion of the ventilation requirement, and result in greater cooling. This could also be operated on an intermittent basis to optimize the soil as a heat sink.

The high capital cost of the tempering systems required either a reduction in the material quantities or costs. Material quantities might be reduced through the use of materials with improved heat transfer characteristics or smaller diameters. The use of alternate materials for the pipe and fittings could also significantly affect the overall costs of the systems.

From an economic standpoint, it may not make sense to implement the entire system at once. Distributing the capital costs over a period of time might improve the overall value of the system. Shorter lengths and fewer laterals would contribute less heat to the system, but is this outweighed by the cost savings.

5.0 Conclusions

The principal conclusion of this thesis is that Soil-Air Tempering is not an economically feasible alternative for pre-treating ventilation air into a swine farrowing to nursery operation. Based upon the temperature database generated by Murray (1987), none of the pipe configurations compared favourably against a conventional, electric heat system.

The EAW of a conventional system was estimated to be ($1,104), which represented an equivalent annual cost. This value was base upon the estimated annual consumption of electricity, and an annual rate of inflation of 5% over a 20 year period. There were no associated capital costs. The EAW's of the soil-air tempering systems ranged from ($6,104) to ($14,998).

The break even capital costs of the soil-air tempering systems were calculated to be equivalent to the NPV of the conventional system. The NPV of the conventional system was ($8,250). The values shown in Table 4.12 illustrated that the cost of materials and labour would have to be drastically reduced to make these alternatives viable.

Rudimentary sensitivity analyses showed that interest rates, inflation, and the cost of electricity had little effect on the economics of the systems.

The winter simulations predicted ventilation rates for moisture control within the established guidelines of Agriculture Canada and VIDO. This would in part validate the results of the winter models.

The systems were perceived to have little benefit for summer cooling. The summer simulation showed little difference in room temperature between the soil-air tempering systems and the absence of any cooling. There was no attempt made to develop a cost-benefit analysis of the systems for summer cooling, because the system costs were of an order of magnitude greater than a conventional system.

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APPENDIX A

List of Programs

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1 REM WINMOD1.BAS                              WRITTEN: FEBRUARY 19, 1987
2 REM SOIL-AIR TEMPERING SIMULATION FOR WINTER CONDITIONS.  THE PUPRPOSE OF THE
      MODEL IS TO DETERMINE THE NUMBER OF LATERALS NECESSARY FOR EACH PIPE
      CONFIGURATION.
3 REM THIS PROGRAM READS DATA FROM PREPARED DATA FILES D?00?.PRN FROM A HARD
      DISK.   IT CREATES NEW FILES TC?0?.PRN ALSO STORED ON THE HARD DISK.
6 REM THE PROGRAM REQUIRES NO INPUTS FROM THE USER, BUT THE SUBDIRECTORY
      CONTAINING THE TEMPERATURE DATA MUST BE ACCESSED BEFORE OPERATION OF THE
      PROGRAM.  THE PROGRAM RUNS ON GWBASIC.
7 REM THE PROGRAM ASSUMES THAT THE HEAT BALANCE BETWEEN THE FARROWING, NURSERY,
      AND HALLS EQUILIBRIATES WITH TIME. THIS HEAT BALANCE WILL INDICATE EITHER
      THE NEED FOR SUPPLEMENTAL HEAT, OR TEMPERATURE CONTROL.
8 REM THE PROGRAM OUTPUT CONSISTS OF AMBIENT AND OUTLET TEMPERATURE, THE HEAT
      BALANCE (W), AND THE NUMBER OF PIPES NECESSARY FOR MOISTURE CONTROL.
9 REM THERE IS NO ACCOUNTING FOR TEMPERATURE CONTROL VENTILATION BUILT INTO THIS
      PROGRAM. THE HEAT BALANCE ALONE WOULD INDICATE THE NEED FOR FURTHER
      VENTILATION.
10 REM PROGRAM CONSTANTS
20 REM AREAS
30 AR=682.9: AG=75:AC(0)=648: AC(1)=377.4: AC(2)=151.2: AC(3)=119.4
40 AW(1)=118: AW(2)=105: AW(3)=50: AIW(1)=118: AIW(2)=105: AIW(3)=223
50 REM OUTSIDE FLOOR PERIMETERS
60 P(1)=47.2: P(2)=49.2: P(3)=20
70 REM HEAT TRANSMISSION COEFFICIENTS
80 UC=.286: UW=UC: UR=2.94: UG=4.01: UFL=.714: UIW=1.72
90 REM RELATIVE HUMIDITIES AND INITIAL TEMPERATURES
100 RH(1)=.75: RH(2)=.75: RH(0)=.95
110 T(1)=21: T(2)=27
120 REM SENSIBLE AND LATENT HEAT PRODUCTION OF ANIMALS
130 NA(1)=60: NA(2)=510: NA(3)=0
140 SHEAT(1)=248: SHEAT(2)=41.7: SHEAT(3)=0
150 LHEAT(1)=394: LHEAT(2)=54.3: LHEAT(3)=0
160 REM MSC. VARIABLES
170 PA=101.325: DT3=0
180 DIM KT3(20)
190 DIM KTATT(20)
200 KEY OFF
210 CLS
211 INPUT "INPUT PIPE NUMBER (1,3,OR 4)";PN
212 IF PN=1 OR PN=4 THEN AF=.05: REM PIPES 1 & 4 AIRFLOW .05 m^3/s
213 IF PN=3 THEN AF=.1: REM PIPE 3 AIRFLOW .1 m^3/s
214 CLS
220 INPUT "INPUT STARTING FILE NUMBER";START
230 PRINT
240 INPUT "INPUT LAST FILE NUMBER";LAST
250 CLS
260 FOR X= START TO LAST
270 IF X<10 THEN FILE2$="TC"+MID$(STR$(PN),2)+"0"+MID$(STR$(X),2)
280 IF X<10 THEN FILE$="D"+MID$(STR$(PN),2)+"00"+MID$(STR$(X),2): GOTO 310
290 FILE2$="TC"+MID$(STR$(PN),2)+MID$(STR$(X),2)
300 FILE$="D"+MID$(STR$(PN),2)+"0"+MID$(STR$(X),2)
310 OPEN "R",#1,"C:"+FILE$+".PRN",88
320 FIELD #1,1 AS Q$(1),10 AS DA$,1 AS Q$(2),1 AS Q$(3),8 AS TI$,1 AS Q$(4),40
    AS DUMMY$,8 AS SEN$(6),8 AS SEN$(7),8 AS SEN$(8),1 AS RET$,1 AS LF$
330 OPEN "R",#2,"C:"+FILE2$+".PRN",72
340 FIELD #2,1 AS Q$(1),10 AS DTE$,1 AS Q$(2),1 AS Q$(3),8 AS TME$,1 AS Q$(4),
    4 AS DUM$(1),8 AS TEMP$,4 AS DUM$(2),8 AS OUTTEMP$,4 AS DUM$(3),8 AS DQ$,
    4 AS DUM$(4),8 AS PIPE$,1 AS RET$, 1 AS LF$
350 GET #1,1
360 REC = VAL(DA$)
370 GOSUB 7000
380 FOR I = 2 TO REC
390 LOCATE 12,6:PRINT "COMPUTER PROGRAM IN PROGRESS"
400 LOCATE 25,10: PRINT "RECORD #";I,"  FILE:";FILE$+".PRN"
410 GET #1,I
420 TA=(VAL(SEN$(7))+VAL(SEN$(8)))/2
430 GOSUB 5000
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440 GOSUB 5200
450 T(0)=TA: T(4)=VAL(SEN$(6))
460 GOSUB 5400
470 GOSUB 5800
480 GOSUB 8000
490 NEXT I
500 CLOSE #1: CLOSE #2
510 NEXT X
520 CLOSE
530 CLS
540 LOCATE 12,27: PRINT "COMPUTER PROGRAM FINISHED"
550 END
5000 REM SUBROUTINE WHICH CALCULATES THE TEMPERATURE OF THE HALLWAYS AND THE
        ATTIC. IT IS AN ITERATIVE PROCESS DEPENDING ON CONSECUTIVE VALUES.
5010 Y=1
5020 KT3(Y) = ((T(1)*AIW(1)+T(2)*AIW(2))*UIW+TA*AW(3)*UW)/((AIW(1)+AIW(2))*UIW+
              AW(3)*UW)
5030 KTATT(Y) = ((T(1)*AC(1)+T(2)*AC(2)+KT3(Y)*AC(3))*UC+TA*(AR*UR+AG*UG))/(UC*
                (AC(1)+AC(2)+AC(3))+AR*UR+AG*UG)
5040 Z=2
5050 KT3(Z) = ((T(1)*AIW(1)+T(2)*AIW(2))*UIW+TA*AW(3)*UW+KTATT(Z-1)*AC(3)*UC)/((
                AIW(1)+AIW(2))*UIW+AW(3)*UW+AC(3)*UC)
5060 KTATT(Z) = ((T(1)*AC(1)+T(2)*AC(2)+KT3(Z)*AC(3))*UC+TA*(AR*UR+AG+UG))/(UC*
                (AC(1)+AC(2)+AC(3))+AR*UR+AG*UG)
5070 Y=Z
5080 IF ABS(KT3(Z)-KT3(Z-1))<.01 AND ABS(KTATT(Z)-KTATT(Z-1))<.01 THEN GOTO 5100
5090 Z=Z+1: GOTO 5050
5100 T(3)=KT3(Z): TAT=KTATT(Z)
5110 RETURN
5200 REM SUBROUTINE WHICH CALCULATES THE HEAT BALANCE OF THE BUILDING WITHOUT
        VENTILATION
5210 FOR J = 1 TO 3
5220     Q1(J)=AW(J)*UW*(T(J)-TA)
5230     Q2(J)=AIW(J)*UIW*(T(J)-T(3))
5240     Q2(3)=(AIW(1)*(T(3)-T(1))+AIW(2)*(T(3)-T(2)))*UIW
5250     Q3(J)=AC(J)*UC*(T(J)-TAT)
5260     Q4(J)=UFL*P(J)*(T(J)-TA)
5270     Q5(J)=NA(J)*SHEAT(J)
5280     Q5(1)=Q5(1)+NA(1)*250
5290     Q6(J)=Q5(J)-(Q1(J)+Q2(J)+Q3(J)+Q4(J))
5300 NEXT J
5310 RETURN
5400 REM SUBROUTINE WHICH CALCULATES THE PSYCHROMETRIC PROPERTIES OF THE MODELED
        CONTROL VOLUMES. AMBIENT,FARROWING,NURSERY,HALLWAYS,AND PIPE OUTLET.
5410 FOR J=0 TO 4
5420    T=T(J)+273
5430     IF T<273 THEN GOTO 5630
5440    PS= 89.63-7512/T+.024*T-1.165E-05*T^2-1.281E-08*T^3+2.1E-11*T^4
          -12.15*LOG(T)
5450    PS(J)=EXP(PS)
5460 NEXT J
5470 FOR J=0 TO 2
5480    PW(J)=RH(J)*PS(J)
5490    W(J)=.622*(PW(J)/(PA-PW(J)))
5500 NEXT J
5510 W(3)=W(0): W(4)=W(0)
5520 PW(3)=PW(0): RH(3)=PW(3)/PS(3)
5530 PW(4)=PW(0): RH(4)=PW(4)/PS(4)
5540 FOR J=0 TO  4
5550    HS(J)=1.01*(T(J)-0)
5560    HL(J)=W(J)*(2501+1.78*(T(J)-0))
5570    HT(J)=HS(J)+HL(J)
5580 REM ENTHALPY VALUES VALID FOR -50<T>110 CELSIUS
5590    SVOL(J)=(.287*(T(J)+273))/(PA-PW(J))
5600    DEN(J)=(1+W(J))/SVOL(J)
5610 NEXT J
5620 RETURN
5630 PS=24.28-6238/T-.3444*LOG(T)
5640 GOTO 5450
5800 REM SUBROUTINE WHICH CALCULATES THE VENTILATION RATES FOR THE BUILDING.
        FARROWING,NURSERY, AND TOTAL.
5810 REM CALCULATION OF VENT AS A FUNCTION OF LATENT HEAT
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5820 FOR J =1 TO 2
5830    DLAT(J)=HL(J)-HL(3)
5840    VLAT(J)=(LHEAT(J)*NA(J)*3.6)/(DLAT(J)*DEN(3))
5850    Q7(J)=Q6(J)-((HS(J)-HS(3))*(LHEAT(J)*NA(J)/DLAT(J)))
5860 NEXT J
5870 VLAT=VLAT(1)+VLAT(2)
5880 Q7(3)=0
5890 REM          MASS FLOW OF THE VENTILATION AIR IN kg/s.
5900 MFLOW=(LHEAT(1)*NA(1)/(DLAT(1)*1000))+(LHEAT(2)*NA(2)/(DLAT(2)*1000))
5910 REM          HEAT EXTRACTED BY THE SYSTEM IN kW.
5930 REM          NUMBER OF PIPES REQUIRED TO DELIVER THE VENTILATION NEEDS FOR
                  HUMIDITY CONTROL
5940 PIPES=(MFLOW/DEN(4))/AF
5941 IF (PIPES-INT(PIPES)) > .5 THEN PINC = 1 ELSE PINC = 0
5942 PIPES = INT(PIPES) + PINC
5950 REM          THE HEAT LOST BY THE SYSTEM DUE TO THE ADDITION OF
                  VENTILATION AIR.
5960 Q8=MFLOW*(HS(4)-HS(3))*1000
5970 REM          INSTANTANEOUS HEAT BALANCE OF THE SYSTEM
5980 DQ=Q7(1)+Q7(2)+Q8
5990 REM          THE CHANGE IN ENERGY WITHIN THE BUFFER ZONE (HALLWAYS).
6000 MASS3=AC(3)*DEN(3)*2.5: REM kg
6010 VKJ=MFLOW*HS(4)*10800: REM kJ
6020 VKJ3=MASS3*HS(3)
6030 SMIX=(VKJ+VKJ3)/(MFLOW*10800+MASS3): REM kJ/kg
6040 NEWT3=SMIX/1.01: REM C
6050 IF ABS(NEWT3-T(3))<1 THEN RETURN
6060 T(3)=NEWT3
6070 TAT = ((T(1)*AC(1)+T(2)*AC(2)+NEWT3*AC(3))*UC+TA*(AR*UR+AG+UG))/(UC*
          (AC(1)+AC(2)+AC(3))+AR*UR+AG*UG)
6080 RETURN 440
7000 REM SUBROUTINE WHICH FORMATS THE TWO HEADING LINES FOR THE FILE
7010 LSET DTE$=STR$(REC+2): LSET TME$=FILE2$: LSET TEMP$="PIPE #"+MID$(STR$(PN),
     2): LSET OUTTEMP$="": LSET DQ$="": LSET PIPE$=""
7020 FOR G=1 TO 4: LSET Q$(G)=CHR$(34):NEXT G
7030 LSET RET$=CHR$(13): LSET LF$=CHR$(10)
7040 FOR V=1 TO 4: LSET DUM$(V)="    ": NEXT V
7050 PUT #2,1
7051 LSET DTE$="   DATE": LSET TME$="   TIME": LSET TEMP$="AMB.TEMP": LSET
     OUTTEMP$="OUT.TEMP": LSET DQ$="  HEAT": LSET PIPE$="# PIPES"
7052 PUT #2,2
7060 LSET DTE$=" ": LSET TME$=" ": LSET TEMP$="  (C)": LSET OUTTEMP$ ="  (C)":
     LSET DQ$="BALANCE":LSET PIPE$=" "
7070 PUT #2,3
7080 RETURN
8000 REM SUBROUTINE WHICH PUTS THE CALCULATED VALUES ONTO DISK
8010 LSET DTE$=DA$
8020 LSET TME$=TI$
8030 LSET TEMP$=STR$(T(0))
8040 LSET OUTTEMP$=STR$(T(4))
8050 LSET DQ$=STR$(DQ)
8060 LSET PIPE$=STR$(PIPES)
8070 FOR G=1 TO 4: LSET Q$(G)=CHR$(34):NEXT G
8080 LSET RET$=CHR$(13): LSET LF$=CHR$(10)
8090 FOR V=1 TO 4: LSET DUM$(V)="    ": NEXT V
8100 PUT #2,I+2
8110 RETURN
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1 REM WINMOD2.BAS                                    WRITTEN: FEBRUARY 21, 1987
2 REM THIS PROGRAM SIMULATES THE MODEL STRUCTURE WITHOUT AIR TEMPERING. THE
      OBJECT TO DETERMINE THE AMOUNT OF HEAT NECESSARY UNDER THE GIVEN OPER-
      TING CONDITIONS FOR A CONVENTIONAL SYSTEM.
3 REM THIS PROGRAM READS DATA FROM PREPARED DATA FILES D?00?.PRN FROM A HARD
      DISK. IT CREATES NEW FILES TCA00?.PRN, ALSO STORED ON THE HARD DISK.
4 REM THE PROGRAM REQUIRES NO INPUTS FROM THE USER, BUT THE SUBDIRECTORY
      CONTAINING THE TEMPERATURE DATA MUST BE ACCESSED BEFORE OPERATION OF THE
      PROGRAM.  THE PROGRAM RUNS ON GWBASIC.
6 REM THE PROGRAM ASSUMES THAT THE HEAT BALANCE BETWEEN THE FARROWING, NURSERY,
      AND HALLS EQUILIBRIATES WITH TIME. THIS HEAT BALANCE WILL INDICATE EITHER
      THE NEED FOR SUPPLEMENTAL HEAT, OR TEMPERATURE CONTROL.
7 REM THE PROGRAM OUTPUT CONSISTS OF AMBIENT TEMPERATURE, THE HEAT BALANCE (W),
      THE MASS FLOW RATE, AND THE VENTILATION RATE NECESSARY FOR MOISTURE
      CONTROL.
8 REM THERE IS NO ACCOUNTING FOR TEMPERATURE CONTROL VENTILATION BUILT INTO THIS
      PROGRAM. THE HEAT BALANCE ALONE WOULD INDICATE THE NEED FOR FURTHER
      VENTILATION.
10 REM PROGRAM CONSTANTS
20 REM AREAS
30 AR=682.9: AG=75:AC(0)=648: AC(1)=377.4: AC(2)=151.2: AC(3)=119.4
40 AW(1)=118: AW(2)=105: AW(3)=50: AIW(1)=118: AIW(2)=105: AIW(3)=223
50 REM OUTSIDE FLOOR PERIMETERS
60 P(1)=47.2: P(2)=49.2: P(3)=20
70 REM HEAT TRANSMISSION COEFFICIENTS
80 UC=.286: UW=UC: UR=2.94: UG=4.01: UFL=.714: UIW=1.72
90 REM RELATIVE HUMIDITIES AND INITIAL TEMPERATURES
100 RH(1)=.75: RH(2)=.75: RH(0)=.95
110 T(1)=21: T(2)=27
120 REM SENSIBLE AND LATENT HEAT PRODUCTION OF ANIMALS
130 NA(1)=60: NA(2)=510: NA(3)=0
140 SHEAT(1)=248: SHEAT(2)=41.7: SHEAT(3)=0
150 LHEAT(1)=394: LHEAT(2)=54.3: LHEAT(3)=0
160 REM MSC. VARIABLES
170 PA=101.325: DT3=0
180 DIM KT3(20)
190 DIM KTATT(20)
200 KEY OFF
210 CLS
220 INPUT "INPUT STARTING FILE NUMBER";START
230 PRINT
240 INPUT "INPUT LAST FILE NUMBER";LAST
250 CLS
260 FOR X= START TO LAST
270 IF X<10 THEN FILE2$="TCA40"+MID$(STR$(X),2)
280 IF X<10 THEN FILE$="D400"+MID$(STR$(X),2): GOTO 310
290 FILE2$="TCA4"+MID$(STR$(X),2)
300 FILE$="D40"+MID$(STR$(X),2)
310 OPEN "R",#1,"C:"+FILE$+".PRN",88
320 FIELD #1,1 AS Q$(1),10 AS DA$,1 AS Q$(2),1 AS Q$(3),8 AS TI$,1 AS Q$(4),48 A
S DUMMY$,8 AS SEN$(7),8 AS SEN$(8),1 AS RET$,1 AS LF$
330 OPEN "R",#2,"C:"+FILE2$+".PRN",72
340 FIELD #2,1 AS Q$(1),10 AS DTE$,1 AS Q$(2),1 AS Q$(3),8 AS TME$,1 AS Q$(4),
    4 AS DUM$(1),8 AS TEMP$,4 AS DUM$(2),8 AS DQ$,4 AS DUM$(3),8 AS MFLOW$,
    4 AS DUM$(4),8 AS VLAT$,1 AS RET$, 1 AS LF$
350 GET #1,1
360 REC = VAL(DA$)
370 GOSUB 7000
380 FOR I = 2 TO REC
390 LOCATE 12,6:PRINT "COMPUTER PROGRAM IN PROGRESS"
400 LOCATE 25,10: PRINT "RECORD #";I,"  FILE:";FILE$+".PRN"
410 GET #1,I
420 TA=(VAL(SEN$(7))+VAL(SEN$(8)))/2
430 GOSUB 5000
440 GOSUB 5200
450 T(0)=TA
460 GOSUB 5400
470 GOSUB 5800
480 GOSUB 8000
490 NEXT I
500 CLOSE #1: CLOSE #2
510 NEXT X
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520 CLOSE
530 CLS
540 LOCATE 12,27: PRINT "COMPUTER PROGRAM FINISHED"
550 END
5000 REM SUBROUTINE WHICH CALCULATES THE TEMPERATURE OF THE HALLWAYS AND THE
        ATTIC. IT IS AN ITERATIVE PROCESS DEPENDING ON CONSECUTIVE VALUES.
5010 Y=1
5020 KT3(Y) = ((T(1)*AIW(1)+T(2)*AIW(2))*UIW+TA*AW(3)*UW)/((AIW(1)+AIW(2))*UIW+
             AW(3)*UW)
5030 KTATT(Y) = ((T(1)*AC(1)+T(2)*AC(2)+KT3(Y)*AC(3))*UC+TA*(AR*UR+AG*UG))/(UC*
               (AC(1)+AC(2)+AC(3))+AR*UR+AG*UG)
5040 Z=2
5050 KT3(Z) = ((T(1)*AIW(1)+T(2)*AIW(2))*UIW+TA*AW(3)*UW+KTATT(Z-1)*AC(3)*UC)/((
               AIW(1)+AIW(2))*UIW+AW(3)*UW+AC(3)*UC)
5060 KTATT(Z) = ((T(1)*AC(1)+T(2)*AC(2)+KT3(Z)*AC(3))*UC+TA*(AR*UR+AG+UG))/(UC*
               (AC(1)+AC(2)+AC(3))+AR*UR+AG*UG)
5070 Y=Z
5080 IF ABS(KT3(Z)-KT3(Z-1))<.01 AND ABS(KTATT(Z)-KTATT(Z-1))<.01 THEN GOTO 5100
5090 Z=Z+1: GOTO 5050
5100 T(3)=KT3(Z): TAT=KTATT(Z)
5110 RETURN
5200 REM SUBROUTINE WHICH CALCULATES THE HEAT BALANCE OF THE BUILDING WITHOUT
        VENTILATION
5210 FOR J = 1 TO 3
5220     Q1(J)=AW(J)*UW*(T(J)-TA)
5230     Q2(J)=AIW(J)*UIW*(T(J)-T(3))
5240     Q2(3)=(AIW(1)*(T(3)-T(1))+AIW(2)*(T(3)-T(2)))*UIW
5250     Q3(J)=AC(J)*UC*(T(J)-TAT)
5260     Q4(J)=UFL*P(J)*(T(J)-TA)
5270     Q5(J)=NA(J)*SHEAT(J)
5280     Q5(1)=Q5(1)+NA(1)*250
5290     Q6(J)=Q5(J)-(Q1(J)+Q2(J)+Q3(J)+Q4(J))
5300 NEXT J
5310 RETURN
5400 REM SUBROUTINE WHICH CALCULATES THE PSYCHROMETRIC PROPERTIES OF THE MODELED
        CONTROL VOLUMES. AMBIENT,FARROWING,NURSERY,HALLWAYS,AND PIPE OUTLET.
5410 FOR J=0 TO 3
5420    T=T(J)+273
5430     IF T<273 THEN GOTO 5630
5440    PS= 89.63-7512/T+.024*T-1.165E-05*T^2-1.281E-08*T^3+2.1E-11*T^4
           -12.15*LOG(T)
5450    PS(J)=EXP(PS)
5460 NEXT J
5470 FOR J=0 TO 2
5480    PW(J)=RH(J)*PS(J)
5490    W(J)=.622*(PW(J)/(PA-PW(J)))
5500 NEXT J
5510 W(3)=W(0)
5520 PW(3)=PW(0): RH(3)=PW(3)/PS(3)
5540 FOR J=0 TO 3
5550    HS(J)=1.01*(T(J)-0)
5560    HL(J)=W(J)*(2501+1.78*(T(J)-0))
5570    HT(J)=HS(J)+HL(J)
5580 REM ENTHALPY VALUES VALID FOR -50<T>110 CELSIUS
5590    SVOL(J)=(.287*(T(J)+273))/(PA-PW(J))
5600    DEN(J)=(1+W(J))/SVOL(J)
5610 NEXT J
5620 RETURN
5630 PS=24.28-6238/T-.3444*LOG(T)
5640 GOTO 5450
5800 REM SUBROUTINE WHICH CALCULATES THE VENTILATION RATES FOR THE BUILDING.
        FARROWING,NURSERY, AND TOTAL.
5810 REM CALCULATION OF VENT AS A FUNCTION OF LATENT HEAT
5820 FOR J =1 TO 2
5830    DLAT(J)=HL(J)-HL(3)
5840    VLAT(J)=(LHEAT(J)*NA(J)*3.6)/(DLAT(J)*DEN(3))
5850    Q7(J)=Q6(J)-((HS(J)-HS(3))*(LHEAT(J)*NA(J)/DLAT(J)))
5860 NEXT J
5870 VLAT=VLAT(1)+VLAT(2)
5880 Q7(3)=0
5890 REM        MASS FLOW OF THE VENTILATION AIR IN kg/s.
5900 MFLOW=(LHEAT(1)*NA(1)/(DLAT(1)*1000))+(LHEAT(2)*NA(2)/(DLAT(2)*1000))
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5950 REM        THE HEAT LOST BY THE SYSTEM DUE TO THE ADDITION OF
                VENTILATION AIR.
5960 Q8=MFLOW*(HS(0)-HS(3))*1000
5970 REM        INSTANTANEOUS HEAT BALANCE OF THE SYSTEM
5980 DQ=Q7(1)+Q7(2)+Q8
5990 REM        THE CHANGE IN ENERGY WITHIN THE BUFFER ZONE (HALLWAYS).
6000 MASS3=AC(3)*DEN(3)*2.5: REM kg
6010 VKJ=MFLOW*HS(0)*10800: REM kJ
6020 VKJ3=MASS3*HS(3): REM kJ
6030 SMIX=(VKJ+VKJ3)/(MFLOW*10800+MASS3): REM kJ/kg
6040 NEWT3=SMIX/1.01
6050 IF ABS(NEWT3-T(3))<1 THEN RETURN
6060 T(3)=NEWT3
6070 TAT = ((T(1)*AC(1)+T(2)*AC(2)+NEWT3*AC(3))*UC+TA*(AR*UR+AG+UG))/(UC*
          (AC(1)+AC(2)+AC(3))+AR*UR+AG*UG)
6080 RETURN 440
7000 REM SUBROUTINE WHICH FORMATS THE TWO HEADING LINES FOR THE FILE
7010 LSET DTE$=STR$(REC+2): LSET TME$=FILE2$: LSET TEMP$="PIPE #4": LSET
     DQ$="": LSET MFLOW$="PROGRAM:": LSET VLAT$="WINMOD2"
7020 FOR G=1 TO 4: LSET Q$(G)=CHR$(34):NEXT G
7030 LSET RET$=CHR$(13): LSET LF$=CHR$(10)
7040 FOR V=1 TO 4: LSET DUM$(V)="    ": NEXT V
7050 PUT #2,1
7051 LSET DTE$="   DATE": LSET TME$="  TIME": LSET TEMP$="AMB.TEMP": LSET
     DQ$="  HEAT": LSET MFLOW$="  MASS": LSET VLAT$="  VENT"
7052 PUT #2,2
7060 LSET DTE$=" ": LSET TME$=" ": LSET TEMP$="  (C)": LSET DQ$="BALANCE":
     LSET MFLOW$=" kg/s": LSET VLAT$="  m^3/h"
7070 PUT #2,3
7080 RETURN
8000 REM SUBROUTINE WHICH PUTS THE CALCULATED VALUES ONTO DISK
8010 LSET DTE$=DA$
8020 LSET TME$=TI$
8030 LSET TEMP$=STR$(T(0))
8050 LSET DQ$=STR$(DQ)
8055 LSET MFLOW$=STR$(MFLOW)
8060 LSET VLAT$=STR$(VLAT)
8070 FOR G=1 TO 4: LSET Q$(G)=CHR$(34):NEXT G
8080 LSET RET$=CHR$(13): LSET LF$=CHR$(10)
8090 FOR V=1 TO 4: LSET DUM$(V)="    ": NEXT V
8100 PUT #2,I+2
8110 RETURN
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1 REM WINMOD3.BAS                          WRITTEN: FEBRUARY 21, 1987
2 REM REVISION OF PROGRAM WINMOD1.BAS. THIS SIMULATION WAS DESIGNED TO YIELD THE
      NUMBER OF LATERALS AND THE VENTILATION RATES NECESSARY FOR EACH SOIL-AIR
      CONCIFURATION.  29 m LATERALS WERE MODELED.
3 REM THIS PROGRAM READS DATA FROM PREPARED DATA FILES D*00*.PRN FROM A HARD
      DISK. IT CREATES NEW FILES TCB*07.PRN, ALSO STORED ON THE HARD DISK.
4 REM THE PROGRAM ASSUMES THAT THE HEAT BALANCE BETWEEN THE FARROWING, NURSERY,
      AND HALLS EQUILIBRIATES WITH TIME. THIS HEAT BALANCE WILL INDICATE EITHER
      THE NEED FOR SUPPLEMENTAL HEAT, OR TEMPERATURE CONTROL.
5 REM THE PROGRAM OUTPUT CONSISTS OF AMBIENT AND OUTLET TEMPERATURE, THE HEAT
      BALANCE (W), THE MASS FLOW RATE,THE VENTILATION RATE, AND THE NUMBER OF
      PIPES NECESSARY FOR MOISTURE CONTROL.
6 REM THE PROGRAM REQUIRES NO INPUTS FROM THE USER, BUT THE SUBDIRECTORY
      CONTAINING THE TEMPERATURE DATA MUST BE ACCESSED BEFORE OPERATION OF THE
      PROGRAM.
7 REM THE PROGRAM ASSUMES THAT THE HEAT BALANCE BETWEEN THE FARROWING, NURSERY,
      AND HALLS EQUILIBRIATES WITH TIME. THIS HEAT BALANCE WILL INDICATE EITHER
      THE NEED FOR SUPPLEMENTAL HEAT, OR TEMPERATURE CONTROL.
8 REM THE PROGRAM OUTPUT CONSISTS OF AMBIENT AND OUTLET TEMPERATURE, THE HEAT
      BALANCE (W), THE MASS FLOW RATE,THE VENTILATION RATE, AND THE NUMBER OF
      PIPES NECESSARY FOR MOISTURE CONTROL.
9 REM THERE IS NO ACCOUNTING FOR TEMPERATURE CONTROL VENTILATION BUILT INTO THIS
      PROGRAM. THE HEAT BALANCE ALONE WOULD INDICATE THE NEED FOR FURTHER
      VENTILATION.
10 REM PROGRAM CONSTANTS
20 REM AREAS
30 AR=682.9: AG=75:AC(0)=648: AC(1)=377.4: AC(2)=151.2: AC(3)=119.4
40 AW(1)=118: AW(2)=105: AW(3)=50: AIW(1)=118: AIW(2)=105: AIW(3)=223
50 REM OUTSIDE FLOOR PERIMETERS
60 P(1)=47.2: P(2)=49.2: P(3)=20
70 REM HEAT TRANSMISSION COEFFICIENTS
80 UC=.286: UW=UC: UR=2.94: UG=4.01: UFL=.714: UIW=1.72
90 REM RELATIVE HUMIDITIES AND INITIAL TEMPERATURES
100 RH(1)=.75: RH(2)=.75: RH(0)=.95
110 T(1)=21: T(2)=27
120 REM SENSIBLE AND LATENT HEAT PRODUCTION OF ANIMALS
130 NA(1)=60: NA(2)=510: NA(3)=0
140 SHEAT(1)=248: SHEAT(2)=41.7: SHEAT(3)=0
150 LHEAT(1)=394: LHEAT(2)=54.3: LHEAT(3)=0
160 REM MSC. VARIABLES
170 PA=101.325: DT3=0
180 DIM KT3(20)
190 DIM KTATT(20)
200 KEY OFF
210 CLS
211 INPUT "INPUT  PIPE NUMBER (1,3,OR 4)";PN
212 IF PN=1 OR PN=4 THEN AF=.05: REM PIPES 1 & 4 AIR FLOW .05 m^3/s
213 IF PN =3 THEN AF=.1: REM PIPE 3 AIRFLOW .1 m^3/s
214 CLS
220 INPUT "INPUT STARTING FILE NUMBER";START
230 PRINT
240 INPUT "INPUT LAST FILE NUMBER";LAST
250 CLS
260 FOR X= START TO LAST
270 IF X<10 THEN FILE2$="TCB"+MID$(STR$(PN),2)+"0"+MID$(STR$(X),2)
280 IF X<10 THEN FILE$="D"+MID$(STR$(PN),2)+"00"+MID$(STR$(X),2): GOTO 310
290 FILE2$="TCB"+MID$(STR$(PN),2)+MID$(STR$(X),2)
300 FILE$="D"+MID$(STR$(PN),2)+"0"+MID$(STR$(X),2)
310 OPEN "R",#1,"C:"+FILE$+".PRN",88
320 FIELD #1,1 AS Q$(1),10 AS DA$,1 AS Q$(2),1 AS Q$(3),8 AS TI$,1 AS Q$(4),40 A
S DUMMY$,8 AS SEN$(6),8 AS SEN$(7),8 AS SEN$(8),1 AS RET$,1 AS LF$
330 OPEN "R",#2,"C:"+FILE2$+".PRN",96
340 FIELD #2,1 AS Q$(1),10 AS DTE$,1 AS Q$(2),1 AS Q$(3),8 AS TME$,1 AS Q$(4),4
AS DUM$(1),8 AS TEMP$,4 AS DUM$(2),8 AS OUTTEMP$,4 AS DUM$(3),8 AS DQ$(1),4 AS D
UM$(4),8 AS MFLOW$(1),4 AS DUM$(5),8 AS VLAT$(1),4 AS DUM$(6),8 AS PIPE$,1 AS RE
T$, 1 AS LF$
350 GET #1,1
```

```
360 REC = VAL(DA$)
370 GOSUB 7000
380 FOR I = 2 TO REC
390 LOCATE 12,6:PRINT "COMPUTER PROGRAM IN PROGRESS"
400 LOCATE 25,10: PRINT "RECORD #";I,"  FILE:";FILE$+".PRN"
410 GET #1,I
420 TA=(VAL(SEN$(7))+VAL(SEN$(8)))/2
430 GOSUB 5000
440 GOSUB 5200
450 T(0)=TA: T(4)=VAL(SEN$(6))
460 GOSUB 5400
470 GOSUB 5800
480 GOSUB 8000
490 NEXT I
500 CLOSE #1: CLOSE #2
510 NEXT X
520 CLOSE
530 CLS
540 LOCATE 12,27: PRINT "COMPUTER PROGRAM FINISHED"
550 END
5000 REM SUBROUTINE WHICH CALCULATES THE TEMPERATURE OF THE HALLWAYS AND THE
        ATTIC. IT IS AN ITERATIVE PROCESS DEPENDING ON CONSECUTIVE VALUES.
5010 Y=1
5020 KT3(Y) = ((T(1)*AIW(1)+T(2)*AIW(2))*UIW+TA*AW(3)*UW)/((AIW(1)+AIW(2))*UIW+
            AW(3)*UW)
5030 KTATT(Y) = ((T(1)*AC(1)+T(2)*AC(2)+KT3(Y)*AC(3))*UC+TA*(AR*UR+AG*UG))/(UC*
              (AC(1)+AC(2)+AC(3))+AR*UR+AG*UG)
5040 Z=2
5050 KT3(Z) = ((T(1)*AIW(1)+T(2)*AIW(2))*UIW+TA*AW(3)*UW+KTATT(Z-1)*AC(3)*UC)/((
               AIW(1)+AIW(2))*UIW+AW(3)*UW+AC(3)*UC)
5060 KTATT(Z) = ((T(1)*AC(1)+T(2)*AC(2)+KT3(Z)*AC(3))*UC+TA*(AR*UR+AG+UG))/(UC*
              (AC(1)+AC(2)+AC(3))+AR*UR+AG*UG)
5070 Y=Z
5080 IF ABS(KT3(Z)-KT3(Z-1))<.01 AND ABS(KTATT(Z)-KTATT(Z-1))<.01 THEN GOTO 5100
5090 Z=Z+1: GOTO 5050
5100 T(3)=KT3(Z): TAT=KTATT(Z)
5110 RETURN
5200 REM SUBROUTINE WHICH CALCULATES THE HEAT BALANCE OF THE BUILDING WITHOUT
        VENTILATION
5210 FOR J = 1 TO 3
5220      Q1(J)=AW(J)*UW*(T(J)-TA)
5230      Q2(J)=AIW(J)*UIW*(T(J)-T(3))
5240      Q2(3)=(AIW(1)*(T(3)-T(1))+AIW(2)*(T(3)-T(2)))*UIW
5250      Q3(J)=AC(J)*UC*(T(J)-TAT)
5260      Q4(J)=UFL*P(J)*(T(J)-TA)
5270      Q5(J)=NA(J)*SHEAT(J)
5280      Q5(1)=Q5(1)+NA(1)*250
5290      Q6(J)=Q5(J)-(Q1(J)+Q2(J)+Q3(J)+Q4(J))
5300 NEXT J
5310 RETURN
5400 REM SUBROUTINE WHICH CALCULATES THE PSYCHROMETRIC PROPERTIES OF THE MODELED
        CONTROL VOLUMES. AMBIENT,FARROWING,NURSERY,HALLWAYS,AND PIPE OUTLET.
5410 FOR J=0 TO 4
5420     T=T(J)+273
5430      IF T<273 THEN GOTO 5630
5440     PS= 89.63-7512/T+.024*T-1.165E-05*T^2-1.281E-08*T^3+2.1E-11*T^4
           -12.15*LOG(T)
5450     PS(J)=EXP(PS)
5460 NEXT J
5470 FOR J=0 TO 2
5480     PW(J)=RH(J)*PS(J)
5490     W(J)=.622*(PW(J)/(PA-PW(J)))
5500 NEXT J
5510 W(3)=W(0): W(4)=W(0)
5520 PW(3)=PW(0): RH(3)=PW(3)/PS(3)
5530 PW(4)=PW(0): RH(4)=PW(4)/PS(4)
5540 FOR J=0 TO  4
5550     HS(J)=1.01*(T(J)-0)
5560     HL(J)=W(J)*(2501+1.78*(T(J)-0))
5570     HT(J)=HS(J)+HL(J)
5580 REM ENTHALPY VALUES VALID FOR -50<T>110 CELSIUS
5590     SVOL(J)=(.287*(T(J)+273))/(PA-PW(J))
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```
5600    DEN(J)=(1+W(J))/SVOL(J)
5610 NEXT J
5620 RETURN
5630 PS=24.28-6238/T-.3444*LOG(T)
5640 GOTO 5450
5800 REM SUBROUTINE WHICH CALCULATES THE VENTILATION RATES FOR THE BUILDING.
        FARROWING,NURSERY, AND TOTAL.
5810 REM CALCULATION OF VENT AS A FUNCTION OF LATENT HEAT
5820 FOR J =1 TO 2
5830    DLAT(J)=HL(J)-HL(3)
5840    VLAT(J)=(LHEAT(J)*NA(J)*3.6)/(DLAT(J)*DEN(3))
5850    Q7(J)=Q6(J)-((HS(J)-HS(3))*(LHEAT(J)*NA(J)/DLAT(J)))
5860 NEXT J
5870 VLAT=VLAT(1)+VLAT(2)
5880 Q7(3)=0
5890 REM         MASS FLOW OF THE VENTILATION AIR IN kg/s.
5900 MFLOW=(LHEAT(1)*NA(1)/(DLAT(1)*1000))+(LHEAT(2)*NA(2)/(DLAT(2)*1000))
5910 REM         HEAT EXTRACTED BY THE SYSTEM IN kW.
5930 REM         NUMBER OF PIPES REQUIRED TO DELIVER THE VENTILATION NEEDS FOR
                 HUMIDITY CONTROL
5940 PIPES=(MFLOW/DEN(4))/AF
5941 IF (PIPES-INT(PIPES)) > .5 THEN PINC = 1 ELSE PINC = 0
5942 PIPES = INT(PIPES) + PINC
5950 REM         THE HEAT LOST BY THE SYSTEM DUE TO THE ADDITION OF
                 VENTILATION AIR.
5960 Q8=MFLOW*(HT(4)-HT(3))*1000
5970 REM         INSTANTANEOUS HEAT BALANCE OF THE SYSTEM
5980 DQ=Q7(1)+Q7(2)+Q8
5990 REM         THE CHANGE IN ENERGY WITHIN THE BUFFER ZONE (HALLWAYS).
6000 MASS3=AC(3)*DEN(3)*2.5: REM kg
6010 VKJ=MFLOW*HS(4)*10800: REM kJ
6020 VKJ3=MASS33*HS(3): REM kJ
6030 SMIX=(VKJ+VKJ3)/(MFLOW*10800+MASS3): REM kJ/kg
6040 NEWT3=SMIX/1.01
6050 IF ABS(NEWT3-T(3))<1 THEN RETURN
6060 T(3)=NEWT3
6070 TAT = ((T(1)*AC(1)+T(2)*AC(2)+NEWT3*AC(3))*UC+TA*(AR*UR+AG+UG))/(UC*
           (AC(1)+AC(2)+AC(3))+AR*UR+AG*UG)
6080 RETURN 440
7000 REM SUBROUTINE WHICH FORMATS THE TWO HEADING LINES FOR THE FILE
7010 LSET DTE$=STR$(REC+2):LSET TME$=FILE2$:LSET TEMP$="PIPE #4":LSET OUTTEMP$
     ="":LSET DQ$(1)="PROGRAM:":LSET MFLOW$(1)="WINMOD3":LSET VLAT$(1)="":LSET
     PIPE$=""
7020 FOR G=1 TO 4: LSET Q$(G)=CHR$(34):NEXT G
7030 LSET RET$=CHR$(13): LSET LF$=CHR$(10)
7040 FOR V=1 TO 6: LSET DUM$(V)="    ": NEXT V
7050 PUT #2,1
7051 LSET DTE$="   DATE": LSET TME$="  TIME": LSET TEMP$="AMB.TEMP": LSET
     OUTTEMP$="OUT.TEMP": LSET DQ$(1)="  HEAT": LSET MFLOW$(1)="  MASS": LSET
     VLAT$(1)="  VENT.": LSET PIPE$="# PIPES"
7052 PUT #2,2
7060 LSET DTE$=" ": LSET TME$=" ": LSET TEMP$="  (C)": LSET OUTTEMP$ ="  (C)":
     LSET DQ$(1)="BALANCE":LSET MFLOW$(1)="  kg/s": LSET VLAT$(1)="  m^3/h":LSET
     PIPE$=" "
7070 PUT #2,3
7080 RETURN
8000 REM SUBROUTINE WHICH PUTS THE CALCULATED VALUES ONTO DISK
8010 LSET DTE$=DA$
8020 LSET TME$=TI$
8030 LSET TEMP$=STR$(T(0))
8040 LSET OUTTEMP$=STR$(T(4))
8050 LSET DQ$(1)=STR$(DQ)
8054 LSET MFLOW$(1)=STR$(MFLOW)
8056 LSET VLAT$(1)=STR$(VLAT)
8060 LSET PIPE$=STR$(PIPES)
8070 FOR G=1 TO 4: LSET Q$(G)=CHR$(34):NEXT G
8080 LSET RET$=CHR$(13): LSET LF$=CHR$(10)
8090 FOR V=1 TO 6: LSET DUM$(V)="    ": NEXT V
8100 PUT #2,I+2
8110 RETURN
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1 REM WINMOD4.BAS                            WRITTEN: FEBRUARY 21, 1988
2 REM REVISION OF PROGRAM WINMOD1.BAS. THIS SIMULATION WAS DESIGNED TO YIELD THE
    NUMBER OF LATERALS AND THE VENTILATION RATES NECESSARY FOR EACH OF THE
    SOIL-AIR TEMPERING CONFIGURATIONS. 20 m LATERALS WERE MODELED.
5 REM THIS PROGRAM READS DATA FROM PREPARED DATA FILES D*00*.PRN FROM A HARD
    DISK. IT CREATES NEW FILES TCD***.PRN, ALSO STORED ON THE HARD DISK.
6 REM THE PROGRAM REQUIRES NO INPUTS FROM THE USER, BUT THE SUBDIRECTORY
    CONTAINING THE TEMPERATURE DATA MUST BE ACCESSED BEFORE OPERATION OF THE
    PROGRAM.  THE PROGRAM OPERATES ON GWBASIC.
7 REM THE PROGRAM ASSUMES THAT THE HEAT BALANCE BETWEEN THE FARROWING, NURSERY,
    AND HALLS EQUILIBRIATES WITH TIME. THIS HEAT BALANCE WILL INDICATE EITHER
    THE NEED FOR SUPPLEMENTAL HEAT, OR TEMPERATURE CONTROL.
8 REM THE PROGRAM OUTPUT CONSISTS OF AMBIENT AND OUTLET TEMPERATURE, THE HEAT
    BALANCE (W), AND THE NUMBER OF PIPES NECESSARY FOR MOISTURE CONTROL.
9 REM THERE IS NO ACCOUNTING FOR TEMPERATURE CONTROL VENTILATION BUILT INTO THIS
    PROGRAM. THE HEAT BALANCE ALONE WOULD INDICATE THE NEED FOR FURTHER
    VENTILATION.
10 REM PROGRAM CONSTANTS
20 REM AREAS
30 AR=682.9: AG=75:AC(0)=648: AC(1)=377.4: AC(2)=151.2: AC(3)=119.4
40 AW(1)=118: AW(2)=105: AW(3)=50: AIW(1)=118: AIW(2)=105: AIW(3)=223
50 REM OUTSIDE FLOOR PERIMETERS
60 P(1)=47.2: P(2)=49.2: P(3)=20
70 REM HEAT TRANSMISSION COEFFICIENTS
80 UC=.286: UW=UC: UR=2.94: UG=4.01: UFL=.714: UIW=1.72
90 REM RELATIVE HUMIDITIES AND INITIAL TEMPERATURES
100 RH(1)=.75: RH(2)=.75: RH(0)=.95
110 T(1)=21: T(2)=27
120 REM SENSIBLE AND LATENT HEAT PRODUCTION OF ANIMALS
130 NA(1)=60: NA(2)=510: NA(3)=0
140 SHEAT(1)=248: SHEAT(2)=41.7: SHEAT(3)=0
150 LHEAT(1)=394: LHEAT(2)=54.3: LHEAT(3)=0
160 REM MSC. VARIABLES
170 PA=101.325: DT3=0
180 DIM KT3(20)
190 DIM KTATT(20)
200 KEY OFF
210 CLS
211 INPUT "INPUT THE PIPE NUMBER (1,3,OR 4)";PN
212 IF PN=1 OR PN=4 THEN AF=.05: REM PIPES 1 & 3 AIRFLOW .05 m^3/s
213 IF PN=3 THEN AF=.1: REM PIPE 3 AIRFLOW .10 m^3/s
214 CLS
220 INPUT "INPUT STARTING FILE NUMBER";START
230 PRINT
240 INPUT "INPUT LAST FILE NUMBER";LAST
250 CLS
260 FOR X= START TO LAST
270 IF X<10 THEN FILE2$="TCD"+MID$(STR$(PN),2)+"0"+MID$(STR$(X),2)
280 IF X<10 THEN FILE$="D"+MID$(STR$(PN),2)+"0"+MID$(STR$(X),2): GOTO 310
290 FILE2$="TCD"+MID$(STR$(PN),2)+MID$(STR$(X),2)
300 FILE$="D"+MID$(STR$(PN),2)+"0"+MID$(STR$(X),2)
310 OPEN "R",#1,"C:"+FILE$+".PRN",88
320 FIELD #1,1 AS Q$(1),10 AS DA$,1 AS Q$(2),1 AS Q$(3),8 AS TI$,1 AS Q$(4),32
    AS DUMMY$,8 AS SEN$(5),8 AS D$,8 AS SEN$(7),8 AS SEN$(8),1 AS RET$,1 AS LF$
330 OPEN "R",#2,"C:"+FILE2$+".PRN",72
340 FIELD #2,1 AS Q$(1),10 AS DTE$,1 AS Q$(2),1 AS Q$(3),8 AS TME$,1 AS Q$(4),
    4 AS DUM$(1),8 AS TEMP$,4 AS DUM$(2),8 AS OUTTEMP$,4 AS DUM$(3),8 AS DQ$,
    4 AS DUM$(4),8 AS PIPE$,1 AS RET$, 1 AS LF$
350 GET #1,1
360 REC = VAL(DA$)
370 GOSUB 7000
380 FOR I = 2 TO REC
390 LOCATE 12,6:PRINT "COMPUTER PROGRAM IN PROGRESS"
400 LOCATE 25,10: PRINT "RECORD #";I,"  FILE:";FILE$+".PRN"
410 GET #1,I
420 TA=(VAL(SEN$(7))+VAL(SEN$(8)))/2
430 GOSUB 5000
440 GOSUB 5200
450 T(0)=TA: T(4)=VAL(SEN$(5))
460 GOSUB 5400
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470 GOSUB 5800
480 GOSUB 8000
490 NEXT I
500 CLOSE #1: CLOSE #2
510 NEXT X
520 CLOSE
530 CLS
540 LOCATE 12,27: PRINT "COMPUTER PROGRAM FINISHED"
550 END
5000 REM SUBROUTINE WHICH CALCULATES THE TEMPERATURE OF THE HALLWAYS AND THE
        ATTIC. IT IS AN ITERATIVE PROCESS DEPENDING ON CONSECUTIVE VALUES.
5010 Y=1
5020 KT3(Y) = ((T(1)*AIW(1)+T(2)*AIW(2))*UIW+TA*AW(3)*UW)/((AIW(1)+AIW(2))*UIW+
             AW(3)*UW)
5030 KTATT(Y) = ((T(1)*AC(1)+T(2)*AC(2)+KT3(Y)*AC(3))*UC+TA*(AR*UR+AG*UG))/(UC*
                (AC(1)+AC(2)+AC(3))+AR*UR+AG*UG)
5040 Z=2
5050 KT3(Z) = ((T(1)*AIW(1)+T(2)*AIW(2))*UIW+TA*AW(3)*UW+KTATT(Z-1)*AC(3)*UC)/((
                 AIW(1)+AIW(2))*UIW+AW(3)*UW+AC(3)*UC)
5060 KTATT(Z) = ((T(1)*AC(1)+T(2)*AC(2)+KT3(Z)*AC(3))*UC+TA*(AR*UR+AG+UG))/(UC*
                (AC(1)+AC(2)+AC(3))+AR*UR+AG*UG)
5070 Y=Z
5080 IF ABS(KT3(Z)-KT3(Z-1))<.01 AND ABS(KTATT(Z)-KTATT(Z-1))<.01 THEN GOTO 5100
5090 Z=Z+1: GOTO 5050
5100 T(3)=KT3(Z): TAT=KTATT(Z)
5110 RETURN
5200 REM SUBROUTINE WHICH CALCULATES THE HEAT BALANCE OF THE BUILDING WITHOUT
        VENTILATION
5210 FOR J = 1 TO 3
5220     Q1(J)=AW(J)*UW*(T(J)-TA)
5230     Q2(J)=AIW(J)*UIW*(T(J)-T(3))
5240     Q2(3)=(AIW(1)*(T(3)-T(1))+AIW(2)*(T(3)-T(2)))*UIW
5250     Q3(J)=AC(J)*UC*(T(J)-TAT)
5260     Q4(J)=UFL*P(J)*(T(J)-TA)
5270     Q5(J)=NA(J)*SHEAT(J)
5280     Q5(1)=Q5(1)+NA(1)*250
5290     Q6(J)=Q5(J)-(Q1(J)+Q2(J)+Q3(J)+Q4(J))
5300 NEXT J
5310 RETURN
5400 REM SUBROUTINE WHICH CALCULATES THE PSYCHROMETRIC PROPERTIES OF THE MODELED
        CONTROL VOLUMES. AMBIENT,FARROWING,NURSERY,HALLWAYS,AND PIPE OUTLET.
5410 FOR J=0 TO 4
5420    T=T(J)+273
5430     IF T<273 THEN GOTO 5630
5440    PS= 89.63-7512/T+.024*T-1.165E-05*T^2-1.281E-08*T^3+2.1E-11*T^4
          -12.15*LOG(T)
5450    PS(J)=EXP(PS)
5460 NEXT J
5470 FOR J=0 TO 2
5480    PW(J)=RH(J)*PS(J)
5490    W(J)=.622*(PW(J)/(PA-PW(J)))
5500 NEXT J
5510 W(3)=W(0): W(4)=W(0)
5520 PW(3)=PW(0): RH(3)=PW(3)/PS(3)
5530 PW(4)=PW(0): RH(4)=PW(4)/PS(4)
5540 FOR J=0 TO  4
5550    HS(J)=1.01*(T(J)-0)
5560    HL(J)=W(J)*(2501+1.78*(T(J)-0))
5570    HT(J)=HS(J)+HL(J)
5580 REM ENTHALPY VALUES VALID FOR -50<T>110 CELSIUS
5590    SVOL(J)=(.287*(T(J)+273))/(PA-PW(J))
5600    DEN(J)=(1+W(J))/SVOL(J)
5610 NEXT J
5620 RETURN
5630 PS=24.28-6238/T-.3444*LOG(T)
5640 GOTO 5450
5800 REM SUBROUTINE WHICH CALCULATES THE VENTILATION RATES FOR THE BUILDING.
        FARROWING,NURSERY, AND TOTAL.
5810 REM CALCULATION OF VENT AS A FUNCTION OF LATENT HEAT
5820 FOR J =1 TO 2
5830    DLAT(J)=HL(J)-HL(3)
5840    VLAT(J)=(LHEAT(J)*NA(J)*3.6)/(DLAT(J)*DEN(3))
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5850    Q7(J) = Q6(J)-((HS(J)-HS(3))*(LHEAT(J)*NA(J)/DLAT(J)))
5860 NEXT J
5870 VLAT=VLAT(1)+VLAT(2)
5880 Q7(3)=0
5890 REM        MASS FLOW OF THE VENTILATION AIR IN kg/s.
5900 MFLOW=(LHEAT(1)*NA(1)/(DLAT(1)*1000))+(LHEAT(2)*NA(2)/(DLAT(2)*1000))
5930 REM        NUMBER OF PIPES REQUIRED TO DELIVER THE VENTILATION NEEDS FOR
                HUMIDITY CONTROL
5940 PIPES=(MFLOW/DEN(4))/AF
5941 IF (PIPES-INT(PIPES)) > .5 THEN PINC = 1 ELSE PINC = 0
5942 PIPES = INT(PIPES) + PINC
5950 REM        THE HEAT LOST BY THE SYSTEM DUE TO THE ADDITION OF
                VENTILATION AIR.
5960 Q8=MFLOW*(HS(4)-HS(3))*1000
5970 REM        INSTANTANEOUS HEAT BALANCE OF THE SYSTEM
5980 DQ=Q7(1)+Q7(2)+Q8
5990 REM        THE CHANGE IN ENERGY WITHIN THE BUFFER ZONE (HALLWAYS).
6000 MASS3=AC(3)*DEN(3)*2.5: REM kg
6010 VKJ=MFLOW*HS(4)*10800: REM kJ
6020 VKJ3=MASS3*HS(3): REM kJ
6030 SMIX=(VKJ+VKJ3)/(MASS3+MFLOW*10800): REM kJ/kg
6040 NEWT3=SMIX/1.01
6050 IF ABS(NEWT3-T(3))<1 THEN RETURN
6060 T(3)=NEWT3
6070 TAT = ((T(1)*AC(1)+T(2)*AC(2)+NEWT3*AC(3))*UC+TA*(AR*UR+AG+UG))/(UC*
          (AC(1)+AC(2)+AC(3))+AR*UR+AG*UG)
6080 RETURN 440
7000 REM SUBROUTINE WHICH FORMATS THE TWO HEADING LINES FOR THE FILE
7010 LSET DTE$=STR$(REC+2): LSET TME$=FILE2$: LSET TEMP$="": LSET OUTTEMP$="":
     LSET DQ$="": LSET PIPE$=""
7020 FOR G=1 TO 4: LSET Q$(G)=CHR$(34):NEXT G
7030 LSET RET$=CHR$(13): LSET LF$=CHR$(10)
7040 FOR V=1 TO 4: LSET DUM$(V)="    ": NEXT V
7050 PUT #2,1
7051 LSET DTE$="   DATE": LSET TME$="  TIME": LSET TEMP$="AMB.TEMP": LSET
     OUTTEMP$="OUT.TEMP": LSET DQ$="  HEAT": LSET PIPE$="# PIPES"
7052 PUT #2,2
7060 LSET DTE$=" ": LSET TME$=" ": LSET TEMP$="  (C)": LSET OUTTEMP$ ="  (C)":
     LSET DQ$="BALANCE":LSET PIPE$=" "
7070 PUT #2,3
7080 RETURN
8000 REM SUBROUTINE WHICH PUTS THE CALCULATED VALUES ONTO DISK
8010 LSET DTE$=DA$
8020 LSET TME$=TI$
8030 LSET TEMP$=STR$(T(0))
8040 LSET OUTTEMP$=STR$(T(4))
8050 LSET DQ$=STR$(DQ)
8060 LSET PIPE$=STR$(PIPES)
8070 FOR G=1 TO 4: LSET Q$(G)=CHR$(34):NEXT G
8080 LSET RET$=CHR$(13): LSET LF$=CHR$(10)
8090 FOR V=1 TO 4: LSET DUM$(V)="    ": NEXT V
8100 PUT #2,I+2
8110 RETURN
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1 REM WINMOD5.BAS                                        WRITTEN: FEBRUARY 21, 1988
2 REM THE PURPOSE OF THE SIMULATION IS TO DETERMINE THE HEAT REQUIREMENT AND
      SAVINGS FOR EACH OF THE SOIL-AIR TEMPERING CONFIGURATIONS.
3 REM THE PROGRAM ALLOWS THE OPERATOR TO SPECIFY THE PIPE #, THE PIPE LENGTH,AND
      THE NUMBER OF PIPES WITHIN THE SYSTEM.  THE SIMULATION WILL ALLOW A COMB-
      INATION OF TEMPERED AND AMBIENT AIR IF THERE IS INSUFFICIENT TEMPERED AIR.
4 REM THIS PROGRAM READS DATA FROM PREPARED DATA FILES D*00*.PRN FROM A HARD
      DISK. IT CREATES NEW FILES TCE*0*.PRN, ALSO STORED ON THE HARD DISK.
5 REM THE PROGRAM REQUIRES NO INPUTS FROM THE USER, BUT THE SUBDIRECTORY
      CONTAINING THE TEMPERATURE DATA MUST BE ACCESSED BEFORE OPERATION OF THE
      PROGRAM.  THE PROGRAM OPERATES ON GWBASIC.
6 REM THE PROGRAM ASSUMES THAT THE HEAT BALANCE BETWEEN THE FARROWING, NURSERY,
      AND HALLS EQUILIBRIATES WITH TIME. THIS HEAT BALANCE WILL INDICATE EITHER
      THE NEED FOR SUPPLEMENTAL HEAT, OR TEMPERATURE CONTROL.
7 REM THE PROGRAM OUTPUT CONSISTS OF AMBIENT AND OUTLET TEMPERATURE, THE HEAT
      BALANCE (W), TOTAL VENTILATION, TEMPERED VENTILATION, AND THE AMBIENT
      VENTILATION.
8 REM THERE IS NO ACCOUNTING FOR TEMPERATURE CONTROL VENTILATION BUILT INTO THIS
      PROGRAM. THE HEAT BALANCE ALONE WOULD INDICATE THE NEED FOR FURTHER
      VENTILATION.
10 REM PROGRAM CONSTANTS
20 REM AREAS
30 AR=682.9: AG=75:AC(0)=648: AC(1)=377.4: AC(2)=151.2: AC(3)=119.4
40 AW(1)=118: AW(2)=105: AW(3)=50: AIW(1)=118: AIW(2)=105: AIW(3)=223
50 REM OUTSIDE FLOOR PERIMETERS
60 P(1)=47.2: P(2)=49.2: P(3)=20
70 REM HEAT TRANSMISSION COEFFICIENTS
80 UC=.286: UW=UC: UR=2.94: UG=4.01: UFL=.714: UIW=1.72
90 REM RELATIVE HUMIDITIES AND INITIAL TEMPERATURES
100 RH(1)=.75: RH(2)=.75: RH(0)=.95
110 T(1)=21: T(2)=27
120 REM SENSIBLE AND LATENT HEAT PRODUCTION OF ANIMALS
130 NA(1)=60: NA(2)=510: NA(3)=0
140 SHEAT(1)=248: SHEAT(2)=41.7: SHEAT(3)=0
150 LHEAT(1)=394: LHEAT(2)=54.3: LHEAT(3)=0
160 REM MSC. VARIABLES
170 PA=101.325: DT3=0
180 DIM KT3(20)
190 DIM KTATT(20)
200 KEY OFF
210 CLS
211 INPUT "INPUT THE PIPE NUMBER (1,3,OR 4)";PN
212 IF PN=1 OR PN=4 THEN AF=.05: REM PIPES 1 & 4 AIRFLOW .05 m^3/s
213 IF PN=3 THEN AF=.1: REM PIPE 3 AIRFLOW .10 m^3/s
214 PRINT
215 INPUT "INPUT THE PIPE LENGTH (20 OR 29 m)";PL
216 PRINT
217 INPUT "INPUT THE NUMBER OF PIPES IN THE SYSTEM";NPIPES
218 CLS
220 INPUT "INPUT STARTING FILE NUMBER";START
230 PRINT
240 INPUT "INPUT LAST FILE NUMBER";LAST
250 CLS
260 FOR X= START TO LAST
265 IF PL=20 GOTO 302
270 IF X<10 THEN FILE2$="TCE"+MID$(STR$(PN),2)+"0"+MID$(STR$(X),2)
280 IF X<10 THEN FILE$="D"+MID$(STR$(PN),2)+"00"+MID$(STR$(X),2): GOTO 310
290 FILE2$="TCE"+MID$(STR$(PN),2)+MID$(STR$(X),2)
300 FILE$="D"+MID$(STR$(PN),2)+"0"+MID$(STR$(X),2)
301 GOTO 310
302 IF X<10 THEN FILE2$="TCF"+MID$(STR$(PN),2)+"0"+MID$(STR$(X),2)
303 IF X<10 THEN FILE$="D"+MID$(STR$(PN),2)+"00"+MID$(STR$(X),2): GOTO 310
304 FILE2$="TCF"+MID$(STR$(PN),2)+MID$(STR$(X),2)
305 FILE$="D"+MID$(STR$(PN),2)+"0"+MID$(STR$(X),2)
310 OPEN "R",#1,"C:"+FILE$+".PRN",88
315 IF PL=20 GOTO 325
320 FIELD #1,1 AS Q$(1),10 AS DA$,1 AS Q$(2),1 AS Q$(3),8 AS TI$,1 AS Q$(4),40
    AS DUMMY$,8 AS SEN$(6),8 AS SEN$(7),8 AS SEN$(8),1 AS RET$,1 AS LF$
321 GOTO 330
325 FIELD #1,1 AS Q$(1),10 AS DA$,1 AS Q$(2),1 AS Q$(3),8 AS TI$,1 AS Q$(4),32
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```
   AS DUMMY$,8 AS SEN$(6),8 AS DUM$,8 AS SEN$(7),8 AS SEN$(8),1 AS RET$,1 AS LF$
330 OPEN "R",#2,"C:"+FILE2$+".PRN",96
340 FIELD #2,1 AS Q$(1),10 AS DTE$,1 AS Q$(2),1 AS Q$(3),8 AS TME$,1 AS Q$(4),
    4 AS DUM$(1),8 AS TEMP$,4 AS DUM$(2),8 AS OUTTEMP$,4 AS DUM$(3),8 AS DQ$
345 FIELD #2,58 AS DUMMY$,4 AS DUM$(4),6 AS VENT$,4 AS DUM$(5),6 AS VT$,4 AS
    DUM$(6),6 AS VA$,1 AS RET$,1 AS LF$
350 GET #1,1
360 REC = VAL(DA$)
370 GOSUB 7000
380 FOR I = 2 TO REC
390 LOCATE 12,6:PRINT "COMPUTER PROGRAM IN PROGRESS"
400 LOCATE 25,10: PRINT "RECORD #";I,"  FILE:";FILE$+".PRN"
410 GET #1,I
420 TA=(VAL(SEN$(7))+VAL(SEN$(8)))/2
430 GOSUB 5000
440 GOSUB 5200
450 T(0)=TA: T(4)=VAL(SEN$(6))
460 GOSUB 5400
470 GOSUB 5800
480 GOSUB 8000
490 NEXT I
500 CLOSE #1: CLOSE #2
510 NEXT X
520 CLOSE
530 CLS
540 LOCATE 12,27: PRINT "COMPUTER PROGRAM FINISHED"
550 END
5000 REM SUBROUTINE WHICH CALCULATES THE TEMPERATURE OF THE HALLWAYS AND THE
        ATTIC. IT IS AN ITERATIVE PROCESS DEPENDING ON CONSECUTIVE VALUES.
5005 IF W=1 GOTO 5120
5010 Y=1
5020 KT3(Y) = ((T(1)*AIW(1)+T(2)*AIW(2))*UIW+TA*AW(3)*UW)/((AIW(1)+AIW(2))*UIW+
             AW(3)*UW)
5030 KTATT(Y) = ((T(1)*AC(1)+T(2)*AC(2)+KT3(Y)*AC(3))*UC+TA*(AR*UR+AG*UG))/(UC*
                (AC(1)+AC(2)+AC(3))+AR*UR+AG*UG)
5040 Z=2
5050 KT3(Z) = ((T(1)*AIW(1)+T(2)*AIW(2))*UIW+TA*AW(3)*UW+KTATT(Z-1)*AC(3)*UC)/((
                 AIW(1)+AIW(2))*UIW+AW(3)*UW+AC(3)*UC)
5060 KTATT(Z) = ((T(1)*AC(1)+T(2)*AC(2)+KT3(Z)*AC(3))*UC+TA*(AR*UR+AG+UG))/(UC*
                (AC(1)+AC(2)+AC(3))+AR*UR+AG*UG)
5070 Y=Z
5080 IF ABS(KT3(Z)-KT3(Z-1))<.01 AND ABS(KTATT(Z)-KTATT(Z-1))<.01 GOTO 5100
5090 Z=Z+1: GOTO 5050
5100 T(3)=KT3(Z): TAT=KTATT(Z)
5110 RETURN
5120 TAT = ((T(1)*AC(1)+T(2)*AC(2)+NEWT3*AC(3))*UC+TA*(AR*UR+AG+UG))/(UC*
          (AC(1)+AC(2)+AC(3))+AR*UR+AG*UG)
5130 T(3)=NEWT3: W=0
5140 GOTO 5110
5200 REM SUBROUTINE WHICH CALCULATES THE HEAT BALANCE OF THE BUILDING WITHOUT
        VENTILATION
5210 FOR J = 1 TO 3
5220     Q1(J)=AW(J)*UW*(T(J)-TA)
5230     Q2(J)=AIW(J)*UIW*(T(J)-T(3))
5240     Q2(3)=(AIW(1)*(T(3)-T(1))+AIW(2)*(T(3)-T(2)))*UIW
5250     Q3(J)=AC(J)*UC*(T(J)-TAT)
5260     Q4(J)=UFL*P(J)*(T(J)-TA)
5270     Q5(J)=NA(J)*SHEAT(J)
5280     Q5(1)=Q5(1)+NA(1)*250
5290     Q6(J)=Q5(J)-(Q1(J)+Q2(J)+Q3(J)+Q4(J))
5300 NEXT J
5310 RETURN
5400 REM SUBROUTINE WHICH CALCULATES THE PSYCHROMETRIC PROPERTIES OF THE MODELED
        CONTROL VOLUMES. AMBIENT,FARROWING,NURSERY,HALLWAYS,AND PIPE OUTLET.
5410 FOR J=0 TO 4
5420    T=T(J)+273
5430     IF T<273 THEN GOTO 5630
5440    PS= 89.63-7512/T+.024*T-1.165E-05*T^2-1.281E-08*T^3+2.1E-11*T^4
          -12.15*LOG(T)
5450    PS(J)=EXP(PS)
5460 NEXT J
5470 FOR J=0 TO 2
```

```
5480    PW(J)=RH(J)*PS(J)
5490    W(J)=.622*(PW(J)/(PA-PW(J)))
5500 NEXT J
5510 W(3)=W(0): W(4)=W(0)
5520 PW(3)=PW(0): RH(3)=PW(3)/PS(3)
5530 PW(4)=PW(0): RH(4)=PW(4)/PS(4)
5540 FOR J=0 TO  4
5550    HS(J)=1.01*(T(J)-0)
5560    HL(J)=W(J)*(2501+1.78*(T(J)-0))
5570    HT(J)=HS(J)+HL(J)
5580 REM ENTHALPY VALUES VALID FOR -50<T>110 CELSIUS
5590    SVOL(J)=(.287*(T(J)+273))/(PA-PW(J))
5600    DEN(J)=(1+W(J))/SVOL(J)
5610 NEXT J
5620 RETURN
5630 PS=24.28-6238/T-.3444*LOG(T)
5640 GOTO 5450
5800 REM SUBROUTINE WHICH CALCULATES THE VENTILATION RATES FOR THE BUILDING.
        FARROWING,NURSERY, AND TOTAL.
5810 REM CALCULATION OF VENT AS A FUNCTION OF LATENT HEAT
5820 MAMB=0: AMBVENT=0
5830 FOR J =1 TO 2
5840    DLAT(J)=HL(J)-HL(3)
5850    VLAT(J)=(LHEAT(J)*NA(J)*3.6)/(DLAT(J)*DEN(3))
5860    Q7(J)=Q6(J)-((HS(J)-HS(3))*(LHEAT(J)*NA(J)/DLAT(J)))
5870 NEXT J
5880 VLAT=VLAT(1)+VLAT(2)
5890 Q7(3)=0
5900 REM         MASS FLOW OF THE VENTILATION AIR IN kg/s.
5910 MFLOW=(LHEAT(1)*NA(1)/(DLAT(1)*1000))+(LHEAT(2)*NA(2)/(DLAT(2)*1000))
5920 MTEMP=MFLOW: VTEMP=(MFLOW/DEN(4))*1000
5930 REM         NUMBER OF PIPES REQUIRED TO DELIVER THE VENTILATION NEEDS FOR
                 HUMIDITY CONTROL
5940 PIPES=(MFLOW/DEN(4))/AF
5950 IF (PIPES-INT(PIPES)) > .5 THEN PINC = 1 ELSE PINC = 0
5960 PIPES = INT(PIPES) + PINC
5970 IF PIPES<NPIPES OR PIPES=NPIPES GOTO 6030
5975    PIPES=NPIPES
5980    MAXTEMP=AF*NPIPES: VTEMP=MAXTEMP*1000
5990    AMBVENT=((MFLOW-(MAXTEMP*DEN(4)))/DEN(0))*1000
6000    MTEMP=MAXTEMP*DEN(4)
6010    MAMB=MFLOW-MTEMP
6020 REM         THE HEAT LOST BY THE SYSTEM DUE TO THE ADDITION OF
                 VENTILATION AIR.
6030 Q8=(MTEMP*(HS(4)-HS(3))+MAMB*(HS(0)-HS(3)))*1000
6040 REM         INSTANTANEOUS HEAT BALANCE OF THE SYSTEM
6050 DQ=Q7(1)+Q7(2)+Q8
6060 REM         THE CHANGE IN ENERGY WITHIN THE BUFFER ZONE (HALLWAYS).
6070 MASS3=AC(3)*DEN(3)*2.5
6080 VKJ=(MTEMP*HS(4)+MAMB*HS(0))*10800
6090 VKJ3=MASS3*HS(3)
6100 SMIX=(VKJ+VKJ3)/((MTEMP+MAMB)*10800+MASS3)
6110 NEWT3=SMIX/1.01
6115 VENT=AMBVENT+VTEMP
6120 IF ABS(NEWT3-T(3)) < 1 THEN RETURN
6130 W=1: GOSUB 5000
6140 RETURN 440
7000 REM SUBROUTINE WHICH FORMATS THE TWO HEADING LINES FOR THE FILE
7010 LSET DTE$=STR$(REC+2): LSET TME$=FILE2$: LSET TEMP$="": LSET OUTTEMP$="":
     LSET DQ$="": LSET VENT$="": LSET VT$="": LSET VA$=""
7020 FOR G=1 TO 4: LSET Q$(G)=CHR$(34):NEXT G
7030 LSET RET$=CHR$(13): LSET LF$=CHR$(10)
7040 FOR V=1 TO 6: LSET DUM$(V)="    ": NEXT V
7050 PUT #2,1
7060 LSET DTE$="   DATE": LSET TME$="  TIME": LSET TEMP$="AMB.TEMP": LSET
     OUTTEMP$="OUT.TEMP": LSET DQ$="  HEAT": LSET VENT$=" VENT"
7065 LSET VT$="  TV": LSET VA$="  AV"
7070 PUT #2,2
7080 LSET DTE$=" ": LSET TME$=" ": LSET TEMP$="  (C)": LSET OUTTEMP$ ="  (C)":
     LSET DQ$="BALANCE":LSET VENT$="(L/s)":LSET VT$="(L/s)":LSET VA$="(L/s)"
7090 PUT #2,3
7100 RETURN
```

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8000 REM SUBROUTINE WHICH PUTS THE CALCULATED VALUES ONTO DISK
8010 LSET DTE$=DA$
8020 LSET TME$=TI$
8030 LSET TEMP$=STR$(T(0))
8040 LSET OUTTEMP$=STR$(T(4))
8050 LSET DQ$=STR$(DQ)
8060 LSET VENT$=STR$(VENT)
8061 LSET VT$=STR$(VTEMP)
8062 LSET VA$=STR$(AMBVENT)
8070 FOR G=1 TO 4: LSET Q$(G)=CHR$(34):NEXT G
8080 LSET RET$=CHR$(13): LSET LF$=CHR$(10)
8090 FOR V=1 TO 6: LSET DUM$(V)="    ": NEXT V
8100 PUT #2,I+2
8110 RETURN
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10 REM SUMMOD1.BAS                                    WRITTEN:NOVEMBER 26, 1987
11 REM SUMMER SIMULATION PROGRAM. THE PROGRAM WAS DESIGNED TO PREDICT THE
       VENTILATION REQUIREMENT AND THE RESULTING ROOM TEMPERATURES FOR THE
       STRUCTURE.  THE CAPACITY OF THE SOIL-AIR TEMPERING SYSTEM WAS TO BE
12 REM DETERMINED BY THE WINTER SIMULATIONS.  BLENDING OF AMBIENT AND TEMPERED
       AIR WILL BE DONE TO RECOMENDED MAXIMUM AIR FLOW RATES.
13 REM THE PROGRAM OUTPUT CONSISTS OF THE AMBIENT, OUTLET AND ROOM TEMPERATURES,
       THE RELATIVE HUMIDITY AND VENTILATION RATES WITHIN EACH ROOM, AND THE
       BUILDING TOTALS.
14 REM THE USER MUST SPECIFY WHICH PIPE AND WHO MANY LATERALS ARE TO BE MODELED.
       THE PROGRAM ONLY MODELS FOR THE 29 m LONG LATERALS.
20 REM THIS PROGRAM READS DATA FROM PREPARED DATA FILES C:MEANSP4.PRN. IT
       CREATES A NEW FILE C:CPIPE4.PRN.
30 REM THE PROGRAM REQUIRES NO INPUTS FROM THE USER. THE PROGRAM RUNS ON GWBASIC
40 REM PROGRAM CONSTANTS
50 REM AREAS
60 AR=682.9: AG=75: AC(0)=648: AC(1)=377.4: AC(2)=151.2: AC(3)=119.4
70 AIW(1)=118: AIW(2)=105: AIW(3)=223
71 DIM AW(4,5)
72 AW(1,1)=92.5: AW(1,2)=   0: AW(1,3)=  0: AW(1,4)=25.5
73 AW(2,1)=   0: AW(2,2)=21  : AW(2,3)=105: AW(2,4)= 9
74 AW(3,1)=15.5: AW(3,2)=28.5: AW(3,3)=  3: AW(3,4)= 3
80 REM OUTSIDE FLOOR PERIMETERS
90 DIM P(3,4)
91 P(1,1)=37!: P(1,2)=   0: P(1,3)= 0  : P(1,4)=10.2
92 P(2,1)=   0: P(2,2)= 3.6: P(2,3)=42!: P(2,4)= 3.6
93 P(3,1)= 6.2: P(3,2)=11.4: P(3,3)= 1.2: P(3,4)= 1.2
100 REM HEAT TRANSFER COEFFICIENTS
110 UC=.286: UW=UC: UR=2.94: UG=4.01: UFL=.714: UIW=1.72
111 REM DECREMENT FACTORS FOR ROOF,GABLES,WALLS,AND FOUNDATION
112 LR=.3: LGS=3.96: LGN=3.97
113 LW(1)=.12: LW(2)=.13: LW(3)=.13: LW(4)=.14
114 LFL(1)=.025: LFL(2)=.039: LFL(3)=.032: LFL(4)=.043
120 REM RELATIVE HUMIDITY AND INITIAL TEMPERATURES
130 RH(1)=.75: RH(2)=.75
135 DIM TE(4,2,5): DIM TR(4,2)
140 TR(1,1)=21: TR(2,1)=27: NT(1)=21: NT(2)=27
150 REM SENSIBLE AND LATENT HEAT PRODUCTION OF ANIMALS
160 NA(1)=60: NA(2)=510: NA(3)=0
170 SHEAT(1)=248: SHEAT(2)=41.7: SHEAT(3)=0
180 LHEAT(1)=394: LHEAT(2)=54.3: LHEAT(3)=0
190 REM MSC. VARIABLES
200 PA=101.325
210 DIM KT3(20): DIM KTATT(20)
230 REM RELATIVE HUMIDITY DATA AS A FUNCTION OF MONTH AND HOUR OF DAY.
240 DIM RELH(5,8)
250 FOR X = 1 TO 5
260     FOR Y = 1 TO 8
270             READ RELH(X,Y)
280     NEXT Y
290 NEXT X
300 REM RELATIVE HUMIDITY DATA FOR THE MONTH OF MAY
310 DATA 72,72,72,60,48,50,51,62
320 REM RELATIVE HUMIDITY DATA FOR THE MONTH OF JUNE
330 DATA 77,76,75,64,52,54,55,66
340 REM RELATIVE HUMIDITY DATA FOR THE MONTH OF JULY
350 DATA 81,80,79,67,54,57,59,70
360 REM RELATIVE HUMIDITY DATA FOR THE MONTH OF AUGUST
370 DATA 81,82,83,68,53,57,60,71
380 REM RELATIVE HUMIDITY DATA FOR THE MONTH OF SEPTEMBER
390 DATA 80,83,85,70,55,61,67,74
400 KEY OFF: CLS
401 DIM D$(15)
402 INPUT "INPUT THE PIPE NUMBER (1,3,OR 4)";PN
403 IF PN=1 OR PN=4 THEN AF=.05: REM PIPES 1 AND 4 AIRFLOW .05 m^3/s
404 IF PN=3 THEN AF=.1: REM PIPE 3 AIRFLOW .10 m^3/s
405 PRINT
406 INPUT "INPUT THE NUMBER OF PIPES";NPIPES
407 CLS
410 OPEN "R",#1,"C:MEANSP"+MID$(STR$(PN),2)+".PRN",136
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420 FIELD #1,1 AS Q1$,10 AS DAT$,1 AS Q2$,1 AS Q3$,8 AS TIM$,1 AS Q4$,8 AS S$(0)
    ,8 AS M$(0)
430 FOR I = 1 TO 6
440     FIELD #1,(22+I*16) AS DUMMY$,8 AS S$(I),8 AS M$(I)
450 NEXT I
460 FIELD #1,134 AS DUM$,1 AS R$,1 AS L$
470 OPEN "R",#2,"C:CPIPE"+MID$(STR$(PN),2)+".PRN",178
480 FIELD #2,1 AS Q$(1),10 AS DTE$,1 AS Q$(2),1 AS Q$(3),8 AS TME$,1 AS Q$(4)
490 FOR K = 1 TO 2
500 FIELD #2,(22+(K-1)*50) AS DUMMY$,4 AS D$(5*K-4),8 AS T$(K),4 AS D$(5*K-3),4
AS R$(K),4 AS D$(5*K-2),6 AS V$(K),4 AS D$(5*K-1),6 AS V2$(K),4 AS D$(5*K),6 AS
V3$(K)
510 NEXT K
520 FIELD #2,122 AS DUM$,4 AS D$(11),6 AS V$(3),4 AS D$(12),8 AS T$(3),4 AS D$(1
3),8 AS T$(4),4 AS D$(14),6 AS V2$(3),4 AS D$(15),6 AS V3$(3),1 AS RET$,1 AS LF$
530 GET #1,1
540 REC=VAL(DAT$): SET=VAL(TIM$)
550 GOSUB 7000
560 I=2: J=1
570 WHILE I < REC
575 CLS
580     LOCATE 12,6:PRINT "COMPUTER PROGRAM IN PROGRESS"
590     WHILE J < SET+1
595             LOCATE 25,25:PRINT "RECORD #";J;" OF";SET;" RECORDS
600             IF J > 1 GOTO 690
610                     GET #1,I
620                     FOR K=0 TO 6: TB(K)=VAL(S$(K)):M(K)=VAL(M$(K)):NEXT K
625                     T(0)=TB(0): T(2)=TB(2): T(4)=TB(4)
630                     GOSUB 4120
650                     I=I+1: J=J+1
660                     GET #1,I
670                     FOR K=0 TO 6:T(K)=VAL(S$(K)):M(K)=VAL(M$(K)):NEXT K
680                     GOTO 720
690             FOR K=0 TO 6:TB(K)=T(K):NEXT K
700             GET #1,I
710             FOR K=0 TO 6:T(K)=VAL(S$(K)):M(K)=VAL(M$(K)):NEXT K
720             MOIS=VAL(MID$(DAT$,1,2))
730             HEURE=VAL(MID$(TIM$,1,2))
740             RH(0)=RELH(MOIS-4,(HEURE+2)/3)/100
750             GOSUB 4000
760             GOSUB 5000
770             GOSUB 5200
780             GOSUB 5400
790             GOSUB 5800
800             GOSUB 6800
810             GOSUB 7100
820             I=I+1: J=J+1
830     WEND
840     GET #1,I
850     SET=VAL(TIM$)
860     GOSUB 7240
870     I=I+1: J=1
880 WEND
890 CLOSE
900 CLS
910 LOCATE 12,27:PRINT "COMPUTER PROGRAM IS FINISHED"
920 END
4000 REM SUBROUTINE WHICH CALCULATES THE EQUIVALENT OUTSIDE TEMPERATURES FOR THE
        VARIOUS EXPOSURES AND DECREMENT FACTORS.
4010 REM THE TEMPERATURES ARE ARRANGED IN A 3-d ARRAY. tHE 1st DIGIT OF THE
        ARGUMENT INDICATES THE LOCATION ON THE BUILDING.
4020 REM 0=ROOF, 1=EXTERIOR WALLS, 2=FLOOR, 3=GABLE ENDS
4030 REM THE 2nd DIGIT REPRESENTS THE TIME FACTOR. 0=PREVIOUS RECORD,
        1=CURRENT RECORD.
4040 REM THE 3rd DIGIT REPRESENTS THE EXPOSURE DIRECTION. 0=ROOF, 1=EAST,
        2=SOUTH, 3=WEST, 4=NORTH.
4050 REM ********************** PREVIOUS TEMPERATURES *************************
4060 TE(0,0,0)=TE(0,1,0): TE(3,0,2)=TE(3,1,2): TE(3,0,4)=TE(3,1,4)
4080 FOR K = 1 TO 4
4090     TE(1,0,K)=TE(1,1,K)
4100     TE(2,0,K)=TE(2,1,K)
4110 NEXT K
```

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4120 REM *********************** CURRENT TEMPERATURES ************************
4130 TE(0,1,0)=M(0)+9.4#*LR*(T(0)-M(0))/UR
4140 TE(3,1,2)=M(2)+9.4#*LGS*(T(2)-M(2))/UG
4150 TE(3,1,4)=M(4)+9.4#*LGN*(T(4)-M(2))/UG
4160 FOR K = 1 TO 4
4170    TE(1,1,K)=M(K)+9.4#*LW(K)*(TB(K)-M(K))/UW
4180    TE(2,1,K)=M(K)+9.4#*LFL(K)*(TB(K)-M(K))/UFL
4190 NEXT K
4200 RETURN
5000 REM SUBROUTINE WHICH CALCULATES THE TEMPERATURE OF THE HALLWAYS AND THE
        ATTIC. IT IS AN ITERATIVE PROCESS DEPENDING ON CONSEQUATIVE VALUES.
5005 IF J > 2 GOTO 5140
5010 Y=1
5020 AMBIENT=(TE(1,0,1)*AW(3,1)+TE(1,0,2)*AW(3,2)+TE(1,0,3)*AW(3,3)+TE(1,0,4)
           *AW(3,4))*UW
5030 DENOM=(AW(3,1)+AW(3,2)+AW(3,3)+AW(3,4))*UW
5040 KT3(Y)=((TR(1,1)*AIW(1)+TR(2,1)*AIW(2))*UIW+AMBIENT)/((AIW(1)+AIW(2))*UIW
           +DENOM)
5050 KTATT(Y)=((TR(1,1)*AC(1)+TR(2,1)*AC(2)+KT3(Y)*AC(3))*UC+TE(0,1,0)*AR*UR
            +(TE(3,1,2)+TE(3,1,4))*AG*UG)/(UC*(AC(1)+AC(2)+AC(3))+AR*UR+AG*UG)
5060 Z=2
5070 KT3(Z)=((TR(1,1)*AIW(1)+TR(2,1)*AIW(2))*UIW+AMBIENT+KTATT(Z-1)*AC(3)*UC)/
          ((AIW(1)+AIW(2))*UIW+DENOM+AC(3)*UC)
5080 KTATT(Z)=((TR(1,1)*AC(1)+TR(2,1)*AC(2)+KT3(Z)*AC(3))*UC+TE(0,1,0)*AR*UR+
            (TE(3,1,2)+TE(3,1,4))*AG*UG)/(UC*(AC(1)+AC(2)+AC(3))+AR*UR+AG*UG)
5090 Y=Z
5100 IF ABS(KT3(Z)-KT3(Z-1)) < .01 AND ABS(KTATT(Z)-KTATT(Z-1)) < .01 GOTO 5120
5110 Z=Z+1: GOTO 5070
5120  TR(3,1)=KT3(Z): TATP=KTATT(Z): TAT=KTATT(Z)
5130 RETURN
5140 TATP=TAT
5150 TAT=((TR(1,1)*AC(1)+TR(2,1)*AC(2)+TR(3,1)*AC(3))*UC+TE(0,1,0)*AR*UR+
        (TE(3,1,2)+TE(3,1,4))*AG*UG)/(UC*(AC(1)+AC(2)+AC(3))+AR*UR+AG*UG)
5160 GOTO 5130
5200 REM SUBROUTINE WHICH CALCULATES THE HEAT BALANCE OF THE BUILDING WITHOUT
        VENTILATION
5205 FOR H=1 TO 3:Q1(H)=0:Q4(H)=0:NEXT H
5210 FOR H= 1 TO 3
5220    FOR K = 1 TO 4
5230    QT1(H,K)=AW(H,K)*UW*(TR(H,1)-TE(1,0,K))
5240            Q1(H)=Q1(H)+QT1(H,K)
5250            QT4(H,K)=UFL*P(H,K)*(TR(H,1)-TE(2,0,K))
5260            Q4(H)=Q4(H)+QT4(H,K)
5270    NEXT K
5280    Q2(H)=AIW(H)*UIW*(TR(H,1)-TR(3,1))
5290    Q2(3)=(AIW(1)*(TR(3,1)-TR(1,1))+AIW(2)*(TR(3,1)-TR(2,1)))*UIW
5300    Q3(H)=AC(H)*UC*(TR(H,1)-TATP)
5310    Q5(H)=NA(H)*SHEAT(H)
5320    Q5(1)=NA(1)*(250+SHEAT(1))
5330    Q6(H)=Q5(H)-(Q1(H)+Q2(H)+Q3(H)+Q4(H))
5340 NEXT H
5350 RETURN
5400 REM SUBROUTINE WHICH CALCULATES THE PSYCHROMETRIC PROPERTIES OF THE MODELED
        CONTROL VOLUMES. AMBIENT,FARROWING,NURSERY,HALLWAYS,AND PIPE OUTLET.
5410 TR(0,1)=T(5): TR(0,0)=TB(5)
5420 TR(4,1)=T(6): TR(4,0)=TB(6)
5430 FOR H = 0 TO 4
5440    T=TR(H,1)+273
5450    IF T < 273 GOTO 5730
5460    PS=89.63-7512/T+.024*T-1.165E-05*T^2-1.281E-08*T^3+2.1E-11*T^4
          -12.15*LOG(T)
5470    PS(H)=EXP(PS)
5480 NEXT H
5490 REM CALCULATION OF ABSOLUTE HUMIDITY AND VAPOUR PRESSURE FOR AMBIENT,
        FARROWING,AND NURSERY AIR.
5500 FOR H = 0 TO 2
5510    PW(H)=RH(H)*PS(H)
5520    W(H)=.622*(PW(H)/(PA-PW(H)))
5530 NEXT H
5540 REM ABSOLUTE HUMIDITY,VAPOUR PRESSURE,AND REL. HUMIDITY OF THE HALL AND
        OUTLET AIR.
5550 W(3)=W(0): W(4)=W(0)
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5560 PW(3)=PW(0): RH(3)=PW(3)/PS(3)
5570 PW(4)=PW(0): RH(4)=PW(4)/PS(4)
5580 REM CALCULATION OF OUTLET ABSOLUTE HUMIDITY IF RELATIVE HUMIDITY EXCEEDS
      100% (SATURATION WILL RESULT IN MOISTURE LOSS BY CONDENSATION).
5590 IF RH(4) < 1 OR RH(4) = 1 GOTO 5630
5600    PW(4)=PS(4)
5610    W(4)=.622*(PW(4)/(PA-PW(4))): RH(4)=1!
5620 REM CALCULATION OF SENSIBLE,LATENT,AND TOTAL ENTHALPIES.
5630 REM ENTHALPY VALUES VALID FOR -50 C < T > 110 C.
5640 FOR H = 0 TO 4
5650    HS(H)=1.01*(TR(H,1)-0)
5660    HL(H)=W(H)*(2501+1.78*(TR(H,1)-0))
5670    HT(H)=HS(H)+HL(H)
5680    REM CALCULATION OF SPECIFIC VOLUMES, AND DENSITIES.
5690    SVOL(H)=(.287*(TR(H,1)+273))/(PA-PW(H))
5700    DEN(H)=(1+W(H))/SVOL(H)
5710 NEXT H
5711 FOR K = 1 TO 2
5712    SH(K)=1.01*(NT(K)-0)
5713    Q6(K)=Q6(K)+2500*(SH(K)-HS(K))*AC(K)*DEN(K)/10800: REM DIFFERENCE IN
     SENSIBLE HEAT BETWEEN ROOM TEMP AND DESIRED TEMP,ADDED TO ROOM HEAT LOAD.
5714 NEXT K
5720 RETURN
5730 PS=24.28-6238/T-.3444*LOG(T)
5740 GOTO 5470
5800 REM SUBROUTINE WHICH CALCULATES THE VENTILATION RATES FOR THE BUILDING,
      FAROWING,NURSERY,ANDTOTALS.
5805 VAMB=0: VAMB(1)=0: VAMB(2)=0: VTEMP=0: VTEMP(1)=0: VTEMP(2)=0: X=2
5810 IF TR(0,1) > 19 OR TR(0,1) = 19 THEN L=4:REM SET FOR AIR TEMPERING
5820 IF TR(0,1) < 19 THEN L=0: REM SET FOR AMBIENT VENTILATION
5830 FOR K = 1 TO 2
5840    DSEN(K)=HS(K)-HS(L):REM SENSIBLE HEAT POTENTIAL kJ/kg
5850    MASS(K)=Q6(K)/(DSEN(K)*1000): REM MASS FLOW RATE kg/s
5860    VSEN(K)=MASS(K)*SVOL(L)*1000: REM ROOM VENT. RATE L/s
5870 NEXT K
5880 VSEN=VSEN(1)+VSEN(2): REM TOTAL VENTILATION L/s
5890 MASS=MASS(1)+MASS(2): REM TOTAL MASS FLOW kg/s
5900 MASS3=AC(3)*2.5*DEN(3): REM AIR MASS OF HALLWAY
5910 VKJ=MASS*HS(L)*10800: REM SENSIBLE CONTENT OF VENT. AIR kJ
5920 VKJ3=MASS3*HS(3): REM SENSIBLE CONTENT OF HALLWAY kJ
5930 SMIX=(VKJ+VKJ3)/(MASS*10800+MASS3): REM SENSIBLE CONTENT OF MIX kJ/kg
5940 TMIX=(MASS3*HT(3)+MASS*HT(L)*10800)/(MASS3+MASS*10800):REM TOTAL ENTHALPY
     OF MIX kJ/kg.
5950 LMIX=TMIX-SMIX: REM LATENT CONTENT OF AIR MIXTURE kJ/kg.
5960 NEWT3=SMIX/1.01: REM NEW HALL TEMPERATURE (C).
5970 NEWW=(MASS3*W(3)+MASS*10800*W(L))/(MASS3+MASS*10800): REM MIXTURE ABS.
     HUMIDITY kg/kg.
5980 PW=PA*NEWW/1.622: REM DERIVED FROM LINE 5520
5990 T=NEWT3+273: REM HALL TEMPERATURE (K)
6000 PS=89.63-7512/T+.024*T-1.165E-05*T^2-1.281E-08*T^3+2.1E-11*T^4
        -12.15*LOG(T)
6010 PS=EXP(PS)
6020 NEWRH=PW/PS: REM REL. HUMIDITY OF AIR MIXTURE (%).
6030 NEWVOL=(.287*T)/(PA-PW): REM SPEC. VOLUME OF AIR MIXTURE m^3/kg.
6040 NEWVENT=0
6050 FOR K = 1 TO 2
6060    NEWVENT(K)=(Q6(K)/((HS(K)-SMIX)*1000))*1000*NEWVOL: REM L/s
6070    NEWVENT=NEWVENT+NEWVENT(K)
6080 NEXT K
6090 IF NEWVENT/VSEN > .95 AND NEWVENT/VSEN < 1.05 GOTO 6150
6100 FOR K = 1 TO 2
6110    MASS(K)=Q6(K)/((HS(K)-SMIX)*1000): REM kg/s.
6120    VSEN(K)=MASS(K)*NEWVOL*1000: REM L/s
6130 NEXT K
6140 GOTO 5880
6150 MAXVENT(1)=(NA(1)*110): MAXVENT(2)=(NA(2)*15): REM MAX TOTAL VENTILATION
     14250 L/s.
6155 VSEN(1)=NEWVENT(1): VSEN(2)=NEWVENT(2): VSEN=VSEN(1)+VSEN(2)
6160 IF VSEN(1) > MAXVENT(1) AND L=0 OR VSEN(2) > MAXVENT(2) AND L=0 THEN L=4:
     GOTO 5830
6170 MAXTEMP=AF*NPIPES*1000: REM MAX AIR TEMPERED FLOW L/s
6180 IF VSEN > MAXTEMP AND L=4 THEN GOSUB 6340
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6185 IF L=0 THEN VAMB(1)=VSEN(1): VAMB(2)=VSEN(2): VAMB=VSEN
6186 IF L=4 AND X=2 THEN VTEMP(1)=VSEN(1): VTEMP(2)=VSEN(2): VTEMP=VSEN
6190 FOR K = 1 TO 2
6200    DS(K)=Q6(K)/1000-(HS(K)-SMIX)*MASS(K): REM kW
6210    DL(K)=(LHEAT(K)*NA(K))/1000-(HL(K)-LMIX)*MASS(K):REM LATENT POTENTIAL kW
6220    SENS(K)=HS(K)+DS(K)/MASS(K): REM NEW SENSIBLE CONTENT OF ROOM kJ/kg
6230    LAT(K)=HL(K)+DL(K)/MASS(K): REM NEW LATENT CONTENT OF ROOM kJ/kg>
6240    NT(K)=SENS(K)/1.01: REM NEW ROOM TEMPERATURE (C).
6250    RMASS(K)=AC(K)*2.5*DEN(K): REM ROOM AIR MASS kg.
6260    NEWW(K)=(RMASS(K)*W(K)+NEWW*MASS(K)*10800)/(RMASS+MASS(K)*10800):
        REM NEW ROOM ABSOLUTE HUMIDITY kg/kg
6270    PW(K)=PA*NEWW(K)/1.622: REM ROOM VAPOUR PRESSURE kPa.
6280    T=NT(K)+273
6290    PS=89.63-7512/T+.024*T-1.165E-05*T^2-1.281E-08*T^3+2.1E-11*T^4
           -12.15*LOG(T)
6300    PS(K)=EXP(PS)
6310    NEWRH(K)=PW/PS: REM NEW ROOM RELATIVE HUMIDITY (%).
6320 NEXT K
6330 RETURN
6340 REM SUBROUTINE WHICH HANDLES THE BLENDING OF AMBIENT AND TEMPERED AIR
6350 REM THIS ASSUMES THAT THERE IS INSUFFICIENT TEMPERED AIR TO CONTROL ROOM
         TEMPERATURES, AND AMBIENT TEMPERATURE IS ABOVE THE 19 C SET POINT.
6360 REM A PROBLEM ARISES SINCE THE TWO ROOMS HAVE DIFFERENT OPTIMUM TEMPERA-
         TURES. THIS MEANS THAT THOUGH THE SET POINT FOR THE FARROWING ROOM HAS
6370 REM BEEN EXCEEDED, TEMPERATURE CONTROL OF THE NURSERY MAY STILL BE POSSIBLE
         USING AMBIENT AIR. ONCE THE SET POINT OF THE NURSERY HAS BEEN EXCEEDED
6380 REM THE MAXIMUM VENTILATION WILL BE USED RERGARDLESS OF THE AMBIENT TEMP-
         ERATURE.
6390 VENT1=0: X=0
6395 VTEMP(1)=MAXTEMP*.4632: VTEMP(2)=MAXTEMP-VTEMP(1): VAMB(2)=0:
     VAMB(1)=MAXVENT(1)-VTEMP(1)
6400 VAMB=VAMB(1)+VAMB(2): REM AMBIENT VENTILATION RATE L/s.
6410 VTEMP=MAXVENT: REM TEMPERED AIRFLOW RATE L/s.
6420 MAMB=VAMB/1000*DEN(0): REM AMB. MASS FLOW kg/s.
6430 MTEMP=VTEMP/1000*DEN(4): REM TEMPERED MASS FLOW kg/s.
6440 SMIX=((MAMB*HS(0)+MTEMP*HS(4))*10800+MASS3*HS(3))/((MAMB+MTEMP)*10800
          +MASS3): REM kJ/kg
6450 LMIX=((MAMB*HL(0)+MTEMP*HL(4))*10800+MASS3*HL(3))/((MAMB+MTEMP)*10800
          +MASS3): REM kJ/kg
6460 NEWT3=SMIX/1.01:
6470 NEWW=((MAMB*W(0)+MTEMP*W(4))*10800+MASS3*W(3))/((MAMB+MTEMP)*10800+MASS3)
6480 PW=PA*NEWW/1.622
6490 T=NEWT3+273
6500 PS=89.63-7512/T+.024*T-1.165E-05*T^2-1.281E-08*T^3+2.1E-11*T^4
     -12.15*LOG(T)
6510 PS=EXP(PS)
6520 NEWRH=PW/PS
6530 NEWVOL=(.287*T)/(PA-PW)
6540 NEWDEN=(1+NEWW)/NEWVOL
6549 IF X=1 GOTO 6620
6550 DS(2)=HS(2)-SMIX
6551 IF DS(2)<0 OR DS(2)=0 GOTO 6670
6560 MASS(2)=Q6(2)/(DS(2)*1000): REM kg/s
6570 VSEN(2)=MASS(2)*NEWVOL*1000: REM L/s
6580 VAMB(2)=VSEN(2)-VTEMP(2): REM NEW AMB. VENTILATION RATE L/s
6585 IF VAMB(2) > (MAXVENT(2)-VTEMP(2)) GOTO 6670
6590 IF VENT1/VSEN(2) > .95 AND VENT1/VSEN(2) < 1.05 GOTO 6620
6600 VENT1=VSEN(2)
6610 GOTO 6400
6620 MASS(1)=6.6*NEWDEN
6630 FOR K = 1 TO 2
6640    VSEN(K)=VTEMP(K)+VAMB(K)
6650 NEXT K
6651 VSEN=VSEN(1)+VSEN(2)
6655 IF X=1 THEN MASS(2)=VSEN(2)/(1000*NEWVOL)
6660 RETURN
6670 VAMB(2)=MAXVENT(2)-VTEMP(2): VSEN(2)=MAXVENT(2): X=1
6680 GOTO 6400
6800 REM SUBROUTINE WHICH RE-ASSIGNS VARIABLE VALUES
6810 FOR K = 1 TO 2
6840    RH(K)=NEWRH(K)
6850 NEXT K
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6860 TR(3,0)=TR(3,1)
6870 TR(3,1)=NEWT3
6880 RETURN
7000 REM SUBROUTINE WHICH FORMATS THE TWO HEADING LINES FOR THE FILE
7010 LSET DTE$=STR$(REC): LSET TME$=STR$(SET-1): LSET T$(1)="FARROW":LSET T$(2)
     ="NURSERY": LSET V$(3)="TOTAL": LSET T$(3)="AMBIENT": LSET T$(4)="OUTLET"
7011 LSET V2$(1)="       ": LSET V3$(1)="       ":LSET V2$(2)="       ":LSET V3$(2)
     ="       ":LSET V2$(3)="TOTAL":LSET V3$(3)="TOTAL"
7020 FOR G = 1 TO 4: LSET Q$(G)=CHR$(34): NEXT G
7025 FOR G = 1 TO 2: LSET R$(G)="    ": LSET V$(G)="       ":NEXT G
7030 LSET RET$=CHR$(13): LSET LF$=CHR$(10)
7040 FOR V = 1 TO 15: LSET D$(V)="    ": NEXT V
7050 PUT #2,1
7060 LSET DTE$="   DATE":LSET TME$="  TIME":LSET T$(1)="  TEMP":LSET R$(1)=" RH"
     :LSET V$(1)=" VENT":LSET T$(2)="  TEMP":LSET R$(2)=" RH":LSET V$(2)=" VENT"
7065 LSET V2$(1)=" AMB":LSET V3$(1)=" TEMP":LSET V2$(2)=" AMB": LSET V3$(2)
     =" TEMP":LSET V2$(3)=" AMB":LSET V3$(3)=" TEMP"
7070 LSET V$(3)=" VENT":LSET T$(3)="  TEMP":LSET T$(4)="  TEMP"
7071 FOR G = 1 TO 4: LSET Q$(G)=CHR$(34): NEXT G
7072 FOR V = 1 TO 15: LSET D$(V)="    ": NEXT V
7073 LSET RET$=CHR$(13): LSET LF$=CHR$(10)
7080 PUT #2,2
7090 RETURN
7100 REM SUBROUTINE WHICH PUTS THE CALCULATED VALUES ONTO DISK
7110 LSET DTE$=DAT$
7120 LSET TME$=TIM$
7130 FOR K = 1 TO 2
7140    LSET T$(K)=STR$(NT(K))
7150    LSET R$(K)=STR$(RH(K))
7160    LSET V$(K)=STR$(VSEN(K))
7161    LSET V2$(K)=STR$(VAMB(K))
7162    LSET V3$(K)=STR$(VTEMP(K))
7170 NEXT K
7180 LSET V$(3)=STR$(VSEN): LSET T$(3)=STR$(T(5)): LSET T$(4)=STR$(T(6))
7185 LSET V2$(3)=STR$(VAMB): LSET V3$(3)=STR$(VTEMP)
7190 FOR G = 1 TO 4: LSET Q$(G)=CHR$(34): NEXT G
7200 FOR V = 1 TO 15: LSET D$(V)="    ": NEXT V
7210 LSET RET$=CHR$(13): LSET LF$=CHR$(10)
7220 PUT #2,I
7230 RETURN
7240 REM SUBROUTINE WHICH FORMATS THE HEADING LINE FOR MISSING DATA
7250 LSET DTE$="MISSING   ":LSET TME$=STR$(SET-1)
7260 FOR G = 1 TO 4: LSET Q$(G)=CHR$(34): NEXT G
7270 FOR K = 1 TO 2
7280    LSET T$(K)="        "
7290    LSET R$(K)="    "
7300    LSET V$(K)="      "
7301    LSET V2$(K)="      "
7302    LSET V3$(K)="      "
7310 NEXT K
7320 LSET V$(3)="      ": LSET T$(3)="        ": LSET T$(4)="        "
7321 LSET V2$(3)="      ": LSET V3$(3)="      "
7330 FOR V = 1 TO 15: LSET D$(V)="    ": NEXT V
7340 LSET RET$=CHR$(13): LSET LF$=CHR$(10)
7350 PUT #2,I
7360 RETURN
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10 REM SUMVENT.BAS                                         WRITTEN:FEBRUARY  2, 1987
11 REM THIS PROGRAM SIMULATES THE SUMMER OPERATION OF A CONVENTIONAL VENTILATION
       SYSTEM.  THE PURPOSE OF THE PROGRAM IS TO ESTABLISH THE LEVEL OF
       TEMPERATURE CONTROL POSSIBLE WITHIN THE STRUCTURE.
12 REM THE RESULTS OF THIS PROGRAM WILL THEN BE COMPARED AGAINST THE SOIL-AIR
       TEMPERING RESULTS TO DETERMINE IF THERE ARE ANY SIGNIFICANT DIFFERENCES.
13 REM THE PROGRAM OUTPUT CONSISTS OF THE TEMPERATURE, RELATIVE HUMIDITY, AND
       VENTILATION RATE OF THE ROOMS, THE TOTAL VENTILATION RATE, AND THE
       AMBIENT TEMPERATURE.
20 REM REVISION OF SUMMER PROGRAM SUMMOD1.BAS. THIS PROGRAM READS DATA FROM
       PREPARED DATA FILES C:MEANSP4.PRN. IT CREATES A NEW FILE C:CVENT.PRN.
30 REM THE PROGRAM REQUIRES NO INPUTS FROM THE USER. THE PROGRAM RUNS ON GWBASIC
40 REM PROGRAM CONSTANTS
50 REM AREAS
60 AR=682.9: AG=75: AC(0)=648: AC(1)=377.4: AC(2)=151.2: AC(3)=119.4
70 AIW(1)=118: AIW(2)=105: AIW(3)=223
71 DIM AW(4,5)
72 AW(1,1)=92.5: AW(1,2)=   0: AW(1,3)=  0: AW(1,4)=25.5
73 AW(2,1)=   0: AW(2,2)=21  : AW(2,3)=105: AW(2,4)= 9
74 AW(3,1)=15.5: AW(3,2)=28.5: AW(3,3)=  3: AW(3,4)= 3
80 REM OUTSIDE FLOOR PERIMETERS
90 DIM P(3,4)
91 P(1,1)=37!: P(1,2)=   0: P(1,3)= 0  : P(1,4)=10.2
92 P(2,1)=   0: P(2,2)= 3.6: P(2,3)=42!: P(2,4)= 3.6
93 P(3,1)= 6.2: P(3,2)=11.4: P(3,3)= 1.2: P(3,4)= 1.2
100 REM HEAT TRANSFER COEFFICIENTS
110 UC=.286: UW=UC: UR=2.94: UG=4.01: UFL=.714: UIW=1.72
111 REM DECREMENT FACTORS FOR ROOF,GABLES,WALLS,AND FOUNDATION
112 LR=.3: LGS=3.96: LGN=3.97
113 LW(1)=.12: LW(2)=.13: LW(3)=.13: LW(4)=.14
114 LFL(1)=.025: LFL(2)=.039: LFL(3)=.032: LFL(4)=.043
120 REM RELATIVE HUMIDITY AND INITIAL TEMPERATURES
130 RH(1)=.75: RH(2)=.75
135 DIM TE(4,2,5): DIM TR(4,2)
140 TR(1,1)=21: TR(2,1)=27: NT(1)=21: NT(2)=27
150 REM SENSIBLE AND LATENT HEAT PRODUCTION OF ANIMALS
160 NA(1)=60: NA(2)=510: NA(3)=0
170 SHEAT(1)=248: SHEAT(2)=41.7: SHEAT(3)=0
180 LHEAT(1)=394: LHEAT(2)=54.3: LHEAT(3)=0
190 REM MSC. VARIABLES
200 PA=101.325
210 DIM KT3(20): DIM KTATT(20)
230 REM RELATIVE HUMIDITY DATA AS A FUNCTION OF MONTH AND HOUR OF DAY.
240 DIM RELH(5,8)
250 FOR X = 1 TO 5
260     FOR Y = 1 TO 8
270             READ RELH(X,Y)
280     NEXT Y
290 NEXT X
300 REM RELATIVE HUMIDITY DATA FOR THE MONTH OF MAY
310 DATA 72,72,72,60,48,50,51,62
320 REM RELATIVE HUMIDITY DATA FOR THE MONTH OF JUNE
330 DATA 77,76,75,64,52,54,55,66
340 REM RELATIVE HUMIDITY DATA FOR THE MONTH OF JULY
350 DATA 81,80,79,67,54,57,59,70
360 REM RELATIVE HUMIDITY DATA FOR THE MONTH OF AUGUST
370 DATA 81,82,83,68,53,57,60,71
380 REM RELATIVE HUMIDITY DATA FOR THE MONTH OF SEPTEMBER
390 DATA 80,83,85,70,55,61,67,74
400 KEY OFF: CLS
410 OPEN "R",#1,"C:MEANSP4.PRN",136
420 FIELD #1,1 AS Q1$,10 AS DAT$,1 AS Q2$,1 AS Q3$,8 AS TIM$,1 AS Q4$,8 AS S$(0)
    ,8 AS M$(0)
430 FOR I = 1 TO 6
440     FIELD #1,(22+I*16) AS DUMMY$,8 AS S$(I),8 AS M$(I)
450 NEXT I
460 FIELD #1,134 AS DUM$,1 AS R$,1 AS L$
470 OPEN "R",#2,"C:CVENT.PRN",106
480 FIELD #2,1 AS Q$(1),10 AS DTE$,1 AS Q$(2),1 AS Q$(3),8 AS THE$,1 AS Q$(4)
490 FOR K = 1 TO 2
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500 FIELD #2,(22+(K-1)*29) AS DUMMY$,4 AS D$(3*K-2),8 AS T$(K),4 AS D$(3*K-1),
    4 AS R$(K),4 AS D$(3*K),6 AS V$(K)
510 NEXT K
520 FIELD #2, 80 AS DUM$,4 AS D$(7),6 AS V$(3),4 AS D$(8),8 AS T$(3),1 AS RET$,
    1 AS LF$
530 GET #1,1
540 REC=VAL(DAT$): SET=VAL(TIM$)
550 GOSUB 7000
560 I=2: J=1
570 WHILE I < REC
575 CLS
580     LOCATE 12,6:PRINT "COMPUTER PROGRAM IN PROGRESS"
590     WHILE J < SET+1
595             LOCATE 25,25:PRINT "RECORD #";J;" OF";SET;" RECORDS
600             IF J > 1 GOTO 690
610                     GET #1,I
620                     FOR K=0 TO 5: TB(K)=VAL(S$(K)):M(K)=VAL(M$(K)):NEXT K
625                     T(0)=TB(0): T(2)=TB(2): T(4)=TB(4)
630                     GOSUB 4120
650                     I=I+1: J=J+1
660                     GET #1,I
670                     FOR K=0 TO 5:T(K)=VAL(S$(K)):M(K)=VAL(M$(K)):NEXT K
680                     GOTO 720
690             FOR K=0 TO 5:TB(K)=T(K):NEXT K
700             GET #1,I
710             FOR K=0 TO 5:T(K)=VAL(S$(K)):M(K)=VAL(M$(K)):NEXT K
720             MOIS=VAL(MID$(DAT$,1,2))
730             HEURE=VAL(MID$(TIM$,1,2))
740             RH(0)=RELH(MOIS-4,(HEURE+2)/3)/100
750             GOSUB 4000
760             GOSUB 5000
770             GOSUB 5200
780             GOSUB 5400
790             GOSUB 5800
810             GOSUB 7140
820             I=I+1: J=J+1
830     WEND
840     GET #1,I
850     SET=VAL(TIM$)
860     GOSUB 7280
870     I=I+1: J=1
880 WEND
890 CLOSE
900 CLS
910 LOCATE 12,27:PRINT "COMPUTER PROGRAM IS FINISHED"
920 END
4000 REM SUBROUTINE WHICH CALCULATES THE EQUIVALENT OUTSIDE TEMPERATURES FOR THE
         VARIOUS EXPOSURES AND DECREMENT FACTORS.
4010 REM THE TEMPERATURES ARE ARRANGED IN A 3-d ARRAY. tHE 1st DIGIT OF THE
         ARGUMENT INDICATES THE LOCATION ON THE BUILDING.
4020 REM 0=ROOF, 1=EXTERIOR WALLS, 2=FLOOR, 3=GABLE ENDS
4030 REM THE 2nd DIGIT REPRESENTS THE TIME FACTOR. 0=PREVIOUS RECORD,
         1=CURRENT RECORD.
4040 REM THE 3rd DIGIT REPRESENTS THE EXPOSURE DIRECTION. 0=ROOF, 1=EAST,
         2=SOUTH, 3=WEST, 4=NORTH.
4050 REM ********************** PREVIOUS TEMPERATURES *************************
4060 TE(0,0,0)=TE(0,1,0): TE(3,0,2)=TE(3,1,2): TE(3,0,4)=TE(3,1,4)
4080 FOR K = 1 TO 4
4090    TE(1,0,K)=TE(1,1,K)
4100    TE(2,0,K)=TE(2,1,K)
4110 NEXT K
4120 REM *********************** CURRENT TEMPERATURES *************************
4130 TE(0,1,0)=M(0)+9.4#*LR*(T(0)-M(0))/UR
4140 TE(3,1,2)=M(2)+9.4#*LGS*(T(2)-M(2))/UG
4150 TE(3,1,4)=M(4)+9.4#*LGN*(T(4)-M(2))/UG
4160 FOR K = 1 TO 4
4170    TE(1,1,K)=M(K)+9.4#*LW(K)*(TB(K)-M(K))/UW
4180    TE(2,1,K)=M(K)+9.4#*LFL(K)*(TB(K)-M(K))/UFL
4190 NEXT K
4200 RETURN
5000 REM SUBROUTINE WHICH CALCULATES THE TEMPERATURE OF THE HALLWAYS AND THE
         ATTIC. IT IS AN ITERATIVE PROCESS DEPENDING ON CONSEQUATIVE VALUES.
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5005 IF J > 2 GOTO 5140
5010 Y=1
5020 AMBIENT=(TE(1,0,1)*AW(3,1)+TE(1,0,2)*AW(3,2)+TE(1,0,3)*AW(3,3)+TE(1,0,4)
          *AW(3,4))*UW
5030 DENOM=(AW(3,1)+AW(3,2)+AW(3,3)+AW(3,4))*UW
5040 KT3(Y)=((TR(1,1)*AIW(1)+TR(2,1)*AIW(2))*UIW+AMBIENT)/((AIW(1)+AIW(2))*UIW
          +DENOM)
5050 KTATT(Y)=((TR(1,1)*AC(1)+TR(2,1)*AC(2)+KT3(Y)*AC(3))*UC+TE(0,1,0)*AR*UR
          +(TE(3,1,2)+TE(3,1,4))*AG*UG)/(UC*(AC(1)+AC(2)+AC(3))+AR*UR+AG*UG)
5060 Z=2
5070 KT3(Z)=((TR(1,1)*AIW(1)+TR(2,1)*AIW(2))*UIW+AMBIENT+KTATT(Z-1)*AC(3)*UC)/
        ((AIW(1)+AIW(2))*UIW+DENOM+AC(3)*UC)
5080 KTATT(Z)=((TR(1,1)*AC(1)+TR(2,1)*AC(2)+KT3(Z)*AC(3))*UC+TE(0,1,0)*AR*UR+
          (TE(3,1,2)+TE(3,1,4))*AG*UG)/(UC*(AC(1)+AC(2)+AC(3))+AR*UR+AG*UG)
5090 Y=Z
5100 IF ABS(KT3(Z)-KT3(Z-1)) < .01 AND ABS(KTATT(Z)-KTATT(Z-1)) < .01 GOTO 5120
5110 Z=Z+1: GOTO 5070
5120  TR(3,1)=KT3(Z): TATP=KTATT(Z): TAT=KTATT(Z)
5130 RETURN
5140 TATP=TAT
5150 TAT=((TR(1,1)*AC(1)+TR(2,1)*AC(2)+TR(3,1)*AC(3))*UC+TE(0,1,0)*AR*UR+
       (TE(3,1,2)+TE(3,1,4))*AG*UG)/(UC*(AC(1)+AC(2)+AC(3))+AR*UR+AG*UG)
5160 GOTO 5130
5200 REM SUBROUTINE WHICH CALCULATES THE HEAT BALANCE OF THE BUILDING WITHOUT
       VENTILATION
5205 FOR H=1 TO 3:Q1(H)=0:Q4(H)=0:NEXT H
5210 FOR H= 1 TO 3
5220    FOR K = 1 TO 4
5230    QT1(H,K)=AW(H,K)*UW*(TR(H,1)-TE(1,0,K))
5240            Q1(H)=Q1(H)+QT1(H,K)
5250            QT4(H,K)=UFL*P(H,K)*(TR(H,1)-TE(2,0,K))
5260            Q4(H)=Q4(H)+QT4(H,K)
5270    NEXT K
5280    Q2(H)=AIW(H)*UIW*(TR(H,1)-TR(3,1))
5290    Q2(3)=(AIW(1)*(TR(3,1)-TR(1,1))+AIW(2)*(TR(3,1)-TR(2,1)))*UIW
5300    Q3(H)=AC(H)*UC*(TR(H,1)-TATP)
5310    Q5(H)=NA(H)*SHEAT(H)
5320    Q5(1)=NA(1)*(250+SHEAT(1))
5330    Q6(H)=Q5(H)-(Q1(H)+Q2(H)+Q3(H)+Q4(H))
5340 NEXT H
5350 RETURN
5400 REM SUBROUTINE WHICH CALCULATES THE PSYCHROMETRIC PROPERTIES OF THE MODELED
       CONTROL VOLUMES. AMBIENT,FARROWING,NURSERY,HALLWAYS,AND PIPE OUTLET.
5410 TR(0,1)=T(5): TR(0,0)=TB(5)
5430 FOR H = 0 TO 3
5440    T=TR(H,1)+273
5450    IF T < 273 GOTO 5730
5460    PS=89.63-7512/T+.024*T-1.165E-05*T^2-1.281E-08*T^3+2.1E-11*T^4
          -12.15*LOG(T)
5470    PS(H)=EXP(PS)
5480 NEXT H
5490 REM CALCULATION OF ABSOLUTE HUMIDITY AND VAPOUR PRESSURE FOR AMBIENT,
       FARROWING,AND NURSERY AIR.
5500 FOR H = 0 TO 2
5510    PW(H)=RH(H)*PS(H)
5520    W(H)=.622*(PW(H)/(PA-PW(H)))
5530 NEXT H
5540 REM ABSOLUTE HUMIDITY,VAPOUR PRESSURE,AND REL. HUMIDITY OF THE HALL AND
       OUTLET AIR.
5550 W(3)=W(0)
5560 PW(3)=PW(0): RH(3)=PW(3)/PS(3)
5620 REM CALCULATION OF SENSIBLE,LATENT,AND TOTAL ENTHALPIES.
5630 REM ENTHALPY VALUES VALID FOR -50 C < T > 110 C.
5640 FOR H = 0 TO 3
5650    HS(H)=1.01*(TR(H,1)-0)
5660    HL(H)=W(H)*(2501+1.78*(TR(H,1)-0))
5670    HT(H)=HS(H)+HL(H)
5680    REM CALCULATION OF SPECIFIC VOLUMES, AND DENSITIES.
5690    SVOL(H)=(.287*(TR(H,1)+273))/(PA-PW(H))
5700    DEN(H)=(1+W(H))/SVOL(H)
5710 NEXT H
5711 FOR K = 1 TO 2
```

```
5712     SH(K)=1.01*(NT(K)-0)
5713     Q6(K)=Q6(K)+2500*(SH(K)-HS(K))*AC(K)*DEN(K)/10800: REM DIFFERENCE IN
      SENSIBLE HEAT BETWEEN ROOM TEMP AND DESIRED TEMP,ADDED TO ROOM HEAT LOAD.
5714 NEXT K
5720 RETURN
5730 PS=24.28-6238/T-.3444*LOG(T)
5740 GOTO 5470
5800 REM SUBROUTINE WHICH CALCULATES THE VENTILATION RATES FOR THE BUILDING,
        FAROWING,NURSERY,ANDTOTALS.
5810 L=0: X=0: Y=0: Z=0: ZZ=0
5820 MAXVENT(1)=6600: MAXVENT(2)=7650
5825 FOR K = 1 TO 2
5830     DSEN(K)=HS(K)-HS(L):REM SENSIBLE HEAT POTENTIAL kJ/kg
5835     IF DSEN(K)=0 OR DSEN(K)<0 THEN VSEN(K)=MAXVENT(K):MASS(K)=VSEN(K)/
         (SVOL(0)*1000)
5836     IF K=1 AND DSEN(1)=0 OR K=1 AND DSEN(1)<0 THEN Z=1: GOTO 5860
5837     IF K=2 AND DSEN(2)=0 OR K=2 AND DSEN(2)<0 THEN ZZ=2: GOTO 5860
5840     MASS(K)=Q6(K)/(DSEN(K)*1000): REM MASS FLOW RATE kg/s
5850     VSEN(K)=MASS(K)*SVOL(L)*1000: REM ROOM VENT. RATE L/s
5860 NEXT K
5870 VSEN=VSEN(1)+VSEN(2): REM TOTAL VENTILATION L/s
5880 MASS=MASS(1)+MASS(2): REM TOTAL MASS FLOW kg/s
5890 MASS3=AC(3)*2.5*DEN(3): REM AIR MASS OF HALLWAY
5900 VKJ=MASS*HS(L)*10800: REM SENSIBLE CONTENT OF VENT. AIR kJ
5910 VKJ3=MASS3*HS(3): REM SENSIBLE CONTENT OF HALLWAY kJ
5920 SMIX=(VKJ+VKJ3)/(MASS*10800+MASS3): REM SENSIBLE CONTENT OF MIX kJ/kg
5930 TMIX=(MASS3*HT(3)+MASS*HT(L)*10800)/(MASS3+MASS*10800):REM TOTAL ENTHALPY
     OF MIX kJ/kg.
5940 LMIX=TMIX-SMIX: REM LATENT CONTENT OF AIR MIXTURE kJ/kg.
5950 NEWT3=SMIX/1.01: REM NEW HALL TEMPERATURE (C).
5960 NEWW=(MASS3*W(3)+MASS*10800*W(L))/(MASS3+MASS*10800): REM MIXTURE ABS.
     HUMIDITY kg/kg.
5970 PW=PA*NEWW/1.622: REM DERIVED FROM LINE 5520
5980 T=NEWT3+273: REM HALL TEMPERATURE (K)
5990 PS=89.63-7512/T+.024*T-1.165E-05*T^2-1.281E-08*T^3+2.1E-11*T^4
        -12.15*LOG(T)
6000 PS=EXP(PS)
6010 NEWRH=PW/PS: REM REL. HUMIDITY OF AIR MIXTURE (%).
6020 NEWVOL=(.287*T)/(PA-PW): REM SPEC. VOLUME OF AIR MIXTURE m^3/kg.
6030 IF X = 1 GOTO 6220
6040 NEWVENT=0
6045 IF Z=1 AND ZZ=2 GOTO 6220
6046 IF Z=1 AND ZZ=0 GOTO 6190
6050 FOR K = 1 TO 2
6060     NEWVENT(K)=(Q6(K)/((HS(K)-SMIX)*1000))*1000*NEWVOL: REM L/s
6070     NEWVENT=NEWVENT+NEWVENT(K)
6080 NEXT K
6090 IF NEWVENT/VSEN > .95 AND NEWVENT/VSEN < 1.05 THEN Y=1
6100 FOR K = 1 TO 2
6110     VSEN(K)=NEWVENT(K): REM L/s
6120     MASS(K)=NEWVENT(K)/(NEWVOL*1000): REM kg/s.
6130 NEXT K
6140 IF Y=0 GOTO 5870
6150 IF VSEN(1) > MAXVENT(1) THEN VSEN(1)=MAXVENT(1): X=1
6160 IF VSEN(2) > MAXVENT(2) THEN VSEN(2)=MAXVENT(2): X=1
6165 MASS(1) = VSEN(1)/(NEWVOL*1000)
6170 MASS(2) = VSEN(2)/(NEWVOL*1000)
6175 VSEN=VSEN(1)+VSEN(2)
6180 MASS=MASS(1)+MASS(2)
6185 IF X = 1 GOTO 5890
6189 GOTO 6220
6190 NEWVENT(2)=(Q6(2)/((HS(2)-SMIX)*1000))*1000*NEWVOL
6195 NEWVENT=NEWVENT(2)+VSEN(1)
6200 IF NEWVENT/VSEN > .95 AND NEWVENT/VSEN < 1.05 THEN Y=1
6205 VSEN(2)=NEWVENT(2): MASS(2)=VSEN(2)/(NEWVOL*1000)
6210 IF Y=0 GOTO 5870
6215 GOTO 6150
6220 FOR K = 1 TO 2
6230     DS(K)=Q6(K)/1000-(HS(K)-SMIX)*MASS(K): REM kW
6240     DL(K)=(LHEAT(K)*NA(K))/1000-(HL(K)-LMIX)*MASS(K):REM LATENT POTENTIAL kW
6250     SENS(K)=HS(K)+DS(K)/MASS(K): REM NEW SENSIBLE CONTENT OF ROOM kJ/kg
6260     LAT(K)=HL(K)+DL(K)/MASS(K): REM NEW LATENT CONTENT OF ROOM kJ/kg>
```

```
6270    NT(K)=SENS(K)/1.01: REM NEW ROOM TEMPERATURE (C).
6280    RMASS(K)=AC(K)*2.5*DEN(K): REM ROOM AIR MASS kg.
6290    NEWW(K)=(RMASS(K)*W(K)+NEWW*MASS(K)*10800)/(RMASS+MASS(K)*10800):
        REM NEW ROOM ABSOLUTE HUMIDITY kg/kg
6300    PW(K)=PA*NEWW(K)/1.622: REM ROOM VAPOUR PRESSURE kPa.
6310    T=NT(K)+273
6320    PS=89.63-7512/T+.024*T-1.165E-05*T^2-1.281E-08*T^3+2.1E-11*T^4
           -12.15*LOG(T)
6330    PS(K)=EXP(PS)
6340    NEWRH(K)=PW/PS: REM NEW ROOM RELATIVE HUMIDITY (%).
6350 NEXT K
6360 REM ROUTINE WHICH RE-ASSIGNS VARIABLE VALUES
6370 FOR K = 1 TO 2
6380    RH(K)=NEWRH(K)
6390 NEXT K
6400 TR(3,0)=TR(3,1)
6410 TR(3,1)=NEWT3
6420 RETURN
7000 REM SUBROUTINE WHICH FORMATS THE TWO HEADING LINES FOR THE FILE
7010 LSET DTE$=STR$(REC): LSET TME$=STR$(SET-1): LSET T$(1)="FARROW":LSET T$(2)
     ="NURSERY": LSET V$(3)="TOTAL": LSET T$(3)="AMBIENT"
7020 FOR G = 1 TO 4: LSET Q$(G)=CHR$(34): NEXT G
7030 FOR G = 1 TO 2: LSET R$(G)="     ": LSET V$(G)="      ":NEXT G
7040 LSET RET$=CHR$(13): LSET LF$=CHR$(10)
7050 FOR V = 1 TO 8: LSET D$(V)="    ": NEXT V
7060 PUT #2,1
7070 LSET DTE$="    DATE":LSET TME$="  TIME":LSET T$(1)="  TEMP":LSET R$(1)=" RH"
     :LSET V$(1)=" VENT":LSET T$(2)="  TEMP":LSET R$(2)=" RH":LSET V$(2)=" VENT"
7080 LSET V$(3)=" VENT":LSET T$(3)="  TEMP"
7090 FOR G = 1 TO 4: LSET Q$(G)=CHR$(34): NEXT G
7100 FOR V = 1 TO 8: LSET D$(V)="    ": NEXT V
7110 LSET RET$=CHR$(13): LSET LF$=CHR$(10)
7120 PUT #2,2
7130 RETURN
7140 REM SUBROUTINE WHICH PUTS THE CALCULATED VALUES ONTO DISK
7150 LSET DTE$=DAT$
7160 LSET TME$=TIM$
7170 FOR K = 1 TO 2
7180    LSET T$(K)=STR$(NT(K))
7190    LSET R$(K)=STR$(RH(K))
7200    LSET V$(K)=STR$(VSEN(K))
7210 NEXT K
7220 LSET V$(3)=STR$(VSEN): LSET T$(3)=STR$(T(5))
7230 FOR G = 1 TO 4: LSET Q$(G)=CHR$(34): NEXT G
7240 FOR V = 1 TO 8: LSET D$(V)="    ": NEXT V
7250 LSET RET$=CHR$(13): LSET LF$=CHR$(10)
7260 PUT #2,I
7270 RETURN
7280 REM SUBROUTINE WHICH FORMATS THE HEADING LINE FOR MISSING DATA
7290 LSET DTE$="MISSING   ":LSET TME$=STR$(SET-1)
7300 FOR G = 1 TO 4: LSET Q$(G)=CHR$(34): NEXT G
7310 FOR K = 1 TO 2
7320    LSET T$(K)="       "
7330    LSET R$(K)="    "
7340    LSET V$(K)="      "
7350 NEXT K
7360 LSET V$(3)="      ": LSET T$(3)="        "
7370 FOR V = 1 TO 8: LSET D$(V)="    ": NEXT V
7380 LSET RET$=CHR$(13): LSET LF$=CHR$(10)
7390 PUT #2,I
7400 RETURN
```

```
10 REM   SOL-AIR                                            JANUARY 24,1988
20 REM PROGRAM WHICH READS AMBIENT, AND OUTLET TEMPERATURE DATA, AND COMBINES
       THEM WITH SOLAR RADIATION DATA TO CALCULATE THE SOL-AIR TEMPERATURE.
30 REM THE SOL-AIR TEMPERATURES ARE CALCULATED FOR THE ROOF,EAST,SOUTH AND WEST
       WALLS. (SOL-AIR TEMPERATURE FOR THE NORTH WALL IS ASSUMED EQUAL TO THE
       AMBIENT TEMPERATURE).
40 REM THE SOLAR RADIATION INCIDENT ON THE VARIOUS SURFACES IS CALCULATED USING
       CANADIAN CLIMATE NORMALS DATA, AND THE INCIDENT AND ZENITH ANGLE FOR THE
       CORRESPONDING SURFACE.
50 SL=.7649:CL=.6441: SS(0)=.3162:CS(0)=.9487:B(0)=.7
60 FOR X= 1 TO 3
70      SS(X)=1:CS(X)=0:B(X)=.4
80 NEXT X
90 CG(0)=1:SG(0)=0:CG(1)=0:SG(1)=1:CG(2)=1:SG(2)=0:CG(3)=0:SG(3)=-1
100 KEY OFF
110 REM ROUTINE WHICH ASSIGNS SOLAR INTENSITY VALUES TO AN ARRAY
120 DIM RAD(5,24)
130 FOR I = 1 TO 5
140      FOR J = 0 TO 23
150              READ RAD(I,J)
160      NEXT J
170 NEXT I
180 REM *****  SOLAR RAD. DATA FOR THE MONTH OF MAY         *******************
190 DATA 0,0,0,0,.05,.32,.74,1.21,1.65,2.01,2.27,2.39,2.35,2.21,1.94,1.57,1.15,
        .71,.3,.05,0,0,0,0
200 REM *****  SOLAR RAD. DATA FOR THE MONTH OF JUNE        *******************
210 DATA 0,0,0,.01,.12,.45,.88,1.36,1.78,2.13,2.36,2.51,2.47,2.30,2.02,1.68,1.27
        ,.85,.43,.12,.01,0,0,0
220 REM *****  SOLAR RAD. DATA FOR THE MONTH OF JULY        ********************
230 DATA 0,0,0,0,.09,.4,.85,1.34,1.8,2.19,2.44,2.56,2.55,2.4,2.09,1.71,1.28,.82,
        .39,.08,0,0,0,0
240 REM *****  SOLAR RAD. DATA FOR THE MONTH OF AUGUST      ********************
250 DATA 0,0,0,0,.01,.18,.59,1.07,1.52,1.87,2.11,2.25,2.25,2.08,1.81,1.45,1.02,
        .57,.17,.01,0,0,0,0
260 REM *****  SOLAR RAD. DATA FOR THE MONTH OF SEPTEMBER   ********************
270 DATA 0,0,0,0,0,.02,.23,.63,1.07,1.44,1.67,1.77,1.76,1.61,1.34,.97,.58,.20,
        .02,0,0,0,0,0
280 REM ROUTINE WHICH RECALLS THE DATA FROM THE STORAGE FILE
290 INPUT "INPUT DATA FILENAME";FILE$
300 OPEN "R",#1,"C:"+FILE$+".PRN",88
310 FIELD #1,1 AS Q1$,10 AS DAT$,1 AS Q2$,1 AS Q3$,8 AS TIM$,1 AS Q4$,40 AS
    DUM$,8 AS SEN$(6),8 AS SEN$(7),8 AS SEN$(8),2 AS DUM2$
320 OPEN "R",#2,"C:SOL-AIR.PRN",80
330 FIELD #2,1 AS Q$(1),10 AS DA$,1 AS Q$(2),1 AS Q$(3),8 AS TI$,1 AS Q$(4),8 AS
    T$(0),8 AS T$(1),8 AS T$(2),8 AS T$(3),8 AS T$(4),8 AS T$(5),8 AS T$(6),1 AS
    RET$,1 AS LF$
340 GET #1,1
350 REC = VAL(DAT$)
360 GOSUB 940
370 FOR I = 2 TO REC
380 CLS
390 GET #1,I
400 LOCATE 12,6:PRINT "COMPUTER PROGRAM IN PROGRESS"
410 LOCATE 25,10:PRINT "RECORD #";I;" OF";REC;" RECORDS"
420 IF DAT$="MISSING    " THEN GOSUB 1010
430 MOIS=VAL(MID$(DAT$,1,2)): JOUR=VAL(MID$(DAT$,4,2)): HEURE=VAL(MID$(TIM$,1,2)
    ): AMB=(VAL(SEN$(7))+VAL(SEN$(8)))/2
440 IF MOIS <> 5 GOTO 460
450      N=JOUR+120:E=4
460 IF MOIS <> 6 GOTO 480
470      N=JOUR+151:E=1
480 IF MOIS <> 7 GOTO 500
490      N=JOUR+181:E=-5
500 IF MOIS <> 8 GOTO 520
510      N=JOUR+212:E=-5
520 IF MOIS <> 9 GOTO 540
530      N=JOUR+243:E=4
540 IF MOIS > 4 AND MOIS < 10 GOTO 570
550      PRINT "PROGRAM IS NOT VALID FOR THAT MONTH"
560      CLOSE:END
```

```
570 DECL=23.45*SIN(2*3.14159*(.9863*(284+N))/360)
580 CD=COS(2*3.14159*(DECL/360)): SD=SIN(2*3.14159*(DECL/360))
590 SOLTIME=HEURE+(E-28.93)/60
600 W=2*3.14159*(15*(12-SOLTIME))/360: CW=COS(W):SW=SIN(W)
610 REM CALCULATION OF INCIDENCE AND ZENITH ANGLES FOR EACH OF THE EXTERIOR
       SURFACES.
620 FOR J=0 TO 3
630      K(J)=SD*SL*CS(J)-SD*CL*SS(J)*CG(J)+CD*CL*CS(J)*CW+CD*SL*SS(J)*CW*CG(J)+
              CD*SS(J)*SW*SG(J)
640 NEXT J
650 Z=CD*SL+CD*CW*CL
660 REM ROUTINE WHICH INTERPOLATES THE SOLAR RADIATION VALUE FOR THAT PARTICULAR
       DAY.
670 IF JOUR = 21 THEN H=RAD(MOIS-4,HEURE):GOTO 790
680 IF N>172 AND N<202 GOTO 740
690 IF JOUR > 21 GOTO 720
700      I1=RAD(MOIS-5,HEURE): I2=RAD(MOIS-4,HEURE)
710      H=(I2-I1)*(JOUR+10)/31+I1:GOTO 790
720      I1=RAD(MOIS-4,HEURE):I2=RAD(MOIS-3,HEURE)
730      H=(I2-I1)*(JOUR-21)/31+I1:GOTO 790
740 IF JOUR < 21 GOTO 770
750      I1=RAD(MOIS-4,HEURE):I2=RAD(MOIS-3,HEURE)
760      H=(I2-I1)*(JOUR-21)/30+I1:GOTO 790
770      I1=RAD(MOIS-5,HEURE):I2=RAD(MOIS-4,HEURE)
780      H=(I2-I1)*(JOUR+9)/30+I
790 FOR K=0 TO 3
800      H(K)=H*277.8*K(K)/Z
810      SOL(K)=AMB+B(K)*H(K)/23
820 NEXT K
830 SOL(4)=AMB
840 LSET DA$=DAT$:LSET TI$=TIM$
850 FOR X = 1 TO 4:LSET Q$(X)=CHR$(34):NEXT X
860 LSET RET$=CHR$(13): LSET LF$=CHR$(10)
870 FOR X = 0 TO 4: LSET T$(X)=STR$(SOL(X)):NEXT X
880 LSET T$(5)=STR$(AMB): LSET T$(6)=SEN$(6)
890 PUT #2,I
900 NEXT I
910 CLOSE
920 CLS: LOCATE 12,6:PRINT "COMPUTER PROGRAM FINISHED"
930 END
940 REM SUBROUTINE WHICH SAVES THE HEADING FOR "SOL-AIR.PRN"
950 LSET DA$=STR$(REC):LSET TI$=TIM$
960 FOR X = 1 TO 4:LSET Q$(X)=CHR$(34):NEXT X
970 LSET T$(0)="  ROOF": LSET T$(1)="  EAST": LSET T$(2)=" SOUTH":LSET T$(3)=
    "  WEST": LSET T$(4)=" NORTH": LSET T$(5)="AMBIENT": LSET T$(6)="OUTLET"
980 LSET RET$=CHR$(34): LSET LF$=CHR$(10)
990 PUT #2,1
1000 RETURN
1010 REM SUBROUTINE WHICH PLACES MISSING DATA FLAG INTO THE NEW DATASET.
1020 LSET DA$=DAT$: LSET TI$=TIM$
1030 FOR X = 1 TO 4: LSET Q$(X)=CHR$(34): NEXT X
1040 FOR X = 0 TO 6: LSET T$(X)="        ": NEXT X
1050 LSET RET$=CHR$(13): LSET LF$=CHR$(10)
1060 PUT #2,I
1070 RETURN 900
```

```
10 REM  MEANS1.BAS                                          SEPT. 10, 1987
15 REM                                           LATEST REVISION: JAN. 29, 1988
20 REM PROGRAM WRITTEN TO CALCULATE THE DAILY MEAN VALUES OF SOL-AIR AND AMBIENT
       TEMPERATURES.
30 REM VALUES OF EXPOSURE RELATED SOL-AIR TEMP'S ARE TAKEN FROM SOL-AIR.PRN, AND
       THE DAILY MEANS ARE CALCULATED.
40 REM THE NEW DATASET IS STORED IN THE FILE MEANSP1.PRN.
50 KEY OFF
60 OPEN "R",#1,"C:SOL-AIR1.PRN",80
70 OPEN "R",#2,"C:MEANSP1.PRN",136
110 FIELD #1,1 AS Q1$,10 AS DAT$,1 AS Q2$,1 AS Q3$,8 AS TIM$,1 AS Q4$,8 AS T$(0)
,    8 AS T$(1),8 AS T$(2),8 AS T$(3),8 AS T$(4),8 AS T$(5),8 AS T$(6),1 AS RET$,
     1 AS LF$
120 FIELD #2,1 AS Q$(1),10 AS DA$,1 AS Q$(2),1 AS Q$(3),8 AS TI$,1 AS Q$(4),8 AS
     S$(0),8 AS M$(0)
130 FOR I = 1 TO 6
140 FIELD #2,(22+16*I) AS DUMMY$,8 AS S$(I),8 AS M$(I)
150 NEXT I
160 FIELD #2,134 AS DUMMY$,1 AS RET$,1 AS LF$
170 J=1
180 GET #1,1
190 REC = VAL(DAT$)
200 IF J > REC GOTO 810
210 GET #1,J: I=1
220 SET=VAL(TIM$)
230 GOSUB 840
240 WHILE I < SET OR I = SET
250 IF I > 8 GOTO 460
260 FOR X= 1 TO 8
270    GET #1,X+(J-1)
280    FOR Y = 0 TO 6
290             T(X,Y)=VAL(T$(Y))
300     NEXT Y
310 NEXT X
320 FOR Y = 0 TO 6
330     FOR X = 1 TO 8
340             S(Y)=S(Y)+T(X,Y)
350             M(Y)=S(Y)/8
360     NEXT X
370 NEXT Y
380 GET #1,J
390 LSET DA$=DAT$: LSET TI$=TIM$
400 FOR G=1 TO 4: LSET Q$(G)=CHR$(34): NEXT G
410 FOR G=0 TO 6: LSET S$(G)=T$(G): LSET M$(G)=STR$(M(G)): NEXT G
420 LSET RET$=CHR$(13): LSET LF$=CHR$(10)
430 PUT #2,J: J=J+1: I=9
440 CLS: LOCATE 12,20: PRINT "RECORD #";J;" OF";REC;" RECORDS"
450 GOTO 240
460 FOR X = 2 TO 8
470     FOR Y= 0 TO 6
480             T(X-1,Y)=T(X,Y)
490     NEXT Y
500 NEXT X
510 GET #1,J+7
520 FOR Y=0 TO 6
530     T(8,Y)=VAL(T$(Y))
540 NEXT Y
550 FOR Y=0 TO 6
560     S(Y)=0: M(Y)=0
570     FOR X= 1 TO 8
580             S(Y)=S(Y)+T(X,Y)
590     NEXT X
600      M(Y)=S(Y)/8
610 NEXT Y
620 GET #1,J
630 LSET DA$=DAT$: LSET TI$=TIM$
640 FOR G=1 TO 4: LSET Q$(G)=CHR$(34): NEXT G
650 FOR G=0 TO 6: LSET S$(G)=T$(G): LSET M$(G)= STR$(M(G)): NEXT G
660 LSET RET$=CHR$(13): LSET LF$=CHR$(10)
670 PUT #2,J: J=J+1:I=I+1
```

```
680 CLS: LOCATE 12,20: PRINT "RECORD #";J;" OF";REC;" RECORDS"
690 WEND
700 REM CALCULATIONS FOR THE MEANS FOR THE LAST 8 RECORDS OF EACH SET
705 MARK=J
710 FOR X = MARK TO MARK+6
720     GET #1,X
730     LSET DA$=DAT$: LSET TI$=TIM$
740     FOR G = 1 TO 4: LSET Q$(G)=CHR$(34): NEXT G
750     FOR G = 0 TO 6: LSET S$(G)=T$(G): LSET M$(G)=STR$(M(G)): NEXT G
760     LSET RET$=CHR$(13): LSET LF$=CHR$(10)
770     PUT #2,X: J=J+1
780 CLS: LOCATE 12,20: PRINT "RECORD #";J;" OF";REC;" RECORDS"
790 NEXT X
800 GOTO 200
810 CLS: LOCATE 12,20: PRINT "COMPUTER PROGRAM IS FINISHED"
820 CLOSE
830 END
840 REM SUBROUTINE WHICH PLACES THE HEADING LINES FOR THE DATASET AND THE
    SUBSETS.
850 LSET DA$=DAT$: LSET TI$=TIM$
860 FOR G= 1 TO 4: LSET Q$(G)=CHR$(34): NEXT G
870 LSET S$(0)="  ROOF": LSET S$(1)="  EAST": LSET S$(2)=" SOUTH"
880 LSET S$(3)="  WEST": LSET S$(4)=" NORTH": LSET S$(5)="AMBIENT"
890 LSET S$(6)=" OUTLET"
900 FOR G = 0 TO 6: LSET M$(G)="  MEAN": NEXT G
910 LSET RET$=CHR$(13): LSET LF$=CHR$(10)
920 PUT #2,J: J=J+1
930 RETURN
```

APPENDIX B

Program Variables Nomenclature

Structure Parameters

AR	Roof Area
AG(n)	Gable Area
AC(n)	Ceiling Area n= 1,2,3
AW(n)	Wall Area n=1,2,3
AIW(n)	Inside Wall Area n=1,2,3
P(n)	Length of Floor Perimeter n=1,2,3
	where; 1 = Farrowing Rooms
	2 = Nursery Rooms
	3 = Hall
UC	Ceiling Heat Transmission Coefficient
UW	Exterior Wall Heat Transmission Coefficient
UR	Roof Heat Transmission Coefficient
UG	Gable Heat Transmission Coefficient
UFL	Floor Heat Transmission Coefficient
UIW	Inside Wall Heat Transmission Coefficient

Animal Parameters

RH(n);	System Relative Humidities n=0,1,2
	where; 0 = Ambient
	1 = Farrowing Room
	2 = Nursery
NA(n)	Number of Animals n=1,2,3
SHEAT(n)	Sensible Heat Production per Animal n=0,1,2
LHEAT(n)	Latent Heat Production per Animal n=0,1,2
	where; 1 = Sows
	2 = Nursery
	3 = Hall

Miscellaneous Variables

PA	Atmospheric Pressure
AF	Airflow Rate
TA	Ambient Temperature
TAT	Attic Temperature

Miscellaneous Variables (Continued)

T(n)	Design Room Temperatures n=0,1,2,3,4 where; 0 = Ambient Temperature 1 = Farrowing Room 2 = Nursery Room 3 = Hall 4 = Outlet Temperature
DT3	Iteration Differential Temperature
KT3	Hall Temperature Array
KTATT	Attic Temperature Array
PS(n)	Saturation Vapour Pressure; n = 0 to 4
PW(n)	Actual Vapour Pressure; n = 0 to 4
W(n)	Absolute Moisture Content; n = 0 to 4
HS(n)	Sensible Heat of Room; n = 0 to 4
HL(n)	Latent Heat of Room; n = 0 to 4
HT(n)	Total Enthalpy of each Room; n = 0 to 4
SVOL(n)	Air Specific Volume of each Room; n = 0 to 4
DEN(n)	Air Density of each Room; n = 0 to 4
MAMB	Mass flow of ambient air required
AMBVENT	Volume of ambient air required
DLAT(n)	Difference in latent energy between the hall and the rooms
VLAT(n)	Volume of air required for moisture control
MFLOW	Mass flow of required ventilation air
MTEMP	Mass flow of Tempered air required
VTEMP	Tempered Air Volume Required
PIPES	Number of Pipes Required
PINC	Pipe Increment
MAXTEMP	Maximum Air Flow of System
MASS3	Mass of the control volume within the hall
VKJ	Sensible heat of incoming air
VKJ3	Sensible heat of hall control volume
SMIX	Sensible heat content of hall and incoming air mixture
NEWT3	Resultant hall temperature
VENT	Ventilation Rate

Program Parameters

START	First Data File
LAST	Last Data File
X	Data Count
FILE$	Current Data File
FILE2$	Current Output File
PN	Number of Pipe to Model (1, 3, or 4)

Program Parameters (Continued)

PL	Pipe Length
NPIPES	Number of Pipes in the Simulation
W	Recalculation Flag for Hall Temperatures

Input Variables

DA$	Input Date of Record
TI$	Output Time of Record
SEN$(n)	Sensor Number n=6,7,8
	where; 6 & 7 Ambient temperatures
	8 Pipe Outlet Temperature

Output Variables

DTE$	Output Date of Record
TME$	Output Time of Record
TEMP$	Ambient Air Temperature
OUTTEMP$	Outlet Air Temperature
DQ$	System Heat Balance
VENT$	Total Ventilation Air Required
VT$	Total Tempered Air Supplied
VA$	Total Ambient Air Supplied

APPENDIX C

Net Present Value and Cashflow Tables

Table C-1 Net present value for System 1 - Pipe 1, 29 m pipe lateral lengths

System Capital Cost	$	119,866
Annual Heating Requirement		13,569 kW
Cost Of Electrical Energy	$	0.03 /kW
Net Cost	$	407 /yr
Real Rate of Interest		5.0%
Number of Years		20
Net Present Value		(124,785)

Cash Flow

Year	0	1	2	...	19
System Cost	119,866	0	0		0
Energy Cost	0	407	407		407
Total Annual Cost	119,866	407	407		407

Table C-2 Net present value for System 2 - Pipe 1, 20 m pipe lateral lengths

System Capital Cost	$	100,807
Annual Heating Requirement		15,813 kW
Cost Of Electrical Energy	$	0.03 /kW
Net Cost	$	474 /yr
Real Rate of Interest		5.0%
Number of Years		20
Net Present Value		(106,540)

Cash Flow

Year	0	1	2	...	19
System Cost	100,807	0	0		0
Energy Cost	0	474	474		474
Total Annual Cost	100,807	474	474		474

Table C-3 Net present value for System 3 - Pipe 3, 29 m pipe lateral lengths

System Capital Cost	$	61,256
Annual Heating Requirement		19,721 kW
Cost Of Electrical Energy	$	0.03 /kW
Net Cost	$	592 /yr
Real Rate of Interest		5.0%
Number of Years		20
Net Present Value		(68,406)

Cash Flow

Year	0	1	2	...	19
System Cost	61,256	0	0		0
Energy Cost	0	592	592		592
Total Annual Cost	61,256	592	592		592

Table C-4 Net present value for System 4 - Pipe 3, 20 m Pipe Lengths

System Capital Cost	$	55,594
Annual Heating Requirement		21,896 kW
Cost Of Electrical Energy	$	0.03 /kW
Net Cost	$	657 /yr
Real Rate of Interest		5.0%
Number of Years		20
Net Present Value		(63,532)

Cash Flow

Year	0	1	2	...	19
System Cost	55,594	0	0		0
Energy Cost	0	657	657		657
Total Annual Cost	55,594	657	657		657

Table C-5 Net present value for System 5 - Pipe 4, 29 m pipe lateral lengths

System Capital Cost	$	63,410
Annual Heating Requirement		11,985 kW
Cost Of Electrical Energy	$	0.03 /kW
Net Cost	$	360 /yr
Real Rate of Interest		5.0%
Number of Years		20
Net Present Value		(67,755)

Cash Flow

Year	0	1	2	...	19
System Cost	63,410	0	0		0
Energy Cost	0	360	360		360
Total Annual Cost	63,410	360	360		360

Table C-6 Net present value of System 6 - Pipe 4, 20 m pipe lateral lengths

System Capital Cost	$	52,217
Annual Heating Requirement		16,502 kW
Cost Of Electrical Energy	$	0.03 /kW
Net Cost	$	495 /yr
Real Rate of Interest		5.0%
Number of Years		20
Net Present Value		(58,200)

Cash Flow

Year	0	1	2	...	19
System Cost	52,217	0	0		0
Energy Cost	0	495	495		495
Total Annual Cost	52,217	495	495		495

Table C-7 Net present value of a conventional system - No air tempering.

System Capital Cost	$	8,250
Annual Heating Requirement		30,546 kW
Cost Of Electrical Energy	$	0.03 /kW
Net Cost	$	916 /yr
Real Rate of Interest		5.0%
Number of Years		20
Net Present Value		(19,325)

Cash Flow

Year	0	1	2	...	19
System Cost	19,325	0	0		0
Energy Cost	0	916	916		916
Total Annual Cost	19,325	916	916		916