AN ABSTRACT OF THE THESIS OF

Chaur-Fong Chen for the degree of Master of Science in Agricultural Engineering presented on June 13, 1988.

Title: Microprocessor Control and Numerical Model for Broiler House Summer Ventilation Utilizing a Rockbed Heat Sink

Abstract approved:  

Associate Professor  M.L. Hellickson

Summer brooding often requires some form of air modification in addition to maximizing ventilation rate to prevent animal heat stress and possible death due to hyperthermia. A rockbed thermal storage module was designed and constructed to provide sensibly cooled air for a broiler space under summer brooding conditions. A mathematical model of rockbed thermal storage module was developed to predict performance of the prototype module. Experiments to evaluate the rockbed module under different weather conditions and operating schedules were conducted. The results were presented and compared with the mathematical simulation.

A microprocessor control system was designed and assembled to control the ventilation air that would enter a broiler house during warm weather periods. The system was capable of taking temperature samples from thermocouples and then operate a damper arrangement which determined the mixture of outdoor and sensibly cooled air from rockbed thermal storage module. The cost and feasibility of utilizing a rockbed thermal storage module and a microprocessor control ventilation system were discussed.
Microprocessor Control and Numerical Model for Broiler House Summer Ventilation Utilizing a Rockbed Heat Sink

by

Chaur-Fong Chen

A THESIS
submitted to
Oregon State University

in partial fulfillment of the requirements for the degree of

Master of Science

Completed June 13, 1988
Commencement June 1989
ACKNOWLEDGEMENT

Many support and resources had been contributed to the completion of this thesis project. My deepest gratitude to Dr. Martin L. Hellickson, for his patient guidance and consistent encouragement throughout the completion of this thesis. He was always available not just as an advisor but also as a friend. Special thanks to Dr. Marshall English for his technical and partially financial support during this period.

I would like to share the completion of this thesis with Mr. Kent Martin, chief, Electronic and instrumentation division, Bonneville Power Administration. He had provided great advices on configuring and debugging the electronic circuits.

Dr. Alan Robinson deserves a special thanks for his help in numerical simulation portion of the thesis and the computer resources he provided.

I would also like to express my deep appreciation to the faculty, staff and fellow graduate students in the Agricultural Engineering Department. I will always remember their help and friendship. Special thanks to E. Stuart Baker for his guidance and technical support in computer resources.

And finally, I wish to express my most sincere gratitude to my dear wife Jang Luh for her considerate understanding and the emotional support. She was the invaluable source during this work and had made the completion of this thesis possible.
# TABLE OF CONTENTS

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>INTRODUCTION</strong></td>
<td>1</td>
</tr>
<tr>
<td><strong>LITERATURE REVIEW</strong></td>
<td>5</td>
</tr>
<tr>
<td>Feasibility of Rock Bed Storage System</td>
<td>5</td>
</tr>
<tr>
<td>Rock Bed Storage Applications and Efficiency Analysis</td>
<td>6</td>
</tr>
<tr>
<td>Energy Balance</td>
<td>10</td>
</tr>
<tr>
<td>Simulation of Packed Bed Performance</td>
<td>17</td>
</tr>
<tr>
<td>Broiler House Ventilation</td>
<td>22</td>
</tr>
<tr>
<td>Control Configuration</td>
<td>24</td>
</tr>
<tr>
<td><strong>MATHEMATICAL MODEL</strong></td>
<td>27</td>
</tr>
<tr>
<td>Energy Balance Formulation</td>
<td>27</td>
</tr>
<tr>
<td>Air system</td>
<td>28</td>
</tr>
<tr>
<td>Rock system</td>
<td>30</td>
</tr>
<tr>
<td>Finite Difference Approach</td>
<td>33</td>
</tr>
<tr>
<td>Broiler House Ventilation Model</td>
<td>42</td>
</tr>
<tr>
<td>Broiler house energy balance</td>
<td>42</td>
</tr>
<tr>
<td>Broiler house environmental Requirements</td>
<td>48</td>
</tr>
<tr>
<td>Modeling broiler heat and moisture production</td>
<td>49</td>
</tr>
<tr>
<td>Rock Bed Design and System Operation</td>
<td>53</td>
</tr>
<tr>
<td><strong>MICROPROCESSOR CONTROL SYSTEM</strong></td>
<td>56</td>
</tr>
<tr>
<td>Control Theory</td>
<td>56</td>
</tr>
<tr>
<td>Control Method</td>
<td>57</td>
</tr>
<tr>
<td>System Requirements and Specifications</td>
<td>59</td>
</tr>
<tr>
<td>Central Process Unit (CPU) and Associated Memory</td>
<td>60</td>
</tr>
<tr>
<td>Input, Output and Other Supporting Devices</td>
<td>62</td>
</tr>
<tr>
<td>Data Acquisition and Interface</td>
<td>63</td>
</tr>
<tr>
<td>Controller Block</td>
<td>75</td>
</tr>
<tr>
<td><strong>PROCEDURE</strong></td>
<td>83</td>
</tr>
<tr>
<td>Computer Simulation</td>
<td>83</td>
</tr>
<tr>
<td>Rock Bed Construction</td>
<td>83</td>
</tr>
<tr>
<td>System Properties Measurement</td>
<td>88</td>
</tr>
<tr>
<td>Rock Bed System Operation</td>
<td>90</td>
</tr>
<tr>
<td>Microcomputer Assembly and Modification</td>
<td>92</td>
</tr>
<tr>
<td>Microprocessor Control</td>
<td>93</td>
</tr>
<tr>
<td>Damper Operation</td>
<td>98</td>
</tr>
<tr>
<td><strong>RESULTS AND DISCUSSION</strong></td>
<td>104</td>
</tr>
<tr>
<td>Development Stage Modification</td>
<td>104</td>
</tr>
</tbody>
</table>
Overall Rock Bed Thermal Storage Module Performance .... 104
Computer Simulation ............................................ 111
Circuit Construction and Operation Overview .......... 118
SUMMARY AND DISCUSSIONS ..................................... 121
BIBLIOGRAPHY ...................................................... 125
APPENDICES ......................................................... 130
  APPENDIX A Broiler House Ventilation Model .......... 130
  APPENDIX B Ventilation Requirement Calculation ...... 135
  APPENDIX C Ventilation Requirement Calculation ...... 137
  APPENDIX D Rockbed Performance Simulation Program .... 143
# LIST OF FIGURES

<table>
<thead>
<tr>
<th>Figure</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Computational stencil diagram</td>
<td>37</td>
</tr>
<tr>
<td>2. Terminal points</td>
<td>37</td>
</tr>
<tr>
<td>3. Slicing pattern of rockbed for finite difference</td>
<td>37</td>
</tr>
<tr>
<td>4. Major components of heat gain and cooling load</td>
<td>43</td>
</tr>
<tr>
<td>5. Simulated broiler house construction</td>
<td>52</td>
</tr>
<tr>
<td>6. Simplified negative feedback loop</td>
<td>58</td>
</tr>
<tr>
<td>7. Schematic diagram for a microprocessor system</td>
<td>58</td>
</tr>
<tr>
<td>8. Assembled ET-3400 microcomputer</td>
<td>61</td>
</tr>
<tr>
<td>9. Memory addressing map</td>
<td>61</td>
</tr>
<tr>
<td>10. Characteristics curves for different temperature sensors</td>
<td>64</td>
</tr>
<tr>
<td>11. Thermocouple output conditioning</td>
<td>67</td>
</tr>
<tr>
<td>12. Electric ice point reference</td>
<td>67</td>
</tr>
<tr>
<td>13. Hardware compensation circuit</td>
<td>67</td>
</tr>
<tr>
<td>14. Aliasing error during sampling process</td>
<td>69</td>
</tr>
<tr>
<td>15. Transfer function and error plot for ADC</td>
<td>69</td>
</tr>
<tr>
<td>16. First stage signal processing</td>
<td>72</td>
</tr>
<tr>
<td>17. Peripheral interface adapter internal structure</td>
<td>74</td>
</tr>
<tr>
<td>18. Tri-state buffer for direction control</td>
<td>74</td>
</tr>
<tr>
<td>19. A simplified stepping motor diagram</td>
<td>76</td>
</tr>
<tr>
<td>20. Stepping motor operation sequences</td>
<td>76</td>
</tr>
<tr>
<td>21. Center tapped stepping motor and control circuits</td>
<td>80</td>
</tr>
<tr>
<td>22. High transient voltage when motor shut off</td>
<td>80</td>
</tr>
<tr>
<td>23. Protection diode used to eliminate spike</td>
<td>80</td>
</tr>
<tr>
<td>24. Stepping motor control circuits</td>
<td>81</td>
</tr>
<tr>
<td>25. Complementary gear setup for air mixture</td>
<td>82</td>
</tr>
<tr>
<td>26. Thermocouple matrix setup on diagonal plane</td>
<td>84</td>
</tr>
<tr>
<td>27. Complete schematic diagram of rockbed chamber</td>
<td>86</td>
</tr>
<tr>
<td>28. Rockbed thermal storage module construction details</td>
<td>87</td>
</tr>
<tr>
<td>29. Rockbed front view</td>
<td>89</td>
</tr>
<tr>
<td>30. Sharp edged orifice for air flow measurement</td>
<td>89</td>
</tr>
<tr>
<td>31. Thermocouple node points grid inside rockbed</td>
<td>91</td>
</tr>
<tr>
<td>32. Peripheral interface adapter hierarchical control</td>
<td>95</td>
</tr>
<tr>
<td>33. PIA 6821 and associated circuit layout</td>
<td>96</td>
</tr>
<tr>
<td>34. Simplified rockbed system operation diagram</td>
<td>99</td>
</tr>
<tr>
<td>35. Thermal lag of rockbed outlet temperature</td>
<td>105</td>
</tr>
</tbody>
</table>
36. Time-temperature rate of change of rockbed system ........... 106
37. Internal thermal stability test ........................................ 108
38. Thermal lag operation test ........................................... 108
39. Smoothing effect under inlet temperature variation .............. 110
40. Effects of natural convection inside rockbed ....................... 112
41. Adequacy test for using interpolated initial temperature ....... 113
42. Simulated vs. recorded temperature (rapid change) .............. 115
43. Simulated vs. recorded temperature (mild change) .............. 116
# LIST OF TABLES

<table>
<thead>
<tr>
<th>Table</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Suggested brooding temperature regime</td>
<td>48</td>
</tr>
<tr>
<td>2. Broiler house environmental parameters</td>
<td>48</td>
</tr>
<tr>
<td>3. Approximate commercial broiler brooding dates</td>
<td>49</td>
</tr>
<tr>
<td>4. Simulated broiler house material characteristics</td>
<td>51</td>
</tr>
<tr>
<td>5. Total energy requirement calculation</td>
<td>55</td>
</tr>
<tr>
<td>6. Stepping motor driving bit patterns</td>
<td>82</td>
</tr>
<tr>
<td>7. System parameter values used in simulation</td>
<td>85</td>
</tr>
<tr>
<td>8. Test module system operation modes</td>
<td>91</td>
</tr>
<tr>
<td>9. Lookup table for damper operation</td>
<td>103</td>
</tr>
</tbody>
</table>
Production agriculture consumes approximately three percent of the United State's annual energy budget (Stout, 1984). Three percent may not appear as significant when compared to the nation's large energy consumers, however, this amount is extremely critical to food production. Adequate and timely energy supplies must be readily available to production agriculture and even a temporary shortage could cause drastic reductions in productivity.

Energy prices have increased over 200 percent during the past two decades. This circumstance combined with the realization that many agricultural enterprises are energy intensive have prompted farmers to investigate potential cost saving alternatives. Major economizing efforts have been focused on energy use optimization and substitution of solar, wind, biomass, low-head hydroelectric and geothermal energy for more costly fossil fuels.

The United States Department of Agriculture (USDA, 1980) estimated that 2.29x10 liters (6.06 billion gallons) of gasoline and diesel fuel were consumed for farm production during 1980. Approximately 15 percent (237.8 trillion Btu equivalent) was used for livestock production. Warm confinement livestock and poultry production systems traditionally rely on liquified petroleum (LP) gas as an energy source (Brewer and Dunn, 1975). Brewer (1976) estimated 72 percent of the total energy consumed by the poultry industry was used for commercial broiler brooding, an operation that is essentially continuous throughout the year.

The major alternatives to liquid fuels are ethyl alcohol as a substitute for gasoline and vegetable oil for diesel fuel (Clary 1980). Neither gasoline nor diesel fuel are consumed in large
quantities by the poultry industry. Consequently, other alternatives such as applications of solar and wind energy conversion systems coupled with energy conservation appear to hold greater potential as cost reducing options in confinement poultry and livestock structures. Successful brooding of broilers requires optimum environmental conditions to insure that maximum biological potential is achieved. Optimum temperature ranges for specific growth periods can be estimated from the body thermoneutral range. Sturkie (1965) indicated that the thermoneutral range for a chicken starts at 35 degree C during the first week of age and decreases to 32.5±1.5 degree C by the fifth week. Mature chickens maintain a thermoneutral temperature of 21±3 degree C. Sturkie also reported that rearing broilers in environmental temperatures below the thermoneutral zone causes severe reduction in rate of gain and feed efficiency. The effect of environmental condition being substantially different than the thermoneutral zone was discussed by Timmons et al., (1981). The report stated that whenever environmental temperatures go too far above the thermoneutral zone bird feed intake will be greatly reduced or even stop completely.

High ambient temperatures frequently experienced during summer brooding periods may cause severe heat stress or even death due to hyperthermia. Reduction of broiler house temperatures during warm weather periods has traditionally been attempted with evaporative cooling and misting systems. Mechanical refrigeration systems would allow precise control of environmental conditions, but associated costs remain prohibitive to commercial broiler producers.

Misting systems are cycled on and off by a timer or thermostat to prevent wetting problems of the litter and equipment. This limitation reduces average evaporative efficiency by the on-time percentage as compared to total system running time. Timmons and Baughman (1980) experimentally determined most efficiencies of low
and high pressure misting systems are 15 to 20 percent and 50 percent, respectively. A wet pad evaporative cooling system can achieve 75 percent efficiency or higher depending upon air velocity through the pad. Evaporation cooling systems are restricted to mechanically ventilated buildings and air pressure drop through the pad increases as air velocity increases (Timmons, 1981). These restrictions leave the user with the problem of how to optimize between cooling efficiency, pressure drop and system cost. A warm weather ventilation system that will reliably and economically provide sensibly cooled air to the animal space remains as a distinct need for confinement livestock producers.

Rao and Lee (1982) reported the variation between average daily high and low temperature in Oregon ranged from 5.6 degree C in April to 11.5 degree C in August. Utilization of this diurnal temperature variation could provide part or all of the cooling load requirement in broiler houses located in Pacific Northwest.

McFate (1981) stated that prudent management of warm confinement systems will become one of the most important factors in future energy conservation efforts. Automatic control of poultry house ventilation systems described by Duncan et al. (1975), Muehling (1976) and Bruce (1979) incorporated thermostatically-activated reversible motors to adjust inlet vents and curtains. Performance of these early control systems was dependent upon the timer selected and precision obtainable with a commercially available thermostat.

Leonard and McQuitty (1982) reviewed automation and automatic control as applied to intensive livestock production systems and concluded that a continued reduction in the cost of electronic components make the applications of microprocessors and minicomputers an attractive option. The application of such systems are limited only by the designer's imagination and the availability of adequate sensors and actuators.
Research to provide environmental conditions within confinement livestock and poultry structures that are conducive to maximum production and yet allow the producer to remain cost competitive continue to be of great importance. Energy conservation, application of alternative energy sources and modern management and control systems all hold excellent potential in benefiting this industry. Therefore, this research has focused on the following specific objectives:

1. Design, assemble and test a low cost, flexible, user-oriented microprocessor system to control utilization of a rock bed heat sink for broiler house summer cooling.

2. Develop and test a mathematical model to predict the performance of a rock bed thermal storage module constructed to test the designed control system.

3. Design a simple method that will enable the broiler producer to define and evaluate individual microprocessor control options.
Numerical solutions of the heat and mass transfer processes in rock bed thermal energy storages became obtainable when mainframe and microcomputers were employed to solve the complicated governing equations since Schumann (1929) first analyzed these systems with a two-phase model. Recently, with improved modeling techniques and the invention of sophisticated electronic devices, additional rock bed systems with automatic controllers for agricultural applications have been constructed and tested by Rokeby et al. (1980).

Feasibility of Rock Bed Storage Systems

Thermal energy storage systems are typically designed to accumulate energy when available and release that energy upon demand. Energy may be stored as sensible heat of a solid or liquid, as heat of fusion, or as chemical energy in a reversible chemical reaction. The storage media selected should facilitate energy retrieval and match the demands of the intended operation. Water and packed bed energy storage systems have most frequently been adopted by agricultural enterprises to enhance the efficiency of solar collector heating systems. Meador et al. (1978) achieved 25 percent efficiency in a solar heating study which utilized a water storage tank to provide supplementary heat for baby pigs. The 4.5 cubic meter steel water storage tank was well insulated by 150 mm urethane foam and was buried in a gravel bed to reduce heat loss. Water was recognized as a practical storage medium when used for heat collection and distribution, but Butler and Troeger (1979) stated no need for a heat exchanger and lower initial cost as major advantages of rock bed systems over water systems. Reece (1978) compared air and water systems and concluded that air systems with rock bed storage were simpler, less costly, required less maintenance, were easier to operate, were more efficient and eliminated freezing problems.
Duffie and Beckman (1980) suggested that well designed packed beds using rocks have several desirable characteristics for solar energy applications; high heat transfer coefficient between the air and rock, high degree of temperature stratification within the rock bed, and low energy loss to the environment.

**Rock Bed Storage Applications and Efficiency Analyses**

Agricultural product drying systems that utilize solar energy frequently incorporate rock bed thermal storage. Price et al. (1977) presented experimental results from a passive gravel storage module designed to be assembled inside an existing greenhouse. Air distribution, pressure drop, and energy stored in the packed bed were measured. The study concluded that the designed system had a collector efficiency of approximately 24 percent. Butler (1978) designed a drying system which utilized a well insulated horizontal, crushed rock storage supported by a concrete block plenum. Performance comparisons indicated that the designed solar collector and rock bed storage unit had the ability to provide energy equivalent to a comparable sized conventional electric dryer at lower operational cost.

A wide range of rock bed thermal storage system efficiencies have been reported in the literature. Variations in construction materials used and basic system design may explain some of the inconsistencies in system efficiencies reported, however inconsistencies may also come from low precision data acquisition techniques and equipment used to control the systems. Another source of variation could arise from not accurately accounting for energy accumulated by the collector being used directly by the system without passing through the storage medium. In the later case, efficiency represents overall system performance rather than storage unit efficiency. An experiment that incorporated a rock bed heat
storage system to aid in curing tobacco was conducted by Cundiff (1981). Transient air flow rates were sampled and numerically integrated by a PDP-11 minicomputer to obtain an hourly flow rate. An average solar to thermal conversion efficiency of 85 percent was reported. The research also indicated that a higher system performance could be obtained if sophisticated control and sampling techniques were employed.

Rokeby et al. (1980) investigated the performance of a large rock bed solar heat storage system. The storage contained 180 tonnes of crushed rock designed to supply two days' heating requirement for a broilers confinement building. Control and data sampling were supported by a remote controlled data acquisition system. Integrated measurements of air flow were used to determine energy exchange in various parts of the system. Approximately 80 percent of the solar heat delivered came from the storage unit while the other 20 percent came directly from the collector to the end application. Results of the research showed that diurnal temperature changes have a positive effect on overall efficiency. The rock bed heat storage unit enhanced system efficiency to approximately 100 percent when used to heat outdoor air.

Thermal decay prediction within a rock bed system is an important factor when considering long term applications. Kreith and Kreider (1978) suggested the overall heat transfer coefficient (U value) should be approximately 0.25W/m²°C for liquid storage tanks. The U value recommended for rock bed storages used to provide space heating was twice that of liquid systems. Walton et al. (1979) substantiated the benefit of insulation in a rock bed storage unit. Insulated and uninsulated rock beds were constructed and applied with identical conditions. Figures of isothermal lines and temperature
profiles were drawn and presented for later design reference.

Average daily rock bed temperature was estimated with the following equation:

\[
\sum_{n=1}^{6} 24 U_n A_n (T_R - T_{sn}) = m_a C_a t_c (T_c - T_R)
\]  

(1)

Where

- \(U_n\) = overall heat transfer coefficient through each plane boundary surface of the rock bed, \(W/m^2{\degree}C\)
- \(A_n\) = area of each plane surface, \(m^2\)
- \(T_R\) = average daily rock bed temperature at full charge, \(\degree C\)
- \(T_{sn}\) = average temperature of media adjacent to the surface of each plane boundary of the rock bed, \(\degree C\)
- \(m_a\) = volume air flow rate through collector, \(m^3/sec\)
- \(C_a\) = specific heat of air, \(J/kg{\degree}C\)
- \(\rho_a\) = density of air, \(kg/m^3\)
- \(t_c\) = period of heat collection, \(s\)
- \(T_c\) = average collector outlet temperature during \(t_c\), \(\degree C\)

The equation used to describe the energy balance over a small time increment for the rock bed was:

\[
-C_k \rho_k V dT_R = \sum_{n=1}^{6} U_n A_n (T_R - T_{R_k}) dt
\]  

(2)

where

- \(C_k\) = specific heat of rock, \(J/kg{\degree}C\)
- \(\rho_k\) = bulk density of rock, \(kg/m^3\)
- \(V\) = volume of the rock bed, \(m^3\)
\( T_n \) = average daily rock bed temperature, °C
\( dT_n \) = change of rock bed temperature during dt, °C
\( T_{ke} \) = equilibrium temperature of rock bed, °C
\( t \) = time, h

Integrating Equation 2 and applying an initial condition that at \( t=0 \), the average rock bed temperature is equal to the initial rock bed temperature, Walton et al. (1979) stated the relation between the average temperature and equilibrium temperature of the rock bed as:

\[
\frac{T_n - T_{ke}}{T_{ro} - T_{ke}} = \exp\left( \frac{\sum A_n}{C_{A} p_k V} t \right)
\]

(3)

where

\( T_{ro} \) = initial rock bed temperature

Analysis of Equation 2 indicates that a lower thermal decay can be obtained by minimizing the ratio of rock bed surface area to total rock bed volume. Calculated results show that a large rock bed will have a slower thermal decay than a small rock bed of the same shape. Walton et al. (1979) also suggested insulation with a heat transfer resistance value of R-19 was required to reduce the heat losses through the plenums. Bodman et al. (1980) evaluated in floor storage of solar heat for pigs and suggested a R-3 level of underlying insulation is required to define a storage system.

Prior to 1982 the direction of charging and discharging rock bed thermal storages was considered as an important factor in achieving a high degree of stratification within the bed. High temperatures at the bottom of a rock bed theoretically add free convection effects to a system charged with upward flowing air. Calderwood (1979)
constructed a set of solar rice-drying test facilities which included five dryers and a rock bed heat sink. Upward air movement was used in charging and discharging the system. Overall collection efficiency during eight charging and discharging periods ranged from 29 to 40 percent. Sullivan et al. (1982) presented experimental measurements of the destratification process in a rock bed. A container with cross-sectional measurements of 1.8 meter by 1.8 meter and a depth of 1.8 meter was filled with river gravel. Temperature profiles during charging and reversed charging processes were monitored with nine thermocouples. Equal temperature stratification was maintained regardless of air flow direction which indicated that the system was insensitive to charging direction.

**Energy Balance**

Temperature profiles within a rock bed can be considered as a function of elapsed time and distance from the inlet, since the heat front at a specific location will advance toward the next location during a specified period of time. Rock bed thermal storage performance is frequently described by one of two general types of models. One-phase models are used if the system is considered as a homogeneous medium. Theoretical two-phase models assume heat and mass exchange proceed in both the solid and fluid. Two-phase models form a coupled reaction which more realistically describe the interaction between the air and rock. The heat transfer process within a packed bed system was first analyzed by Anzelius (1926). Schumann (1929) formulated the following energy balance equation based on a two phase model:

**Fluid:**

$$
\Lambda \rho \epsilon \frac{\partial T_F}{\partial \theta} = -\dot{m} C_f \frac{\partial T_f}{\partial x} + h_v A(T_s - T_f)
$$

(4)
Solid:

\[ A \rho_f C_f (1 - \varepsilon) \frac{dT_f}{d\theta} = h_v A (T_f - T_s) \]  

where

\[ A = \text{cross-sectional area of packed bed, m}^2 \]
\[ \rho_f = \text{density of fluid, kg/m}^3 \]
\[ \varepsilon = \text{void fraction} \]
\[ T_f = \text{fluid temperature, °C} \]
\[ \theta = \text{time, h} \]
\[ m_f = \text{mass flow rate of fluid, kg/h} \]
\[ C_f = \text{specific heat of fluid, J/kg·°C} \]
\[ X = \text{distance to inlet, m} \]
\[ h_v = \text{volumetric heat transfer coefficient, J/m}^3·°C·h \]
\[ T_s = \text{temperature of solid, °C} \]
\[ \rho_s = \text{density of solid, kg/m}^3 \]
\[ C_s = \text{specific heat of solid, J/kg·°C} \]

Assumptions required to derive the above equations included:

1. Infinite radial thermal conductivity of the solid
2. Constant properties over the entire system
3. No heat lost to the environment
4. No fluid phase axial heat dispersion or conduction
5. Zero solid phase axial thermal conductivity

Schumann's model has been extended by the derivation of numerous mathematical models to describe packed bed performance. Rosen et al. (1966) introduced a bed length parameter to measure the relative
importance of internal particle conduction. This parameter, also known as effective bed length, was defined by the ratio of a transfer parameter \( Y \) (a function of bed length, void fraction and Stanton Number) to the particle Biot Number \( (Bi) \). Jefferson (1972) showed that the Biot number provided the sole criterion for estimating the relative importance of internal solids conduction within a packed bed. Babcock (1966) reported that if the thermal capacity and axial conduction of air during system operation are significant, the energy balance between fluid and solid can be described by following equations:

**Fluid:**

\[
\begin{align*}
\frac{\epsilon \rho_f C_f}{\rho_f C_f} \frac{dT_f}{dt} &= \frac{-m_f C_f}{\rho_f C_f} \frac{dT_f}{dx} + h_a(T_s - T_f) + \epsilon \lambda_f \frac{\partial^2 T_f}{\partial x^2} \\
\end{align*}
\]

(6)

**Solid:**

\[
(1 - \epsilon) \rho_s C_s \frac{dT_s}{dt} = \frac{h_a(T_f - T_s)}{\rho_s C_s} (1 - \epsilon) \lambda_s \frac{\partial^2 T_s}{\partial x^2}
\]

(7)

where

- \( T_s \) = temperature of solid, \(^\circ\)C
- \( T_f \) = temperature of fluid, \(^\circ\)C
- \( \epsilon \) = void fraction
- \( C_f \) = specific heat of the fluid, \( J/kg\cdot{^\circ}C \)
- \( C_s \) = specific heat of the solid, \( J/kg\cdot{^\circ}C \)
- \( \rho_s \) = density of solid, \( kg/m^3 \)
- \( \rho_f \) = density of fluid, \( kg/m^3 \)
- \( m_f \) = mass flow rate of fluid, \( kg/s \)
- \( t \) = time, hr
\( h_a \) = heat transfer coefficient related to average particle temperature, \( J/hr{\circ}C-m^2 \)

\( \lambda_s \) = thermal conductivity of solid phase, \( W/m{\circ}C \)

\( \lambda_f \) = thermal conductivity of fluid phase, \( W/m{\circ}C \)

\( X \) = distance to inlet, m

Thermal energy dispersion in a packed bed was treated as the sum of individual dispersive mechanisms which occurred within the packed bed. Based on the results of a study by Yagi et al. (1960), Handley and Heggs (1969) suggested that internal particle conduction effects could be ignored when bed length parameter was greater than 60. Research indicated that when a system experiences a flow rate increase, axial thermal conductivity shows a higher increase rate than radial thermal conductivity. The thermal gradients within each cross section will tend to approach a homogeneous state as the air flow rate decreases within a system below a threshold level. Littman et al. (1970) analyzed the frequency response of a packed bed and simplified Babcock's model for systems with small Reynolds Numbers. The modified Littman model was formed by eliminating the last term in Equation 6. An equivalent model was introduced by Riaz (1977), who converted a two-phase model into a one-phase model. Thermal capacity and axial conduction terms in the fluid phase equation were eliminated and axial conduction in the rock was ignored. A characteristic distance \( x_{N2} \) and a characteristic time \( t_{N2} \) were introduced to modify Schumann's model as:

Air:

\[
\rho_a C_a V_a \frac{\partial T_a}{\partial \xi} = h_a (T_a - T_0)
\]  

(8)

Rock:
\[ \rho_k C_k \frac{\partial T_k}{\partial \tau} = h_v (T_a - T_k) \]  

(9)

where:

\( \rho_k \) = density of rock, Kg/m^3

\( C_k \) = specific heat of rock, J/Kg°C

\( V_a \) = velocity of air flow, m/s

\( T_a \) = air temperature, °C

\( \epsilon \) = distance to inlet, m

\( h_v \) = volumetric heat transfer coefficient, kJ/h°C·m^3

\( T_k \) = temperature of rock, °C

A single phase conductivity model was then derived by letting the volumetric heat transfer coefficient \( h_v \) tend to infinity and assuming that \( a C_a \ll b C_b \) and \( K_a \ll K_b \). The simplified equations became:

\[ \frac{\partial T}{\partial \tau} + \frac{V}{\epsilon} \frac{\partial T}{\partial \epsilon} = \delta \frac{\partial^2 T}{\partial \epsilon^2} \]  

(10)

where

\( V = V_a \rho_a C_a / (\rho_b C_b) \), m/s  

(11)

\( \delta = K_k / \rho_k C_k \), m^2/s  

(12)

\( K_k \) = thermal conductivity of rock, W/m°C

\( t \) = time, s

\( \epsilon \) = distance to inlet, m
Riaz (1977) stated that the characteristic parameters of a combined heat transfer and thermal conductivity model of a packed bed are the sum of individual characteristic quantities. The heat transfer coefficient and effective thermal conductivity in the single-phase model were defined by Equations 13 and 14 as:

\[
\frac{1}{h_{eq}} = \frac{1}{h_\nu} + \frac{k_s}{(V_o C_o p_o)^2} \tag{13}
\]

\[
k_{eq} = k_s + \frac{(V_o C_o p_o)^2}{h_\nu} \tag{14}
\]

where

\[h_{eq} = \text{modified heat transfer coefficient, } h^{-\circ C \cdot m^3/kJ}\]

\[k_{eq} = \text{effective thermal conductivity, } W/m^{-\circ C}\]

Five different input conditions were considered separately to obtain an analytical solution of the simplified single-phase model. Final results were presented in the form of generalized plots portraying the space-time distribution of temperature inside the packed bed and provided a usable reference for packed bed design.

Performance analysis of a rock bed can be achieved by mathematical modeling. However, most rock bed systems are designed for one specific application under variable conditions. This leads to a lack of generalization and increases the degree of difficulty in obtaining an appropriate rock bed system when designed using a general model. Whitaker (1972) presented the first functional dimensionless analysis for packed bed systems. The study indicated
that if all physical properties were evaluated at the same reference temperature, the functional dependence of the dimensionless temperature within the packed bed could be specified by:

\[ \Theta = F((\theta, X, Y, Z), (R_e, Pr, \Lambda), N_{bc}) \]  \hspace{1cm} (15)

where

\[ \Theta = \text{dimensionless temperature} \]
\[ \theta, X, Y, Z = \text{location} \]
\[ R_e = \text{Reynolds number} \]
\[ Pr = \text{Prandtl number} \]
\[ \Lambda = \text{dimensionless viscosity} \]
\[ N_{bc} = \text{dimensionless parameters in boundary conditions} \]

Experimental data of packed bed heat transfer analyses from Gamson (1943), Glaser (1958), McConnachie (1963), Taecker (1949), and Wilke (1945) were collected by Whitaker (1972) and processed by dimensionless analysis. The results, which included a correlation function between Reynolds Number, Prandtl Number and Nusselt Number, with minor variations for a wide range of packing materials and different bed arrangements, were illustrated as:

\[ N_u = \left(0.5R_e^{0.5} + 0.2R_e^{0.3}\right) \times Pr^{0.5} \left(\frac{\mu_o}{\mu} \right)^{0.14} \]  \hspace{1cm} (16)

where

\[ N_u = \text{Nusselt number} \]
\(\mu_b =\) fluid viscosity at bulk temperature, Pa-s
\(\mu_w =\) fluid viscosity at wall or surface, Pa-s

Whitaker's report shows that a single correlation can be obtained to predict the heat transfer rate in a randomly packed bed by properly choosing the characteristic length and velocity.

Simulation of Packed Bed Performance

Analytical solutions for two-phase coupled equations involving non-linear inputs are complicated and difficult to obtain. Simplifying two-phase models into one-phase models is a feasible way to obtain solutions if appropriate assumptions are applied. Vortmeyer and Schaefer (1974) presented a one-phase model adapted from Littman's model and assumed the solid and fluid temperatures were not equal. An effective axial thermal conductivity combining the heat transfer coefficients in two-phase models was introduced. The result showed that a two-phase model can be described as:

\[
(1-\epsilon)\rho_s C_s \frac{\partial \theta}{\partial t} = \left( \lambda_s + \frac{m_f C_f}{h_{rt}} \right) \frac{\partial^2 \theta}{\partial x^2} - m_f C_f \frac{\partial \theta}{\partial x} \tag{17}
\]

where

\(\epsilon =\) void fraction
\(\rho_s =\) density of solid, \(kg/m^3\)
\(C_s =\) specific heat of solid, \(J/kg-^\circ C\)
\(\theta =\) temperature of solid, \(^\circ C\)
\(t =\) time, s
\(\lambda_s =\) effective thermal conductivity of stagnant bed, \(W/m-^\circ C\)
\(m_f =\) mass flow rate of fluid, \(kg/m^2-s\)
\(C_f\) = specific heat of fluid, \(J/kg\cdot{}^\circ{}C\)

\(h\) = heat transfer coefficient, \(kJ/h\cdot{}^\circ{}C\cdot{}m^2\)

\(a\) = particle surface area per unit bed volume, \(m^2/m^3\)

\(x\) = distance, m

Prediction of the long term response of a rock bed to complicated non-linear weather data can be achieved by careful examination and simplification of the system parameters instead of simplifying two-phase models into single-phase models. A long term thermal storage model for packed bed solar air heating and cooling systems was designed by Hughes et al. (1976). Hughes assumed the thermal capacitance of air was zero and included a modified number of transfer units (NTU) suggested by Jefferson (1972) to account for the temperature gradient in the gravel. The energy balance equations used in the model were:

**Fluid:**

\[
\frac{T_f}{(X/L)} = NTU(T_s-T_f) + \frac{UPL}{m_fC_f}(T_{env}-T_f)
\]  

(18)

**Solid:**

\[
\frac{\delta T_s}{\delta(\theta/\tau)} = NTU(T_f-T_s)
\]  

(19)

where

\(T_f\) = temperature of fluid, \(^\circ{}C\)

\(X\) = position along the bed, m

\(L\) = length of packed bed, m

\(NTU = h_rAL/mC_f\)  

(20)
$T_s$ = temperature of solid, °C

$U$ = energy loss coefficient, $kJ/ hr^{-°C} - m^3$

$P$ = perimeter of packed bed, m

$T_{env}$ = ambient temperature, °C

$m_f$ = mass flow rate of fluid, kg/h

$C_f$ = specific heat of fluid, $J/kg^{-°C}$

$\tau = \rho_s C_s (1-\epsilon) Al/m_f C_f$, h

$\rho_s$ = density of solid, $kg/m^3$

$C_s$ = specific heat of solid, $J/kg^{-°C}$

$\epsilon$ = void fraction

$A = $ cross-sectional area of packed bed, $m^2$

$$NTU_c = \frac{D}{L(P_s) + \frac{1+B_s/5}{NTU}}$$ (21)

The research also indicated the assumption of no mass transfer was acceptable since the saturated water content of gravel is only 0.03 Kg/Kg. Hughes et al. (1976) concluded that the performance of a packed bed thermal storage system with $NTU_c$ greater than 10 is insensitive to values of $NTU_c$. Consequently, a simple model of a packed bed based on an infinite value of $NTU_c$, is adequate for general purpose rock bed design.

The physical model to derive analytical solutions was based on the assumption of infinite packed bed length. This imposes complexity and limitations to analytical solutions of energy balance equations as shown by Riaz (1977). A different concept must be used when deriving the numerical solution, since numerical solutions treat the packed bed as a sum of finite layers. Numerical analysis techniques became the most effective tool as the availability of the digital computers increased. Baird et al. (1976) presented a
A numerical procedure for calculating temperature distribution in bulk loaded fruits and vegetables. The physical properties of bulk loaded fruits are similar to a packed bed, thus the governing equations can be written as:

\[
\frac{\delta T}{\delta t} = \alpha \left( \frac{\delta^2 T}{\delta r^2} + \frac{2T}{r} \right) \tag{22}
\]

\[
T(r,0) = T_i \tag{23}
\]

\[
\left( k \frac{\delta T}{\delta t} \right)_{r=R} = h(T_s - T_\infty) \tag{24}
\]

where

\( T \) = temperature, °C

\( t \) = time, hr

\( \alpha \) = thermal diffusivity, \( m^2/s \)

\( r \) = radial distance from center, \( m \)

\( k \) = thermal conductivity, \( W/m \cdot °C \)

\( R \) = radius of sphere, \( m \)

\( h \) = convective heat transfer coefficient, \( W/m^2 \cdot °C \)

\( T_s \) = temperature of surface, °C

\( T_\infty \) = ambient temperature, °C

Fourier's law was then applied to the interior, surface, and center nodes located at a circular plane that represented the cross section of a vegetable or a fruit. Results showed the accuracy of a numerical solution for bulk loads depended upon the number of nodes and time increments used. The same method was modified when Eshleman
et al. (1977) developed a numerical model of the energy balance within a packed bed. Eshleman modeled the system and described the solutions as:

**Air:**

\[
\frac{\Delta T_a}{\Delta t} + V_i \frac{\Delta T_a}{\Delta x} = - \frac{h A'}{f \rho_a C_a} (T_a - T_s)
\]  

(25)

**Rock:**

\[
\frac{\Delta T_s}{\Delta t} = \frac{h A'}{(1 - \varepsilon) \rho_s C_s} (T_a - T_s)
\]  

(26)

where

- \( T_a \) = temperature of air, °C
- \( t \) = time increment, s
- \( v_i \) = interstitial velocity, m/s
- \( x \) = distance increment, m
- \( h \) = heat transfer coefficient, \( W/m^2\cdot°C \)
- \( A' \) = ratio of total surface area to bed volume, \( m^2/m^3 \)
- \( \varepsilon \) = void fraction
- \( \rho_a \) = density of air, \( kg/m^3 \)
- \( C_a \) = specific heat of air, \( J/kg\cdot°C \)
- \( T_s \) = temperature of rock, °C
- \( \rho_s \) = density of rock, \( kg/m^3 \)
- \( C_s \) = specific heat of rock, \( J/kg\cdot°C \)
Once a change in rock temperature, $DT_s$, has been calculated over a certain period of time, $Dt$, (from Equation 26), air temperature change, $DT_a$, can be calculated from Equation 25. Inlet temperature variation did not affect solution stability if appropriate time increments were chosen. Eshleman also claimed the use of a single lumped-capacity model (Equation 25) saved computer time by a factor of up to ten. One disadvantage of this explicit solution method is that stability corresponds to step size of the changing variable. Large increment sizes can save computer time but may result in a divergent solution. Stability is, therefore, an essential criteria that must be considered prior to adaptation of the explicit method.

The explicit finite difference method utilizes only part of the adjacent nodes' present value to predict the system's response at the next time step. The effect of earlier time steps are neglected. Implicit finite difference methods improve calculation accuracy by introducing numerical approximations evaluated at an advanced point in time. This method provides solutions with simple and fast convergence criteria which greatly decrease required memory size and computer run time.

**Broiler House Ventilation**

The United States' broiler industry has gained an important role in providing food for human consumption in the developed, free world by utilizing modern efficient production systems. Adapting technology that both provides an environment that tends to maximize the physiological responses of broilers plus heeds the grower's energy budget limitations prompted Morrison (1982) to use that industry as the example of the most advanced use of man's knowledge in food processing animals and serve as a standard for comparison of others. Temperature, wind speed, humidity, radiation, and other climatological variables can easily be measured. Assessment of the
thermal status of livestock when exposed to different combinations of these variables has proven to be much more difficult (McArthur, 1982). Modern broiler production commonly utilizes housing to provide a comfortable artificial environment that maintains the micro-climate around the birds with relatively little variation. Steady improvements in housing have incorporated better controllers, construction techniques, and building materials. All have been essential keys to maximizing rates of gain, feed conversion efficiencies, and reducing labor inputs. Effective system design must integrate house configuration, insulation levels, heat and moisture balances, lighting and other operational equipment with the biological effects of the broilers to meet ventilation demands that continuously vary. The main variation in instantaneous heat gain of a ventilated broiler house is caused by changes in incident solar radiation. Design methods that incorporate transient heat transfer analyses must be used to account for this unstable factor. The existence of non-linear, time dependent surface boundary conditions made most ventilation models difficult to solve. A transformation method which incorporates the transfer function approach to simplify and reduce the time required for computation was introduced by ASHRAE Handbook of Fundamental (1977). Similar research was conducted by Albright (1982) on a steady periodic thermal analysis for livestock housing by using a complex form Fourier series. The procedure assumed each day was embedded in a series of days which permitted the use of harmonic analysis. A system of linear equations for the temperature analysis can then be constructed and solved by a computer program. Albright indicated that the method could provide ±1 oC accuracy in predicting air temperature within a livestock building if ventilation rate was not a function of time. In most broiler house operations, ventilation rates vary with time which decreases the
validity of Albright’s method in predicting the ventilation requirements. Even so, Albright’s method still provided a good way to study effects of building material and construction changes.

**Control Configuration**

Significant energy waste in most ventilation systems comes from inadequate use or operation of controllers. Adequate control can be achieved with a well designed electromechanical controller, however, once the controller has been installed the device is unable to be adjusted to fit later modifications in the system. Most conventional electromechanical controllers utilize solid state devices offering a wide range of applications but lack accuracy necessary to meet specific system requirements. Conventional ventilation control in livestock and poultry confinement systems typically incorporate a series of thermostats that activate one or more fans as inside air temperature increases past a predetermined point. When the temperature decreases to the set point of the thermostat the fan is deactivated. Cole (1980) analyzed animal housing ventilation systems by applying control theory. As categorized by control theory, animal environmental control systems (AECS) are functionally like regulators. When disturbance inputs change, outputs are maintained within a relatively narrow range by referencing to a predetermined point. Parameters involved in the control were categorized as plant, control forces, disturbance inputs, and controlled variables. A basic control equation was set up to illustrate the performance required by users. Additional variables were added as further improvements. The increase of variables led to the need for more comprehensive control and more sophisticated controllers.

The invention of microprocessors combined with electronics has opened a new era of data manipulation and instrumentation. Microprocessor applications in agriculture have been rapidly
increasing especially in the control and monitoring of machinery, processing operations and data acquisition systems. The increase has evolved through a combination of sophisticated design requirements and the increasing availability of low cost computer and electronic hardware. Proper use of microprocessors can result in faster and more accurate control. Also, microprocessors allow control of complicated systems which would be otherwise impractical to handle.

Flexibility is an important key to increased performance of mechanical systems. Microprocessors provide the build-in intelligence which engineering systems must have to respond with speed, accuracy, and efficiency to diverse and changing demands. Microprocessors are inherently flexible and facilitate modification by being programmable and can be upgraded at minimum cost. Microprocessor or mini-computer based control systems allow complicated control strategies simply by changing software. Parsons et al. (1980) adapted a sophisticated system that utilized an Altair 8080A microcomputer to manage the indoor environment control of a plant growth chamber. Mitchell and Drury (1982) utilized a microcomputer and conventional solid state controllers to control a solar heating system. The algorithm used to monitor and control the system was written in BASIC. Mitchell and Drury concluded the advantages of the adopted system were easy change of control strategy, simpler hardware hookup, and reduced variance from the set point. The specific control system used by Mitchell and Drury delivered high performance but introduced the primary disadvantage of high cost. A complete review of automation and automatic control applied to intensive animal production was presented by Leonard and McQuitty (1982). Leonard and McQuitty concluded that the increasing cost of energy, labor, and feed, together with the increasing scale
of farm operations, provide high incentive to further development of optimize automation, automatic control systems, and control strategies.
NUMERICAL MODEL AND SIMULATION OF ROCK BED PERFORMANCE

Energy Balance Formulation

Analytical solutions of the governing partial differential equations used to describe dynamic systems are difficult to obtain due to nonuniformly-distributed parameters that represent the medium properties, irregular boundaries and nonlinear boundary conditions.

All systems that exist in a time-space continuum can be broadly grouped into lumped systems or distributed parameter systems. In a lumped system, the process of discretization is applied to the space variables and time is kept in continuous form. Rock bed thermal storage systems possess continuity in both the time and space domains therefore can be treated as a distributed parameter system for purposes of mathematical model construction. The intrinsic properties of the rock bed medium (gravel) are defined continuously and uniformly while the boundary conditions are likely to experience abrupt changes and provide nonlinearity to the system.

Vemuri and Karplus (1981) examined the cause and effect relationship between excitation (inputs) and response (outputs) and rearranged the existing physical system as, design identification synthesis, direct analysis, and control instrumentation. Direct analysis finds responses for given excitation and system characteristics. A design identification synthesis system uses known excitations and responses to find the system characteristics. A controlled instrumentation system uses known system characteristics and responses to determine what type of excitation is needed. An air-rock simulation model that predicts rock bed outlet temperatures under various input conditions can be categorized as a direct analysis system. Complicated three dimensional descriptions are necessary to fully define actual operation of a rock bed storage unit.
that involves dynamic responses to nonlinear boundary conditions and nonhomogeneous medium properties. Factors that can be controlled or further simplified by the designer are chosen as the assumptions of the rock bed simulation model. The assumptions include:

1. incoming air is in plug flow
2. physical properties of air and rock stay constant
3. vertical thermal gradient within rock bed is negligible
4. thermal radiation effects can be neglected
5. radial direction heat transfer does not exist
6. no heat losses to the environment occur
7. no internal heat generation occurs
8. mass transfer is negligible

The first step in an energy balance analysis of a rock bed is to apply parameter uniformity; i.e., the gravel properties were assumed to be homogeneous, as stated in assumption two. Void fraction, defined as the ratio of air volume to rock bed volume, directly affects the amount of energy that can be stored in the system. By considering a unit control volume with cross sectional area $A_c$ and depth $D_x$, a simple energy balance can be applied where the energy output and energy stored in the control volume are equal to the energy input to the system during a heat exchange cycle. The heat exchange process occurs in both the air phase and rock phase plus interface boundaries such as the rock bed inlet and outlet. Energy balance evaluations are developed for conductive and convective heat transfer as well as internal energy change. The air and rock phase energy analyses were performed as follows:

Air system

(1) Internal energy change
The internal energy change rate that takes place within a unit length through an advance in the time step is:

\[ \Delta E_{ia} = \rho_a \Delta T_o A_c / \Delta t \]  

(27)

where

\[ \rho_a = \text{air density, kg/m}^3 \]
\[ C_a = \text{specific heat of air, J/kg} \cdot \text{°C} \]
\[ A_c = \text{cross-sectional area of rock bed, m}^2 \]
\[ \varepsilon = \text{void fraction} \]
\[ \Delta T_o = \text{air temperature change during t, °C} \]
\[ \Delta t = \text{time increment, s} \]

(2) Conductive heat transfer

The conductive heat exchange caused by the mass rate of air flow per unit cross sectional area per unit period of time is:

\[ \Delta E_{ac} = m A_c C_a T_o / \Delta x \]

(28)

where

\[ m = \text{mass flow rate of air per unit cross sectional area per unit of time, kg/s} \cdot \text{m}^2 \]
\[ \Delta x = \text{distance increment, m} \]

(3) Convective heat transfer

Convective heat transfer in the air phase system is calculated by

\[ \Delta E_{avo} = -h_v A_c (T_o - T_k) \]

(29)
where

\[ h_v = \text{volumetric convective heat transfer coefficient, } J/m^3{\cdot}^\circ C{-}h \]

\[ T_k = \text{point measurement of rock temperature, } ^\circ C \]

Rock System

Evaluation of the rock system energy balance requires analyses similar to those applied to the air system. Response of the boundaries to the phase interface is incorporated in the convective heat transfer term.

(1) Internal energy change

The time rate change of internal energy within a representative rock element is:

\[ \Delta E_{ri} = \rho_r C_r A_r (1 - \varepsilon) T_k / \Delta t \]  \hspace{1cm} (30)

where

\[ \rho_r = \text{rock density, } kg/m^3 \]

\[ C_r = \text{specific heat of rock, } J/kg{\cdot}^\circ C \]

\[ T_k = \text{rock temperature change during } t, ^\circ C \]

(2) Conductive heat transfer

Conductive heat transfer caused by the temperature gradient between each representative unit rock element is closely related to the rock surface contact area as:

\[ \Delta E_{rc} = A_r K_r (T_k / \Delta x) \]  \hspace{1cm} (31)

where
\( K_k \) = thermal conductivity of rock, \( \text{W/m}^{-\circ} \text{k} \)

\( T_k \) = rock temperature change in unit length of bed, \( ^\circ \text{C} \)

(3) Convective heat transfer

The amount of energy exchange caused by convective heat transfer between the rock and air systems is:

\[
\Delta E_k C_v = h_v A_e (T_a - T_k)
\]  

(32)

The governing partial differential equations that describe the overall energy balance across the entire rock bed are obtained through the following procedures.

The equilibrium state of the air system and the rock system within any time period can be described by:

Air:

\[
\frac{\partial T_a}{\partial t} + \frac{m C_a \partial T_a}{\rho_a C_a \partial x} + \frac{h_v}{\rho_a C_a \epsilon} (T_a - T_k) = 0
\]  

(33)

Let \( G = \frac{m C_a}{\rho_a C_a} \) and \( H = \frac{h_v}{\rho_a C_a \epsilon} \)

the above equation can be written as following

\[
\frac{\partial T_a}{\partial t} + G \frac{\partial T_a}{\partial x} + H (T_a - T_k) = 0
\]  

(34)

Rock:

\[
\frac{K_k}{\rho_k C_k (1 - \epsilon)} \frac{\partial^2 T_R}{\partial x^2} + \frac{h_v (T_a - T_k) - K_k}{\rho_k C_k (1 - \epsilon)} = 0
\]  

(35)

Let \( P = \frac{K_k}{\rho_k C_k (1 - \epsilon)} \) and \( \theta = \frac{h_v}{\rho R C_k (1 - \epsilon)} \) and re-write the above equation as
\[
\frac{\partial T_p}{\partial t} - P \frac{\partial^2 T_p}{\partial x^2} - \theta(T_o - T_p) = 0 \tag{36}
\]

The equilibrium state defined by the initial condition changes when boundary excitations take place. Non-zero temperature fluxes applied to the inlet during subsequent time steps initiates the rock bed system thermal process. The energy gradient normal to the inlet plane must be directly proportional to the fluxes provide that the preceding assumptions accurately represent the physical condition of the system. Initial conditions of the rock bed simulation model are:

\[
T_o(x,0) = T_{ol}(x) = \text{constant} \tag{37}
\]

\[
T_p(x,0) = T_{pl}(x) = \text{constant} \tag{38}
\]

where

\[T_{ol} = \text{air temperature before new stage occurs, } ^\circ C\]

\[T_{pl} = \text{rock temperature before new stage occurs, } ^\circ C\]

The Dirichlet boundary condition which provides the driving force to the system is:

\[T_o(0,t) = T_w \tag{39}\]

The air-rock system involves a time-space continuum model that considers the input and output as an infinite number of measurement points in either a time or space domain. The discretization of space variables describes the interconnection of two-terminal elements. The internal thermal processes within an individual two-terminal rock bed element are considered to be negligible. Selection of the two-terminal concept enabled use of a finite difference method to solve the partial differential equations in the rock bed simulation.
**Finite Difference Approach**

The aim of numerical solutions is to reduce a given problem into a discrete mathematical model suitable for solution on a digital computer. The rudiments of finite difference methods developed by mathematicians are straightforward to implement when applied to an air-rock thermal system. The basic objective of a finite difference method is to approximate the time-space domain by a set of spaced points. A set of nodal points is superimposed on the variable field and the governing partial differential equations are simplified into a group of algebraic equations. Approximation for the variables and derivatives are then presented in terms of function values at each node. Analysis was first taken on the rock system governing parabolic equation as shown in Equation 35. Gerald (1980) suggested the following equations could be used to describe the finite difference approximation to parabolic equations:

\[
\frac{\partial^2 T}{\partial x^2 (x=x_i, t=t_j)} = \frac{T_{i+1}^j - 2T_i^j + T_{i-1}^j}{(\Delta x)^2} + ERR(\Delta x)^2 \tag{40}
\]

\[
\frac{\partial T}{\partial t (x=x_i, t=t_j)} = \frac{T_{i+1}^{j+1} - T_i^j}{\Delta t} + ERR(\Delta t) \tag{41}
\]

where

- \(i\) = space index
- \(j\) = time index
- \(\Delta x\) = discrete space increment
- \(t\) = discrete time increment
- \(ERR\) = associated error
The error terms in Equation 40 and 41 have a different order due to the forward difference scheme used in Equation 41. The inconsistency introduced some calculation limitations and unstable solutions. Step size of the independent variables must be chosen carefully to assure accuracy which greatly increases computer processing time. The modified Crank-Nicolson method eliminates this disadvantage by considering Equation (41) as corresponding to the mid-point of a calculated time step. Figure 1 shows the computational stencil diagram which applies the central difference to the mid-point of time and the averaged difference quotients at each space increment.

The air system equations are formulated as:

Air System:

\[
\frac{\partial T_a}{\partial t_{(x-x_i, t+i)}} = \frac{(T_{a_{i+,1}} - T_{a_{i-1,1}})}{\Delta t/2} \tag{42}
\]

\[
\frac{\partial T_a}{\partial x_{(x-x_i, t+i)}} = \frac{1}{2} \left( \frac{T_{a_{i+,1}} - T_{r_{i+,1}} + T_{r_{i+,1}} - T_{r_{i-,1}}}{\Delta x} \right) \tag{43}
\]

Plug equations 42 and 43 into equation 34 and let

\[
C_1 = \frac{h_v}{\rho_a \epsilon_a} \quad \text{and} \quad C_2 = \frac{mC_a}{\rho_a \epsilon_a}
\]

The air system equations become

\[
C_1(T_{a_{i+,1}} - T_{r_{i+,1}}) + C_2 \left( \frac{T_{a_{i+,1}} - T_{a_{i-,1}}}{2\Delta x} \right) + \frac{1}{2} \left( \frac{T_{a_{i+,1}} - T_{a_{i-,1}}}{\Delta t/2} \right) \tag{44}
\]

\[
\left( C_1 + \frac{1}{\Delta t} \right) T_{a_{i+,1}} - C_1 T_{r_{i+,1}} + \frac{C_2}{2\Delta x} T_{a_{i+,1}} - \frac{C_2}{2\Delta x} T_{a_{i-,1}} - \frac{T_{a_{i+,1}}}{\Delta t} = 0 \tag{45}
\]

Rock system:

Apply Equations 42 and 43 to the rock system and let
\[ B_1 = k_r / \rho_r C_r (1 - \epsilon) \quad \text{and} \quad B_2 = h_u / \rho_r C_r (1 - \epsilon) \] yield

\[ B_1 \left( \frac{T_{r_{i-1,j}} - 2T_{r_{i,j}}}{(\Delta x)^2} \right) + B_1 ERR + B_2 (T_{a_{i,j}} - T_{r_{i,j}}) = \frac{T_{r_{i+1,j}} - T_{r_{i,j}}}{\Delta t} \] (46)

Rearrange Equation 46 and obtain

\[ B_1 \frac{\Delta t}{(\Delta x)^2} (T_{r_{i-1,j}} - 2T_{r_{i,j}} + T_{r_{i+1,j}}) + B_1 \Delta t (ERR) + B_2 \Delta t T_{a_{i,j}} + T_{r_{i,j}} = (1 + B_2 \Delta t) T_{r_{i,j}} \] (47)

which can be simplified as

\[ \frac{B_1 \Delta t}{(\Delta x)^2 (1 + B_2 \Delta t)} (T_{r_{i-1,j}} - 2T_{r_{i,j}} + T_{r_{i+1,j}}) + ERR (\Delta x)^2 + B_2 \Delta t T_{a_{i,j}} + T_{r_{i,j}} = T_{r_{i,j}} \] (48)

The first term in Equation 47 represents the total possible error that can be generated with the numerical approximation. By inserting the common gravel properties in Equation 47, the error term becomes small and can be omitted in the first calculation. Equations 47 and 45 were combined as:

\[ -(C_1 + \Delta t) T_{a_{i-1,j}} + C_1 B_2 \Delta t T_{a_{i,j}} + C_1 T_{r_{i,j}} - \frac{C_2}{2 \Delta x} T_{a_{i-1,j}} \]

\[ + \frac{C_2}{2 \Delta x} T_{a_{i,j}} + \frac{T_{a_{i,j}}}{\Delta t} = 0 \] (49)

A general form showing the air system temperature at a specific time and position was obtained by multiplying Equation 49 by \( \delta t \) and letting

\[ A = C_2 \Delta t / (2 \Delta x) \quad \text{and} \quad B = C_1 \Delta t - C_1 B_2 \Delta t^2 \quad \text{and} \quad D = C_1 \Delta t \] as
Equation 50 represents the calculation of the interior temperature field. Driving forces provided by inlet temperature changes must be incorporated as the boundary conditions to obtain the solution. No further treatment was applied to the calculation of inlet temperature since the incoming ambient air temperature controls the temperature profile pattern change. The two terminal element that represents the outlet point is located in a position as shown in Figure 2. One end of the element is attached to the rock bed interior points while the other is attached to the ambient air. Special arrangements must be made to account for the interface between the rock bed outlet and surrounding environment. If the boundary node \( n \) represents the outlet of the rock bed, simple linear extrapolation can be used to estimate the first point just outside of the boundary as:

\[
T_{i-1,j+1} = \frac{1}{2}(T_{i,j} + T_{i-1,j})
\]  

which leads to

\[
T_{i-1,j+1} = 2T_{i,j} - T_{i-1,j}
\]

Substituting Equation 51 into Equation 49, results in the following equation for outlet boundary temperatures:

\[
-AT_{a_i,j-1} - BT_{a_i,j} + AT_{a_i,j+1} = DT_{r_{i,j}} + T_{a_i,j}
\]  

or

\[
-2AT_{a_i,j-1} + (2A - B)T_{a_i,j} = DT_{r_{i,j}} + T_{a_i,j}
\]
Figure 1. Computational stencil diagram

Figure 2. Terminal points

Figure 3. Slicing pattern of rockbed for finite difference
By grouping similar terms in Equation 53 and letting \( E = -2A \) and \( F = 2A - B \), the outlet air temperatures are calculated by:

\[
\sum_{i,j} a_{i,j} \cdot F + T_{a_{i,j},+1} = D + T_{a_{i,j}}
\]  

(55)

A simplified sketch of the rock bed system under the finite difference scheme is shown in Figure 3. The rock bed was divided into \( n \) homogeneous planes. Algebraic equations for boundary and interior points were formulated for each plane throughout the rock bed. The solution to the linear equation system is driven by the known inlet temperatures. The inlet temperature terms, \( T_{a1,j+1} \), vary with time as the ambient temperature moved to right hand side, since ambient temperature is known at any instant. The individual algebraic form of the rock-air interfaces were assembled into matrix form as shown in Equation 56. A matrix consisting of \( n \) simultaneous equations with \( n \) unknowns can be written in a general form:

\[
\bar{A}\bar{x} = \bar{f}
\]  

(56)

where

\[
\bar{A} = \text{the coefficients matrix}
\]

\[
\bar{f} = (f_1, f_2, f_3, \ldots, f_n)^T
\]  

(57)

\[
\bar{x} = (x_1, x_2, x_3, \ldots, x_n)^T
\]  

(58)

Two methods frequently used to solve systems of linear equations are the direct method and the iterative method. A direct method assumes no round off error during computation process disregard the significant digits restricted by computational tools. Consider an iterative method such as Gauss-Siedel method. The results from each operation are treated as the initial guess for the next operation which implies no inherited round off error between each step. Numerous arithmetic operations are involved in the direct solution.
method. The number of operations required to calculate the matrix solution for n unknowns in each time step is eight times n. Carnahan et al. (1969) suggested the direct method should be used when the number of equations involved is less than forty and that iterative methods are appropriate for a system with one hundred or more equations.

The linear equation system constructed from the finite difference scheme forms a matrix with a nonzero element band in the diagonal and zeros elsewhere. Unnecessary repetitive operations made for elements with only zero's causes needless computer running time and memory space. A lower and upper triangle (LU) Decomposition Method called Crout Reduction appeared to be a better alternative.

Consider a matrix \( M \) with diagonal band elements as shown

\[
M = \begin{pmatrix}
\alpha_1 & c_1 & 0 & 0 & \ldots & 0 \\
b_2 & \alpha_2 & c_2 & 0 & \ldots & 0 \\
0 & b_3 & \alpha_3 & c_3 & \ldots & 0 \\
0 & 0 & b_4 & \alpha_4 & \ldots & 0 \\
\vdots & \vdots & \vdots & \vdots & \ddots & \vdots \\
0 & 0 & 0 & 0 & \ldots & \alpha_n \\
\end{pmatrix}
\]

Assume \( M \) can be decomposed into two matrices, \( L \) and \( U \). The \( L \) represents a matrix with 0's entered above the diagonal. The \( U \) represents for a matrix with 1's in the diagonal and 0's below the diagonal. \( M \) can be rewritten as:

\[
M = LU
\]

\[
L = \begin{pmatrix}
\alpha_1 & 0 & 0 & 0 & \ldots & 0 \\
b_2 & \alpha_2 & 0 & 0 & \ldots & 0 \\
0 & b_3 & \alpha_3 & 0 & \ldots & 0 \\
\vdots & \vdots & \vdots & \vdots & \ddots & \vdots \\
0 & 0 & \ldots & \alpha_n & b_n \\
\end{pmatrix}
\]
The following relations were obtained by comparing the original elements as:

\[
\begin{align*}
\alpha_1 &= \alpha_1 \\
\alpha_1 \beta_1 &= c_1 \\
b_2 \beta_1 + \alpha_1 &= \alpha_2 \\
\alpha_2 \beta_2 &= c_2 \\
b_4 \beta_3 + \alpha_4 &= \alpha_4
\end{align*}
\]

Solving for \( \alpha \) and \( \beta \) (from Equation 61 to Equation 65) yields:

\[
\begin{align*}
\alpha_1 &= \alpha_1 \\
\beta_1 &= c_1 / \alpha_1 \\
\alpha_i &= \alpha_i - b_i \beta_{i-1} \\
\beta_i &= c_i / \alpha_i \quad i = 2, 3, 4, ..., n-1 \\
\alpha_n &= \alpha_n - b_n \beta_{n-1}
\end{align*}
\]

All elements in the L and U matrices are obtained through the same procedures which required approximately 8 times n operations. The calculated L and U matrices are condensed into a one dimensional array and stored in the memory space where the original matrix resided.
The system of linear equations that represent heat transfer within a rock bed thermal storage unit are described in general form as:

\[ M \bar{T} = \bar{f} \]  

(71)

where

\[ \bar{T} = \text{column vector which represents unknown temperature} \]

\[ \bar{f} = \text{initial and boundary condition combination column} \]

Matrix \( M \) is then decomposed into \( L \) and \( U \) matrices. Letting \( L(UT) = f \) and \( Z = UT \), equation 71 become: \( LZ = f \) or

\[
\begin{pmatrix}
\alpha_1 & 0 & 0 & 0 & \cdots & 0 \\
0 & \alpha_2 & 0 & 0 & \cdots & 0 \\
0 & 0 & \alpha_3 & 0 & \cdots & 0 \\
\vdots & \vdots & \vdots & \vdots & \ddots & \vdots \\
0 & 0 & 0 & \cdots & \cdots & \alpha_n \\
\end{pmatrix}
\begin{pmatrix}
z_1 \\
z_2 \\
z_3 \\
\vdots \\
z_n \\
\end{pmatrix} =
\begin{pmatrix}
f_1 \\
f_2 \\
f_3 \\
\vdots \\
f_n \\
\end{pmatrix}
\]

The column vector \( Z \) can be readily obtained as

\[ Z_1 = f_1/\alpha_1 \]  

(73)

Reapply the \( UT = Z \) relationship to the matrix after solving \( Z \) column and obtain

\[ Z_i = (f_i - b_i Z_{i-1})/\alpha_i, \quad i=2,3,...,n \]  

(74)

\[
\begin{pmatrix}
1 & \beta_1 & 0 & 0 & \cdots & 0 \\
0 & 1 & \beta_2 & 0 & \cdots & 0 \\
0 & 0 & 1 & \beta_3 & \cdots & 0 \\
\vdots & \vdots & \vdots & \vdots & \ddots & \vdots \\
0 & 0 & 0 & \cdots & 1 & \beta_{n-1} \\
0 & 0 & 0 & 0 & 1 \\
\end{pmatrix}
\begin{pmatrix}
t_1 \\
t_2 \\
t_3 \\
\vdots \\
t_n \\
\end{pmatrix} =
\begin{pmatrix}
z_1 \\
z_2 \\
z_3 \\
\vdots \\
z_n \\
\end{pmatrix}
\]
Backward substitution is then applied to obtain the entire temperature field $t_1, \ldots, t_n$ for one time step as:

$$t_n = Z_n$$  \hspace{1cm} (76)

$$t_i = Z_i - \beta_i t_{i-1}, \hspace{0.5cm} i = n-1, \ldots, 1$$  \hspace{1cm} (77)

Calculated interior point temperatures are moved to the right hand side and become the initial condition of the next time step. The new initial condition combined with the current inlet temperature provide the new right hand side for the next calculation. The updated temperature calculation only repeats Equations 73 through 77. No matrix decomposition is required since the properties of gravel and rock bed were assumed homogeneous and remain constant within the period of interest. The elements in the coefficient matrix remain the same in every time step.

**Broiler House Ventilation Model**

Broiler house energy balance

Proper ventilation of a warm confinement broiler house requires balancing heat gains and losses. The major components of heat gain that become the cooling load within a broiler house during warm weather periods are illustrated in Figure 4. Most of the radiant energy that strikes a building is absorbed by the walls and roof. The absorbed energy does not become a part of the cooling load until released in the form of convection after a certain time lag. Part of the sensible heat produced by the broilers, equipment, and lights is also momentarily stored in the surrounding before becoming part of the cooling load. All the latent heat produced goes directly toward heating the surrounding air. Interior space total heat gain for a given time increment can be expressed as:

$$\dot{q}_t = \dot{q}_{l,t} + \dot{q}_{r,t} + \dot{q}_{s,t} + \dot{q}_{l,t}$$  \hspace{1cm} (78)
Figure 4. Major components of heat gain and cooling load
where

\[ q_t = \text{total heat gain at time } t, \text{ W} \]

\[ q_{c,t} = \text{convective heat transfer from inside boundary surface, W} \]

\[ q_{r,t} = \text{radiation heat transfer between boundary surface and other interior surface, W} \]

\[ q_{s,t} = \text{incoming solar radiation through window, W} \]

\[ q_{e,t} = \text{internal heat gain from lights, equipment, and occupant, W} \]

Heat conduction through the envelope boundaries of a broiler house which is exposed to changing temperature and solar radiation creates non-linear, time-dependent boundary conditions as a major obstacle to achieving solutions to building heat transfer models. The heat transfer function method, which considers the heat removed from the air in the space equal to the cooling load, is the best option to solve the problem. The assumption of constant space air temperature leads to the following equation:

\[
\sum_{i=0}^{1} P_i (q_{x,t-i\Delta} - q_{c,t-i\Delta}) = \sum_{i=0}^{2} g_i (T_i - T_{r,t-i\Delta})
\]

(79)

where

\[ P_i, g_i = \text{transfer function coefficients} \]

\[ q_c = \text{cooling load at various time} \]

\[ T_i = \text{inside temperature used for cooling load calculation, } ^\circ\text{C} \]

\[ T_r = \text{actual inside temperature at various times, } ^\circ\text{C} \]

\[ \Delta = \text{time increment} \]
The transfer function method provides good accuracy through the use of many design details. However, complicated calculations frequently render this method impractical even with the help of a computer. A cooling load temperature difference (CLTD) method used for ventilation design with extensive uses of charts, tables, and various dynamic factors was developed by McQuiston and Parker (1982). The CLTD method uses cooling load temperature difference for walls and roof and incorporates cooling load factors (CLF) to account for solar heat gain and heat transfer from internal sources. The tabulated CLTD was determined using following conditions:

1. Solar radiation incident on a dark surface
2. Design inside temperature equal to 26 °C
3. Maximum outdoor temperature 35 °C with diurnal range of 12 °C
4. Solar radiation taken at 40 degrees North latitude on July 21
5. Outside convective film coefficient, 17 W/m²°C
6. Inside convective film coefficient, 8.3 W/m²°C
7. No air duct or any forced ventilation in ceiling space.

General procedures to calculate external and internal heat gains are:

(1) External Energy Sources
Case 1. Walls and Roof

\[ q_{t,e} = UA(\text{CLTD}) \]  

(80)

where

\[ q_{t,e} = \text{heat gain from external source, W} \]

\[ U = \text{overall heat transfer coefficient, W/m}^2\text{-°C} \]
Tabulated CLTD values which account for existing thermal lag within walls, the roof, and internal surfaces are available for different months, latitudes, building color codes, and environmental conditions. Further adjustments, as shown in Equation 81, are used to compensate for the specific broiler house ventilation requirements under different weather and environmental conditions.

\[ CLTD_e = (CLTD + LM)k + (25.56 - T_i) + (T_{oa} - 29.44) \]  

(81)

where

- CLTD = tabulated value
- LM = correction factor for latitude and month, °C
- K = building color adjustment factor
- \( T_i \) = designed inside temperature, °C
- \( T_{oa} \) = average outside temperature, °C

Case 2. Curtains and Openings.

The heat gain through the curtains and openings are handled by solar heat gain factors (SHGF) which are defined as the maximum solar heat gain for the particular month, orientation, and latitude. Variations of SHGF, building structure types and shading coefficient should be further adjusted by using cooling load factors from McQuiston and Parker (1982). The heat gain from curtains and openings is calculated by

\[ q_{i,c} = A \cdot SC \cdot SHGF \cdot CLF_i \]  

(82)

where

- \( q_{i,c} \) = heat gain from curtain and/or openings, W
A = curtain and/or opening area, \( m^2 \)

SC = shading coefficient

SHGF = solar heat gain factor, \( W/m^2 \)

CLF_t = cooling load factor at time t

(2) Internal Sources

Modern broiler production facilities have a high density of birds in the controlled environment. The residing birds provide significant amounts of sensible and latent heat after the first 3 to 4 weeks. Latent heat becomes part of the cooling load without a time lag. A portion of the sensible heat generated is absorbed by the floor or other internal surfaces therefore starts a thermal lag process. The CLF is a function of entry time and total hours that occupants stay in the building. A CLF value equal to one for sensible heat thermal lag should be used since sensible heat thermal lag is balanced by high bird density in the house and continuous operation of the ventilation system. Lighting equipment is another source of sensible heat gain. A portion of the energy from lighting devices is radiated into the surrounding and stored. The power rating of lighting devices is therefore significantly different from the instantaneous heat gain of the surrounding air. The cooling load caused by lighting devices and other electrical equipment can be calculated by:

\[
q_{t,w} = 3.412 \cdot W \cdot F_u \cdot F_s
\]

(83)

where

\( q_{t,w} \) = instantaneous heat gain from electric equipment, W

\( W \) = summation of all installed light wattage, W

\( F_u \) = use factor

\( F_s \) = special allowance factor for lights required more power than rating wattage
Several different temperature regimes have been recommended for brooding. North (1972) recommended that the inside temperature of a broiler house during summer should be approximately 21 to 24 degrees C for ages of one to four weeks and approximately 18 to 21 degrees C for ages of four to eight weeks. Another suggestion, based on evaluation of feed conversion, mortality, percent body fat, and fuel consumption, was reported by Felton (1974). Felton suggested that brooding at 27 degrees C during the first week and reducing the temperature 2.2 degrees C each following week to 21 degrees C, could be a feasible temperature schedule. Reece (1982) suggested the following general temperature regime guideline for broiler houses:

Table 1 Suggested Brooding Temperature Regime

<table>
<thead>
<tr>
<th>Period during brood</th>
<th>Temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td>First Week</td>
<td>30 to 35 °C</td>
</tr>
<tr>
<td>Second Week</td>
<td>27 to 32 °C</td>
</tr>
<tr>
<td>Third Week</td>
<td>24 to 29 °C</td>
</tr>
<tr>
<td>Fourth week and later</td>
<td>21 to 26 °C</td>
</tr>
</tbody>
</table>

Table 2 Broiler House Environmental Parameters

<table>
<thead>
<tr>
<th>Week</th>
<th>RH (%)</th>
<th>Inside Temp(°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>60</td>
<td>28</td>
</tr>
<tr>
<td>2</td>
<td>60</td>
<td>26</td>
</tr>
<tr>
<td>3</td>
<td>60</td>
<td>24</td>
</tr>
<tr>
<td>4</td>
<td>70</td>
<td>22</td>
</tr>
<tr>
<td>5 and &gt;</td>
<td>70</td>
<td>21</td>
</tr>
</tbody>
</table>

A combination of these reports was analyzed and provided the basis for the design temperatures selected for the broiler house simulation (Table 1). Relative humidity levels were set by referring to North (1972) and Hellickson and Ryan (1984). Table 1 lists the simulation values for each week.

Seasonal temperature changes and environmental condition adjustments were done by combining Equation 81 with the approximate
brooding date (Table 3) and eight week commercial broiler production design inside temperatures (Table 2). Corrected CLTD values were substituted for the original design values.

### Table 3 Approximate Commercial Broiler Brooding Dates

<table>
<thead>
<tr>
<th>Brood No.</th>
<th>Starting Date</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Jan 3</td>
</tr>
<tr>
<td>2</td>
<td>Mar 7</td>
</tr>
<tr>
<td>3</td>
<td>May 9</td>
</tr>
<tr>
<td>4</td>
<td>Jul 5</td>
</tr>
<tr>
<td>5</td>
<td>Sep 6</td>
</tr>
</tbody>
</table>

**Modeling Broiler Heat and Moisture Production**

Computer simulated design relies on the basic models of heat and moisture production from broilers. Such models should include the total heat and moisture generated by the chickens and associated litter under normal growing conditions. The broiler body weight serves as an essential independent variable in heat and moisture production. A relationship between body weight and age was obtained by combining data from North (1972) and Broiler Industry (1980). A regression analysis of the combined data provided Equation 84 which was used to provide the relation of weight versus age used in the ventilation model.

\[
W = 0.138492 + 0.1036 \times 10^{-2}N + 0.152572 \times 10^{-2}N^2 \\
-0.558865 \times 10^{-5}N^3 - 0.696725 \times 10^{-7}N^4
\]  

(84)

where

\( W \) = chicken weight, Kg

\( N \) = days of age
Many investigations have been conducted to determine heat and moisture production values of broilers for different temperatures, and body weights and environmental conditions. Data from Colby et al. (1967) and Longhouse (1968) did not include moisture production from litter. A report by Reece and Deaton (1969) concerning general concept of chicken heat production and some experimental data was informative but lacked of generalized mathematical model and cannot be used for simulation. Reece and Lott (1980) extended the same research to formulate sensible and latent heat production values related to broiler body weight under different environmental temperatures. Combined data from Lampman (1967), Reece and Lott (1980), and North (1972) were used to derive the following broiler sensible and moisture production equations:

\[
SH = 14.2393 - 4.94177W + 1.23608W^2 - 0.228763W^3 + 0.023112W^4
\]  
\[
(85)
\]

\[
H_2O = 3.05428 - 0.196998N + 0.0598057N^2 - 0.00193599N^3 + 0.000019599N^4
\]  
\[
(86)
\]

where

- \( SH \) = sensible heat, Btu/hr-lb
- \( H_2O \) = moisture production, \( lbH_2O/1000\) birds-hr
- \( W \) = chicken weight, lb

The ventilated broiler house model included structural dimensions, insulation levels, and building materials as shown in Figure 5. The 594 square meter floor area (6400 square ft) was designed for 8000 birds under density of 0.074 m\(^2\)/bird (0.8 ft\(^2\)/bird). Heat transfer coefficients for various sections of the
house were calculated based on ASHRAE Handbook of Fundamental (1977) recommendations. The thermal resistance and equivalent U values listed in Table 4, were calculated by assuming one tenth of the wall area contained framing.

### Table 4 Simulated Broiler House Material Characteristics

<table>
<thead>
<tr>
<th></th>
<th>Wall</th>
<th>In-frame</th>
<th>Between frame</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inside surface (still air)</td>
<td>0.68</td>
<td>0.68</td>
<td></td>
</tr>
<tr>
<td>Plywood (1/2 inch)</td>
<td>0.62</td>
<td>0.62</td>
<td></td>
</tr>
<tr>
<td>R-11</td>
<td>-</td>
<td>11</td>
<td></td>
</tr>
<tr>
<td>Plywood (1/2 inch)</td>
<td>0.62</td>
<td>0.62</td>
<td></td>
</tr>
<tr>
<td>Wood stud (2x4)</td>
<td>4.38</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>Outside surface (15 mph)</td>
<td>0.17</td>
<td>0.17</td>
<td></td>
</tr>
<tr>
<td>Total R value</td>
<td>6.47</td>
<td>13.09</td>
<td></td>
</tr>
<tr>
<td>U value (1/10 framing)</td>
<td>-</td>
<td>0.08422</td>
<td></td>
</tr>
</tbody>
</table>

### Roof & ceiling

<table>
<thead>
<tr>
<th></th>
<th>Wall</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Upper side surface (15 mph)</td>
<td>0.17</td>
<td></td>
</tr>
<tr>
<td>Plywood (1/2 inch)</td>
<td>0.62</td>
<td></td>
</tr>
<tr>
<td>Glass wool (4 inch)</td>
<td>14.00</td>
<td></td>
</tr>
<tr>
<td>Plywood (1/2 inch)</td>
<td>0.62</td>
<td></td>
</tr>
<tr>
<td>Inside surface (still air)</td>
<td>0.61</td>
<td></td>
</tr>
<tr>
<td>Total R value</td>
<td>16.02</td>
<td></td>
</tr>
<tr>
<td>U value</td>
<td>0.0624</td>
<td></td>
</tr>
</tbody>
</table>
Figure 5. Simulated broiler house construction
Rock Bed Design and System Operation

A simulation program which incorporated a psychrometric chart routine and a cooling load calculation method was written for the modeled broiler house. Simulations for each brooding period under 1982 and 1983 Corvallis, Oregon weather conditions (Table 5) showed the peak cooling load of the modeled broiler house occurred in August within brood number four for both years (refer to Table 3 for brood number). Dimensions of the rock bed heat sink that would be used to provide sensible cooled air to the broiler space were based on the main goals of minimal construction and operation costs with possible maximum efficiency. Minimal construction and operation costs can be achieved by optimizing storage unit size. System thermal capacity and physical dimensions of the storage unit were calculated with the simple energy balance as:

\[
V_m = \frac{P_{CL}}{(\rho_{rock} \cdot (1-\varepsilon) \cdot C_{pr} \cdot \Delta T_{ave})}
\]

where

- \( V_m \) = designed rock bed volume, \( m^3 \)
- \( \varepsilon \) = void fraction
- \( P_{CL} \) = possible peak cooling load during brooding, \( W/hr \)
- \( \rho_{rock} \) = density of the rock, \( Kg/m^3 \)
- \( C_{pr} \) = specific heat of rock, \( J/kg-^\circ C \)
- \( \Delta T_{ave} \) = designed average difference between ambient and rock bed temperature, \( ^\circ C \)

The design average temperature difference influenced system performance in two ways. In practice, the charging rate should be maintained within certain range. A large charging rate is needed to provide the required thermal balance when the design rock bed
temperature difference is small. The magnitude of air flow rate necessary to match a large charging requirement could exceed the thermal limit of the storage. Energy loss to the environment also resulted from the use of large design temperature difference. Coutier and Farber (1982) recommended that the average temperature difference between ambient and storage unit should range from 5 to 12 °C for passive energy systems and 10 to 20 °C for active systems. The upper level of 12 °C for passive systems was chosen for the well insulated rock bed storage unit (average heat transfer coefficient of .00252 w/m²). The cooling loads under different Corvallis, OR weather conditions are calculated. The required rock bed volume was calculated and then adjusted to fit standard commercial construction material sizes.

The placement of the rock bed was based on the assumption that maximum thermal stratification can be achieved regardless of the air flow direction. Later experimentation confirmed that the destratification of a charged rock bed was independent to the charging direction.
<table>
<thead>
<tr>
<th>Item</th>
<th>Case I</th>
<th>Case II</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outside (F)</td>
<td>33.8</td>
<td>31.6</td>
</tr>
<tr>
<td>Temperature Difference</td>
<td>17.2</td>
<td>17.2</td>
</tr>
<tr>
<td>$E_{\text{out}}$ (watts)</td>
<td>-1932</td>
<td>-1376</td>
</tr>
<tr>
<td>$E_{\text{heat}}$ (watts)</td>
<td>36555</td>
<td>37296</td>
</tr>
<tr>
<td>$E_{\text{moisture}}$ (watts)</td>
<td>-10871</td>
<td>-15499</td>
</tr>
<tr>
<td>$E_{\text{building}}$ (watts)</td>
<td>31158</td>
<td>31921</td>
</tr>
<tr>
<td>$E_{\text{sensible heat}}$ (watts)</td>
<td>3452</td>
<td>3452</td>
</tr>
<tr>
<td>$E_{\text{equip.}}$ (watts)</td>
<td>1943</td>
<td>1922</td>
</tr>
<tr>
<td>Total (watts)</td>
<td>60327</td>
<td>57717</td>
</tr>
</tbody>
</table>
Control Theory

Micro-electronics and modern control theory have been adapted to manipulate operation of numerous physical devices. Basic control systems are capable of receiving and processing a variety of inputs. The ability of these devices to achieve required levels of performance depends upon both the physical arrangement of the transfer set and the software involved. The transfer set is the interconnected series of components which forms the major hardware portion of the system. The control algorithm or software is the series of instructions that are used to achieve the desired operation of the system. The control algorithm is often partitioned into subsets each of which performs a basic task. An overall control solution is then achieved by integrating the subset network. Modern control theories are usually combined with electronics and mechanical devices to perform a complicated task such as an automatic ventilation control system in the project.

Basic servo-mechanisms utilize feedback theory to control components such as transducers, actuators, and controllers. A completed servo control loop performs predetermined tasks through input signal processing by positioning and moving system components. Application of microprocessor control in livestock rearing systems for such purposes as production monitoring, health monitoring, and management have been developed. All involve a combination of software design and servomechanism arrangements. The positive and negative feedback controls are mainly classified according to the processes taken by the control algorithm when it receives transducer signals. The closed loop control system mentioned by Leonard and McQuitty (1982) is a typical negative feedback control and has been adapted widely in industry. A negative feedback control obtains an
error signal by comparing the incoming transducer signals to preset values as shown in Figure 6. The error actuates the controller which provides a control signal that affects the servomechanism in such a way that it reduce the error. The comparison procedure continues until a zero error or a tolerable difference is obtained. The tolerable accuracy is always kept flexible by either software or hardware control. Real time controls carry out predetermined tasks at regular intervals as well as respond to unscheduled external events.

Control Method

Hardware Design Considerations and Design Procedures

Microprocessor based system development is similar to conventional mechanical system design. The general approach of designing a system includes setting specifications, developing control algorithm flow charts, hardware selection and arrangement, testing and debugging. A major effort for this project was placed on developing the kind of hardware needed to achieve the required efficiency early in the design stage.

The minimum configuration for a microprocessor controller requires a central processing unit (CPU), random access memories (RAM's), read only memories (ROM's), bus circuits, and appropriate interface adapters as shown in Figure 7. Three elements often considered in addition to the above minimum configuration are: (1) the actual peripheral equipment that is dictated by the system specification; (2) any conventional electronics required to control the peripherals; (3) the "intelligence" that enables the system to perform the required control and data processing functions. System flexibility is accomplished by incorporation of a keyboard which allows the adjustment of control variable settings.

Allowable heat stress and minimum ventilation requirements are
Figure 6. Simplified negative feedback loop

Figure 7. Schematic diagram for a microprocessor system
essential factors to be considered in the design of a controller. A wide range of temperatures will be encountered in chicken production (Reece, 1982). Relative humidity (RH) has little direct effect on the poultry when inside temperatures are within the range of 10 to 26.7 °C. Relative humidity becomes a critical factor in chicken growth and survival only when poultry are exposed to temperature over 27 °C (Reece, 1982). Reece et al. (1972) indicated that eight week old broilers can survive at 17 percent relative humidity when environment temperatures rise to 41 °C. The mortality from high temperature stresses became appreciable at 33 percent relative humidity and reached severe levels as relative humidity increased to 39 percent.

Automated control systems have been developed for ventilated broiler houses. A method presented by Kay and Allison (1983) was closely reviewed to help in selecting basic system components. Kay and Allison utilized an MM6801 monoboard microcomputer and a Tektronix 8002A microprocessor development system to develop control algorithms and hardware interface circuits. The choice of the hardware components was based on a feasibility study prior to start the project. The merits of various technical solutions, economical considerations, simplicity, and flexibility were evaluated. Flexible and easily interfaced hardware were preferred in system development due to a highly limited budget.

System Requirements and Specifications

An ET-3400 microcomputer system obtainable in a kit form from Heath Company, was selected. The model ET-3400 Microprocessor trainer is especially designed as a learning tool to teach microprocessor operation, programming, and applications. Assembly of the computer requires some practical experience, but overall operations are easily performed. A complete ET-3400 is divided into
several sectors which include CPU, RAM, ROM, Display, Keyboard Buffer, Address buffer, Control Buffer, Data Input/Output (I/O), Binary Data, and interface breadboard. All sectors are identified on the face board. Figure 8 is a picture of an assembled system.

Central Process Unit (CPU) and Associated Memory

The center of the control unit is a Motorola MC 6808 microprocessor which contains approximately 5000 logic circuits and an internal clock oscillator with a driver circuit. Instructions, issued by the CPU are transferred to designated devices through an 8-bit bidirectional data bus. Total of 64,000 addresses can be encoded by the address bus due to the MC 6808 8 bit structure. Input and output interface unit (such as the PIA) are assigned to specific memory areas by the combined decoding of selected address and control lines as shown in Figure 9. The use of address decoding combined with software control further provides the ability to undertake flexible applications to substitute software for special purpose chips, such as an A/D converter. Reducing the use of combinational logic units allows more flexibility to the system since the total number of devices that a microprocessor can drive is limited by the allowable current sink of the microprocessor and PIA's. Even with the use of a standard peripheral interface adapter, such as PIA MC6821, the output port can only drive a certain number of external devices. The externally connected components usually require additional signal amplification or conditioning. All of these procedures will increase the working load for the PIA and CPU. A common approach to minimize complexity is to use compatible electronic components that share the same electrical characteristics. Typically, a microprocessor can drive seven to ten of these devices directly without signal modifications or temporary storage.

Following the instructions from the CPU, supporting information
Figure 8. Assembled ET-3400 microcomputer

Figure 9. Memory addressing map
is retrieved from or stored to memory devices. Memory contents on the data bus were decoded by specific address lines. The complexity of decoding was minimized by linear decoding of multiple address lines. Without linear decoding, the peripheral devices would need to be assigned to some specific memory block for microprocessor access. Linear decoding utilizes the memory map concept as shown in Figure 7. Memory devices connected to the data and address buses are assigned to a user defined combination of address signals. Two MCM 2114 RAMs combined with a ROM provide a 3 kilo-byte memory capacity. The 2 kilo-byte RAMs facilitate testing and debugging the design program. A routine for scanning and displaying the keyboard status, the contents of accumulators, control registers, program counter, and index register is masked into the ROM to facilitate direct communication between the CPU and the user.

**Input, Output and Other Supporting Devices**

Most microprocessor-based systems normally have a particular control signal set which can activate a specific memory unit. Data transfer to peripheral devices such as PIA's typically has another independent set of control signals. A different approach was used in the system designed for this project. The CPU does not make a distinction between memory devices and the peripheral devices. The working chips were activated by control signals on the address bus combined with software procedures.

Instructions and data can be entered through the dual function keyboard in hexadecimal form or through direct access points which connected directly to address and data buses using binary code. Data bus contents are available in either direct access to the data bus or in visual display with hexadecimal form by a set of 7-segment light
emitting diodes (LED's). No further memory backup (disk or cassette drive) or data communication system (RS 232, serial, and parallel port) were used in the system.

Data Acquisition and Interface

The primary control signal source in a closed loop operation comes from a linear combination of output feedback and preset input as shown in Figure 6. The signals must be amplified to the voltage level required. The MC6808 microprocessor is a transistor to transistor logic (TTL) compatible device. The TTL devices are designed to operate within a power source range of ± 5 volts which dictates that input and output signals should be limited within -5 to +5 volts. The project design assumed that TTL devices would be used throughout to insure compatibility with the central processor unit. The design based on TTL specifications does not restrict the possible interface to the CMOS system, since the operation of the CMOS family does not require precise voltage levels. Different characteristics are embedded in signals from various analog devices. Most primitive signals contain noise and a certain degree of fluctuation which cannot be used by electronic devices. Special procedures are used to pre-process primitive signals into a usable form.

(1) Transducer and Signal Conditioning

Commercially available temperature measurement transducers can be categorized into thermocouples, resistance temperature detectors (RTD), thermistors, and integrated circuit sensors (Figure 10). All of these incorporate distinctly different characteristics curves and have certain advantages and disadvantages depending upon the required application. Thermocouples were selected as the temperature measurement transducer to interface with the system. Typically, broiler houses experience slow and steady temperature changes. The steady broiler house temperature changes produces a steadily changing
Figure 10. Characteristics curves for different temperature sensors
thermocouple emf output. Theoretically, only static signals require preprocessing such as conversion, linearization, and amplification.

Type T thermocouples generate analog signals smaller than 10 millivolts, which were too low to be directly used by the system. The following procedures were used to adjust the signal to usable level:

1. Amplification
2. Noise rejection
3. Input protection
4. Cold junction compensation

Non-linear characteristics of the thermocouple were compensated for by an industrial standard conversion formula (Zuch 1979):

$$T_{\text{correction}} = A + \frac{E_b}{E_n} \left( B + C \frac{E_b}{E_n} \right)$$

where

$E_b = \text{output voltage from resistor bridge}$

$E_n = \text{input voltage}$

$A, B, C = \text{network bridge parameters, dimensionless}$

Figure 11 shows the unconditioned reading $T_{\text{uc}}$ obtained using the input from voltage-temperature curve. The corrected temperature $T_{\text{cr}}$ can be found by tracing through the intersection of the linearized curve and the constant temperature ($T_{\text{uc}}$) line. Andrews (1982) applied the finite difference concept to a program which treats the linearized curve as a finite set of straight lines and then computes a correction factor $D_t$. The process requires many repetitive calculations which made it unsuitable for the microprocessor used here. Another simple and straightforward method was adopted in this
project to allow the use of minimum system memory space. The method combined the National Bureau of Standards Thermocouple Table with system software compensation. A table of voltage output from thermocouples and corresponding binary code was constructed through the calibration of operational amplifier outputs. Tables of temperatures and equivalent voltages were calculated using a seventh order nested polynomial form representing the T type thermocouple. A look up table was constructed to correct the binary code and temperature in degree Celsius.

Thermocouple accuracy also depends on the temperature control of the reference junction. The reference junction should be arranged so that the thermal emf at the instrument end can be reduced as much as possible. An ice bath appears to be an easy way to accomplish this but would be impractical for most in-field measurements. Alternative methods such as an electrical bridge, thermoelectric-refrigeration or a heated oven reference are available. The electrical bridge method was selected due to the low cost compared to both the thermoelectric refrigeration and heated oven reference methods. The electronic ice point reference method used for the cold junction compensation utilizes a self-compensating bridge network, a temperature sensitive resistance component, and a stable DC power supply as shown in Figure 12.

The bridge output voltage is proportional to the off balanced electrical force between reference point temperature at \( T_2 \) and the measurement point at \( T_1 \). Variations of the reference point surrounding temperature introduce a thermal voltage which causes some measurement error. However, the error voltage is balanced by an equivalent voltage (opposite polarity) generated by the temperature sensitive resistor which assures high precision for a wide
Figure 11. Thermocouple output conditioning

Figure 12. Electric ice point reference

Figure 13. Hardware compensation circuit
temperature range. The electronic ice point reference circuit was constructed for a T type thermocouple interface as shown in Figure 13.

(2) Signal sampling considerations

The thermocouple output signal was assumed to be static in most cases. However, environmental conditions may change rapidly. Phase shift and lag variation at high frequency exist when environmental conditions change rapidly. The phase shift calculation with signals containing more than one frequency is very complicated. No special handling procedure was taken since high precision control was not required. Another signal sampling problem is aliasing that occurs when the sampling rate is slower than twice of the maximum frequency (Nyquist sampling theorem). An erroneous sampling of data at a half frequency of the true signal is illustrated in Figure 14. A low-pass filter was employed ahead of the analog-to-digital converter to eliminate unwanted high frequency signals and input noise. No other compensation is required since the lag caused by the additional low-pass filter itself is not significant.

(3) Sampling Devices and Circuit Layout

An analog signal is still not directly compatible with the data input register inside the microprocessor after passing through all the conditioning procedures. Only a few types of microprocessors have on-chip analog to digital converter (A/D converters) ready to accept and process incoming analog signals without additional external signal conditioning. An ADC0804 analog to digital converter was chosen to perform data conversion.

Continuous analog signals are broken into finite discrete events during the analog to digital conversion process. The signal clipping distorts the original signal and may cause errors. Manufacturers
Figure 14. Aliasing error during sampling process

Figure 15. Transfer function and error plot for ADC
usually provide transfer function and error plot information so the user can select a product with desired accuracy. Figure 15 shows the transfer function and error plot as found in a perfect A/D converter. The transfer function is used to illustrate the analog input range corresponding to one digit resolution. In general, a ±1/2 least significant bit (LSB) quantizing error embedded in all A/D converters can not be eliminated. The ADC 0804LCN is manufactured with ±1 least significant bit (LSB) accuracy. An 8 bit A/D converter is capable of representing 256 binary numbers. Assuming the temperatures to be monitored will range from 0 to 50°C, the LSB is then weighted as .2°C, and a ±1 LSB accuracy limits the maximum conversion error to 0.4°C which provides a reasonable accuracy for the controlled poultry environment. An A/D converter requires a finite amount of time to access and convert the analog signal to its binary equivalent. The aperture time was decided using the method presented by Andrews (1982). Conversion accuracy of the converter was guaranteed at a clock frequency of less than 640 KHz. The selected A/D converter, when operated with a 640 KHz clock rate, requires a 103 us conversion time which is capable of handling an input frequency of up to 10 Hz. Converter calibration was made through feedback adjustment of an operational amplifier located before the analog input. Temperatures ranging from 0 to 45 degrees need to be converted to corresponding voltages between 0 and 5 volt. Converted digital output values for an eight-bit converter must be in the range of 0 to 255 (equivalent to binary values from 00000000 to 11111111). Manufacturer guaranteed accuracy can be achieved when the internal clock frequency, $f_{clk}$, operates close to 640 KHz. By this criterion, the acceptable clock speed ranged between 275 and 1600 nano-second. The external resistor-capacitor (RC) was selected based on the following calculation:
\[ f_{\text{clk}} = \frac{1}{1.1 \cdot RC} \] (89)

where

R: external resistance (ohm)
C: external capacitance (farad)

If R=10,000 ohm and \( f_{\text{clk}} = 640,000 \text{ Hz} \), then the calculated external capacitor required is 150 pf.

A multiplexing technique was used to monitor seven thermocouple temperature channels. Signals from the sensors were connected to the relative input channel of an MC14051 analog multiplexer. Channel selection signals generated by software are transmitted to the multiplexer channel select pins (pins 9, 10, 11). No signal pre-conditioning is required at this point. A circuit containing a LM 317 operational amplifier and accessory electronic devices (Figure 16) was set up for the first stage signal conditioning after signals passed through the multiplexer. A start-conversion signal is generated by the microprocessor immediately after the selected output channel sends the analog data into the converter. The start conversion signal sent to the A/D converter activates the conversion processes. Digital data from the A/D converter is then gated through a peripheral interface adapter (PIA 6821) to the CPU. The "end of conversion signal" from the converter status line returns the peripheral interface adapter to its stand-by mode.

Switching of the electronic gate in the A/D converter during the data sampling process causes transient power level changes (spikes) which may introduce conversion errors. A low inductance tantalum filter-capacitor was connected to the pin 20 of the A/D converter power supply. Analog ground pin (pin 8 of ADC) was also isolated from both digital signal and analog signal to minimize conversion
Figure 16. First stage signal processing
error. The analog to digital conversion circuits were constructed on prototype circuit boards for operational convenience. Details of communication signals between the A/D converter, PIA, multiplexer, and MPU were discussed in the software design section.

(4) Peripheral interface adapter

The basic concept of using a PIA instead of other conventional combinational logic devices was to reduce the number of components used. The PIA chip handles the routine control and frees the MPU to complete the major calculations. The peripheral interface adapter also performs a temporary storage and signal reconditioning between the high-speed MPU and low-speed I/O devices. Figure 17 shows the block diagram of the PIA internal structure. The peripheral interface adapter is a programmable device of which the register configuration can be changed at will by program instructions. The special programmable features enable the PIA to perform both input and output functions according to the control task requested. No separate address decoder was used since the chip has its own on board address decoding ability. Details of the PIA and functional test are described in PROCEDURE section.

(5) Data Communication

Data input and output were directly tied to the I/O block in the trainer. A tri-state device was used to communicate with the MPU and to avoid the data flow conflicts caused by the bidirectional data bus in the system. Tri-state devices are made especially for gating digital information to or from data, address, or control buses. The device has three output states: the high (logic 1), low (logic 0), and high impedance states (stand-by). The logic 1 state activates the data buffer to receive data from the MPU, the logic 0 state activates the data buffer to send data to the MPU. The high
Figure 17. Peripheral interface adapter internal structure

Figure 18. Tri-state buffer for direction control
impedance state virtually disconnects the device from bus. Figure 18 indicates two 74LS243 tri-state buffers used to perform the gating. The direction of data flow was controlled by the output of the eight-input not-and (NAND) gate (marked as RE in Figure 18). The CPU transmits data to peripheral devices when the output of the NAND gate is high and receives data from external devices when the output of the NAND gate goes low.

Controller Block

Stepping motors are widely used in electromechanical positioning system to translate electrical pulses into fixed mechanical movements. Stepping motor applications include such things as printers, floppy disk drivers, numerical controlled machinery, and robots. Most stepping motors are controlled by a controller that generates a sequential current flow pattern into the motor winding. Commercially available stepping motor controllers provide multifunction control but at fairly high cost. A controller circuit was constructed with simple electronic devices that followed the stepping motor control techniques, but at reduced cost.

(1) Stepping Motor Control Techniques

Figure 19 is a simplified diagram of a basic stepping motor. A drive controller can be a pulse translator that provides the drive pattern, or a complicated circuit that provides driving pattern with acceleration and electronic damping. A permanent magnet stepping motor consists of a series of permanent magnets radially distributed on a rotor shaft surrounded by electromagnets attached to a stationary housing. The motor can be driven by energizing the electromagnets, which generates a sequential field pattern to drive the motor. Torque is produced as the motor magnets try to align with the polarity of the magnetic field. Figure 20 describes a series of
Figure 19. A simplified stepping motor diagram

Figure 20. Stepping motor operation sequences
motions caused by activating the stepping motor. The rotor consists of an axially oriented magnet with two hubs on each end of the magnet. The teeth at the north end are 180 degree out of phase from the teeth at south end. The stator has teeth as shown in Figure 20 (outside stationary parts). The teeth on the stator will never exactly line up with the teeth of rotor due to the difference in teeth number between the stator and rotor. The attraction between stator and rotor, known as residual torque, is the major force to hold the shaft at its current position when power is removed.

Details of the driving procedure to energize the magnetic field are also illustrated in Figure 20. The first step, as shown in Figure 20.a, is to energize pole A as north (positive polarity), pole C as south (negative polarity), and de-energize both pole B and pole D. The south end of the rotor tends to line up with pole A as shown in Figure 20.a. The second step is to energize pole D as south and pole B as north, with pole A and pole C de-energized. The rotor rotates a small angle in a clockwise direction to line up with pole B. The next step is to energize pole C as north and pole A as south. The rotor then lines up with pole C. The final step to complete a 90 degree clockwise rotation is to energize pole D as north and pole B as south. The rotor rotates again and lines up with pole D. High resolution rotation can be achieved by machining the stator and rotor into many teeth while each tooth functions as a pole. Stepping motors obtain small angle step increments by using a large numbers of poles. The center tapped stepping motor used in this project is different from the standard permanent magnet stepping motor. A center tapped stepping motor has twice as many windings as the standard motor. The diagrams of the center tapped motor and control circuit are shown in Figure 21. The clockwise rotation can be changed to counter-clockwise rotation by reversing the current flow direction in the winding or by using the alternate half of the center
tapped winding. A center tapped stepping motor with thinner windings provides a higher resistance and a lower time constant. External devices required to drive the stepping motor include: switches, inductive clamp diodes, dropping resistor, and a DC power supply.

(2) Stepping Motor Rating and Specification

Primary problems caused by transient energy generating effects in the motor are shown in Figure 22.a, 22.b, 22.c. The stepping motor operation is a sequential energize and de-energize of the motor windings. The opening of logic control switch induces electric current flow and energizes the windings. Further logic switch closure causes current flow pattern changes. An instantaneous voltage rise will occur with a continuous current inflow after a switch closure since the windings inside the motor act essentially as an inductor. The transient high voltage could easily damage the switching transistor. Various methods are adopted by industries for protection against transient high voltage. The simplest method, as shown in Figure 23.a, was employed in this project. A diode (IN 4002) was connected from the winding back to the power supply. This caused the current in the inductor passing through the diode to flow back to the power supply and eventually die out. Operational characteristics of the voltage and current are shown in Figure 23.b and 23.c.

A SLO-SYN model M062-FD03 stepping motor requires a 24 volt power supply and a current rating of 1.6 amp/winding with 3.3 ohm/winding resistance and 9 mH/winding inductance. Average winding current should be limited according to the manufacturer's specification. The current was limited to the maximum rating by a dropping resistor. The calculated resistor value for the specific motor was 11 ohms at 50 watts resistor/winding.

(3) Control Sequence
The external control circuit built to drive the center tapped stepping motor is shown in Figure 24. Drive bit patterns for sequential switching of the full step operation have been listed in Table 6.a. Clockwise rotation follows the step sequence 1-2-3-4-1. To reverse the direction of motor rotation one must perform switching steps in the order of 1-4-3-2-1. Resolution of the M062-FD03 stepping motor is 200 steps per revolution, each step provided 1.8 degree rotation. Half-step operation can be obtained by changing the logic sequence as shown in Table 6.b. The maximum step change rate, 3.141 %, happens when the stepping motor rotates from 88.2 degree to 90 degree (or from 90 degree to 91.8 degree). The half step maximum change rate 1.23%. Full step operation was assumed to provide sufficient accuracy. A bit pattern generator was build to operate under full step operation.

A complementary relationship between two air intake dampers was set up to achieve the required mixing ratio requirement (Figure 25). The damper angles needed were calculated and converted into step pulses by the microprocessor. Pulses from the logic sequence circuit trigger the motor and move the dampers to the correct positions to provide a required air mixture for specific temperature condition.
Figure 21. Center tapped stepping motor and control circuits

Figure 22. High transient voltage when motor shut off

Figure 23. Protection diode used to eliminate spike
Figure 24. Stepping motor control circuits
Figure 25. Complementary gear setup for air mixture

Table 6. Stepping motor driving bit patterns
(a) Full-step sequence

<table>
<thead>
<tr>
<th>STEP</th>
<th>SW1</th>
<th>SW2</th>
<th>SW3</th>
<th>SW4</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>OFF</td>
<td>ON</td>
<td>OFF</td>
<td>ON</td>
</tr>
<tr>
<td>2</td>
<td>OFF</td>
<td>ON</td>
<td>ON</td>
<td>OFF</td>
</tr>
<tr>
<td>3</td>
<td>ON</td>
<td>OFF</td>
<td>ON</td>
<td>OFF</td>
</tr>
<tr>
<td>4</td>
<td>ON</td>
<td>OFF</td>
<td>OFF</td>
<td>ON</td>
</tr>
<tr>
<td>1</td>
<td>OFF</td>
<td>ON</td>
<td>OFF</td>
<td>ON</td>
</tr>
</tbody>
</table>

(b) Half-step sequence

<table>
<thead>
<tr>
<th>STEP</th>
<th>SW1</th>
<th>SW2</th>
<th>SW3</th>
<th>SW4</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>OFF</td>
<td>ON</td>
<td>OFF</td>
<td>ON</td>
</tr>
<tr>
<td>2</td>
<td>OFF</td>
<td>ON</td>
<td>OFF</td>
<td>OFF</td>
</tr>
<tr>
<td>3</td>
<td>OFF</td>
<td>ON</td>
<td>ON</td>
<td>OFF</td>
</tr>
<tr>
<td>4</td>
<td>OFF</td>
<td>OFF</td>
<td>ON</td>
<td>OFF</td>
</tr>
<tr>
<td>5</td>
<td>OFF</td>
<td>OFF</td>
<td>ON</td>
<td>OFF</td>
</tr>
<tr>
<td>6</td>
<td>ON</td>
<td>OFF</td>
<td>OFF</td>
<td>ON</td>
</tr>
<tr>
<td>7</td>
<td>ON</td>
<td>OFF</td>
<td>OFF</td>
<td>ON</td>
</tr>
<tr>
<td>8</td>
<td>OFF</td>
<td>OFF</td>
<td>OFF</td>
<td>ON</td>
</tr>
<tr>
<td>1</td>
<td>OFF</td>
<td>ON</td>
<td>OFF</td>
<td>ON</td>
</tr>
</tbody>
</table>
PROCEDURE

Computer Simulation

The mathematical model of the rock bed thermal storage module was written in standard FORTRAN V and ran on the Oregon State University CDC Cyber 170 computer. The periodic function of a simple sine wave was considered sufficient to represent typical diurnal temperature variation and was used as input to test and debug the simulation program. System variables required to complete a simulation include: air density, specific heat, and velocity; rock density, specific heat, equivalent spherical diameter, void fraction, and rock bed length. Mathematical variables required were: total number of temperature measuring points along the rock bed, time increment, and data output interval. Each numerical simulation required an initial temperature condition at each node point. Nodal points were established by superimposing a theoretical grid upon the rock bed module that sliced the rectangular box into six layers each approximately .3 meter thick. Temperature distribution within the rock bed module was monitored by 35 thermocouples. Necessary temperature data for each layer along the rock bed were obtained from grouped thermocouple arrangement as shown in Figure 26. Computer simulations were performed for a homogeneous temperature distribution initial condition at each node and for nonuniformly distributed initial temperature conditions at any time during any operation cycle.

Rock Bed Construction

Simulation of summer brooding conditions with the broiler house ventilation model providing input to the rock bed thermal model was
Figure 26. Thermocouple matrix setup on diagonal plane
used to determine the ventilation requirement necessary to size the thermal storage test module. System parameters used in the simulation are listed in Table 7.

Table 7. System Parameter Values Used in Simulation

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air velocity, m/s</td>
<td>0.269</td>
</tr>
<tr>
<td>Density of air, kg/m³</td>
<td>1.204</td>
</tr>
<tr>
<td>Specific heat of air, J/kg-k</td>
<td>1012</td>
</tr>
<tr>
<td>Average diameter of rock, m</td>
<td>0.0286</td>
</tr>
<tr>
<td>Density of rock, kg/m³</td>
<td>1695</td>
</tr>
<tr>
<td>Specific heat of rock, J/kg-k</td>
<td>900</td>
</tr>
<tr>
<td>Void fraction</td>
<td>0.4375</td>
</tr>
<tr>
<td>Rock bed length, m</td>
<td>2.1589</td>
</tr>
</tbody>
</table>

The computer generated dimensions of the test module were adjusted slightly to allow the exterior covering to be made from standard sized plywood and minimize waste of materials. The nominal dimensions of the test rock bed were 2.4 meters by 1.8 meters by 1.2 meters high (Figure 27). The inside space available for rock storage was approximately 3.8 cubic meters. The test rock bed thermal storage was placed where direct sunshine could not reach before solar noon. The bottom and four side walls of the module were insulated with R-11 fiberglass blanket and the the top was insulated with R-13 fiberglass blanket (Figure 28) to provide additional protection from solar radiation incident upon the surface after solar noon.

River run gravel that ranged from 1.9 cm to 3.8 cm in diameter was used to form the rock bed. The bulk density, void fraction, and equivalent spherical diameter; defined as the diameter of a sphere with a volume equal to the average sized individual rock were
Figure 27. Complete schematic diagram of rockbed chamber
Figure 28. Rockbed thermal storage module construction details
determined by water displacement method.

Air flow for charging and discharging the rock bed was provided by a single speed, .55 meter diameter fan located at one end of the module. The fan was oriented such that air was forced through the module. A three meter long 0.39 m diameter metal duct was built and attached to a .48 m opening at the opposite end of the module (Figure 29) to facilitate air flow measurements.

**System Properties Measurement**

Accurate measurements of air flow are essential input values for the computer simulation. Two measurement methods were selected; a hot wire anemometer and a sharp-edged orifice. Hot wire anemometer measurements are based on resistance variations that depended on temperature, composition, and air viscosity. Ambient air with specific heat and temperatures exceeding the range for which the instrument was calibrated adversely affect accuracy. A non-uniformly distributed air velocity profile inside the duct also introduce the difficulty in achieving the required accuracy with a hot wire anemometer.

Although the nonuniform velocity profile introduced by the fan at the inlet was practically eliminated as the air passed through the rock bed, a sharp-edged orifice was installed in the exhaust duct as a primary measurement device. A sharp-edged orifice creates a pressure differential that can be accurately measured with a manometer. The operating principle of converting energy from kinetic to potential is less likely to be influenced by air irregular velocity profile than a hot wire anemometer. The air passing through a rock bed tends to act as a flow straightener and was considered uniform enough for sharp-edged orifice flow measurement. An acrylic sharp edged orifice (Figure 30) with 0.39 meter outside and 0.31 meter inside diameters was constructed. The plate was installed 1.8
Figure 29. Rockbed front view

Figure 30. Sharp edged orifice for air flow measurement
meter from rock bed outlet. Air pressure taps were located at .39 m before and .14 m after the orifice plate. Air flow rates were obtained by measuring the pressure drop across the sharp edged orifice. Back up measurements were taken with a hot wire anemometer.

The 35 thermocouple temperature measuring grid was located on a diagonal plane of the rock bed (Figure 31) to expected symmetric characteristics of the rock bed temperature profile. An iron wire grid was constructed inside the rock bed to provide support for the thermocouples. Exact positioning of each thermocouple was accomplished by attachment to the wire grid before the rock was placed in the module. All thermocouples were connected to two programmable recording potentiometers equipped with digital display, folded paper printout, and perforated paper tape output for easy interfacing with an optical reader and computer manipulation.

**Rock bed System Operation**

The rock bed thermal storage module was operated from August 22, 1984 to September 17, 1984. Various modes of operation were conducted to determine specific operational capabilities (Table 8). The experimental modes tested were:

1. **Continuous operations**

   The system was operated under normal weather conditions for a relatively long period (3-5 days). Charging and discharging procedures were repeated during the experiment. Overall module performance and thermal lags under different weather conditions were evaluated.

2. **Discrete short term dynamic change operation**

   Charge and discharge procedures were altered periodically to create several discrete operation modes. The discrete operations emphasized monitoring module response to
Table 8. Test module system operation modes

<table>
<thead>
<tr>
<th>Operation Mode</th>
<th>Duration</th>
</tr>
</thead>
<tbody>
<tr>
<td>Continuous</td>
<td>8/24 22:00 - 8/27 09:00</td>
</tr>
<tr>
<td></td>
<td>8/29 12:40 - 8/31 10:40</td>
</tr>
<tr>
<td></td>
<td>9/05 22:00 - 9/09 09:10</td>
</tr>
<tr>
<td>Discrete</td>
<td>8/27 15:00 - 8/28 09:35</td>
</tr>
<tr>
<td></td>
<td>8/28 13:00 - 8/29 08:00</td>
</tr>
<tr>
<td></td>
<td>9/04 21:00 - 9/05 21:00</td>
</tr>
<tr>
<td>Steady state</td>
<td>8/28 09:35 - 8/28 13:00</td>
</tr>
<tr>
<td></td>
<td>9/14 08:40 - 9/14 13:00</td>
</tr>
<tr>
<td></td>
<td>9/15 09:30 - 9/16 16:50</td>
</tr>
</tbody>
</table>
individual charging and discharging, under different initial conditions. System dynamic response to a sudden change in ambient conditions was studied to analyze how fast the module could respond. Reducing temperature variation as well as limiting maximum temperature in the broiler space is desired during hot weather. Maximum thermal stress occurs when ambient temperature increases rapidly. System dynamic response analyses with respect to ambient temperature variations are essential in scheduling ventilation that utilizes the rock bed thermal storage module to condition air brought into the broiler space.

3. Steady state operation

The rock bed was charged to an equilibrium condition (either high or low temperature profile) in which the temperatures were approximately equal throughout. The fan was then turned off and the rock bed left undisturbed. Any temperature gradient that exists between individual rock particles and the adjacent air will induce buoyancy motion. This buoyancy induced natural convection becomes the driving force for an temperature balance with ambient.

The total potential of the rock bed to preserve current thermal conditions under different environmental impacts was analyzed.

**Microcomputer Assembly and Modification**

The purchased microcomputer kit needs to be assembled and tested before putting into control system. A total of 16 hours was required to completely assemble the entire ET-3400 microcomputer system. The system was tested to insure proper operation prior to connect the microcomputer to peripheral interface adapter, motor control and temperature sensing circuits. System test includes information
display data input/output, and memory function was done by continuously running a 12 hour clock program written in machine language for two days. The clock program accessed initial time values in the registers and add internal clock frequency to the register which leads to the next displayed time. The updated time was then displayed each second. Testing the assembled unit was accomplished without proper logic and analog analyzers. Further chips interchanging was proceeded after the malfunction IC chips were identified. The TTL IC chips used in the project are sensitive to the proper power supply. A Stable DC power source was provided by a 5 and 12 volt supply circuit to the printed circuit board. The transformer generates excess heat which must be dissipated to the environment from a heat sink. Limited space under the face board caused electronic components to touch transformer wires and accumulate heat. The high temperature environment caused the voltage regulators to malfunction and provide unstable power. The power supply wires were rearranged to eliminate contact, components were spaced to allow sufficient cooling and a damaged voltage regulator was replaced. The microcomputer system was connected to the PIA, temperature sensing, and stepping motor control circuits after system test was completed. The control algorithm developed for the rock bed ventilation module was then applied to the completed system.

**Microprocessor Control**

The microprocessor control formulated by interfacing the temperature sensing and stepping motor control circuits to the microcomputer system was tested to insure desired operation. The microprocessor control tests included hardware verification and software debugging. The pseudo-code programs written in Motorolla MC 6808 assembly language were translated into machine language (binary code) due to the lack of a compiler and cross assembler. The
executable instructions were entered through a hexadecimal keyboard and stored in the RAMs.

Hardware verification included thermocouple data acquisition interfacing and stepping motor control bit pattern generation. Each was tested individually according to operational tasks. Integrated control operation test combining software and hardware were performed after individual tests were completed. Integrated data acquisition and motor control testing procedures were accomplished through the PIA configuration. The PIA which is functionally divided into a central process side and peripheral side as shown earlier in Figure 17, is a programmable neutral device which will not perform either input or output functions before receiving a control command issued by the software. Port A of the PIA 6821 was configured as the output for motor control and analog temperature signal multiplexing. Port B was configured as the input device for the A/D converter. The data flow and function of the PIA were configured by the Software requests. The corresponding control step was proceeded as shown in Figure 32.

The chip select lines (CS\_0, CS\_1, CS\_2) are used to access PIA. The operation of the PIA and associated circuits can be described by Figure 33. A high signal (logic 1) was send to CS\_0 and CS\_1 with a low signal (logic 0) to CS\_2. The internal registers become active after PIA is selected. Port A and port B were then selected by presenting a logic 0 at RS\_1 line or a logic 1 at RS\_1, respectively. Once the port selections were completed, a logic 1 was sent to the RS\_0 for port control register selection before the data flow direction can be decided. The data direction was selected first by keeping bit 2 of control register at low state (logic 0) and sending a combination of binary codes to the registers according to the
Figure 32. Peripheral interface adapter hierarchical control
Figure 33. PIA 6821 and associated circuit layout
desired input and output direction. The data transfer between CPU, temperature sensing, and motor control circuits is then proceeded following the PIA configuration.

**Temperature Sensing Circuits**

The test of temperature sensing circuits involved many software and hardware modification efforts. The precision temperature sensors used in the thermocouple cold junction compensation circuit were calibrated first. The manufacturer suggested single point calibration was performed. The sensor has a linear output which is proportional to absolute temperature with extrapolated sensor output equal 0 volt at 0 °K(-273.15 °C). Inaccuracy over the full operational temperature range therefore can be calibrated with the single point calibration. A second adjustment was made on the temperature compensation reference for the copper-constantant type thermocouple. A single type T thermocouple was connected to the differential analog voltage input port in ADC 0804 LCN analog to digital converter. A temperature-voltage table was reconstructed for the temperature range from 0 to 50 degree Celsius as shown in Appendix I. The calibrated results indicate a straight line can be used to approximate the temperature to digital conversion table. Since the T type thermocouple temperature to voltage line passes through the origin, therefore no offset is needed. Operational amplifier adjustment was emphasized on the upper end of thermocouple output. The calibrated 50 degree Celsius equivalent to a 2.02 millivolt output was amplified to binary number 11111111 by adjusting the gain of operational amplifier. The readjusted circuits functioned properly during the later test.
**Damper Operation**

The end device for the microprocessor control system was a pair of dampers used to regulate air flow into the conditioned broiler space (Figure 34). Outside air and sensibly cooled air from the rock bed thermal storage module were proportionally mixed to the desired temperature before entering the broiler house. The proper mixing ratio was obtained from the basic energy balance between outside, sensibly cooled, and desired air temperature as:

\[ \delta E_{oa-ca} + \delta E_{rb-ca} = \delta E_{sp-ca} \]  

(90)

where

\[ \delta E_{oa-ca} = \text{energy difference between outside and inside air} \]
\[ \delta E_{rb-ca} = \text{energy difference between sensibly cooled and inside air} \]
\[ \delta E_{sp-ca} = \text{energy difference between desired and inside air} \]

Damper angle arrangement was calculated from the following approaches which were derived from the above energy balance.

\[ C_p \cdot V_1 (T_{out} - T_c)/\rho_1 + C_p \cdot V_2 (T_{rock} - T_c)/\rho_2 \]

\[ = C_p \left( \frac{V_1}{\rho_1} + \frac{V_2}{\rho_2} \right) (T_{sp} - T_c) \]  

(91)

where

\[ T_{out} = \text{outside air temperature, } ^\circ C \]
\[ T_c = \text{lower limit of conditioned space temperature, } ^\circ C \]
\[ T_{rock} = \text{air temperature at rock bed exit, } ^\circ C \]
\[ T_{sp} = \text{desired set point temperature, } ^\circ C \]
\[ V_1 = \text{volume of outside air, } m^3 \]
Figure 34. Simplified rockbed system operation diagram
\[ V_2 = \text{volume of air from rock bed, } m^3 \]

\[ C_{pi} = \text{specific heat of air at each state, } J/kg.K \]

\[ \rho_i = \text{density of air at each state, } kg/m^3 \]

The expected rock bed operation temperature range was from 13 °C to 30 °C. The density and specific heat of ambient and sensibly cooled air were assumed equal which allowed equation 90 to be simplified to:

\[ m_1(T_{out} - T_c) + m_2(T_{rock} - T_c) = (m_1 + m_2)(T_{sp} - T_c) \] (92)

or

\[ \frac{m_1}{m_1 + m_2}(T_{out} - T_c) + \frac{m_2}{m_1 + m_2}(T_{rock} - T_c) = T_{sp} - T_c \] (93)

where

\[ m_1, m_2 = \text{air flow mass of outside and sensibly cooled air, respectively, kg/s} \]

The air mass flow rate was regulated by the dampers as shown in Figure 34. The elliptical area occupied by each damper (when oriented at any angle other than 0 degree or 90 degree) is equal to \((a^2-b^2)/a\) (a is the major axis and b the minor axis). Ventilation air must pass through an area equal to the difference between the circular area of the duct and the elliptical area occupied by the damper projection. Thus the area available for air flow is equal to \(nr^2(1-\cos \theta)\), where \(r\) equals the pipe radius and \(\theta\) is the damper angle from a vertical plane inside the pipe. The mass flow of air through the pipe is therefore proportional to the available
cross-sectional area (i.e., \(m = a \cdot A \cdot a(1 - \cos\theta)\)). The total air flow is simply the sum through each duct. Damper rotation angle was derived from the following relationship:

\[
\pi r^2 (1 - \cos\theta_1) + \pi r^2 (1 - \cos\theta_2) = \pi r^2
\]

(94)

where

\[\theta_1 = \text{damper angle in outdoor air pipe, degree}\]
\[\theta_2 = \text{damper angle in sensibly cooled air pipe, degree}\]

Substitution of the above relationship into Equation 92 produced:

\[
(1 - \cos\theta_1)T_{\text{out}} + (1 - \cos\theta_2)T_{\text{rock}} = (m_1 + m_2)T_{\text{sp}}
\]

(95)

or

\[
\sin\theta_1 T_{\text{out}} + \cos\theta_1 T_{\text{rock}} = (\sin\theta_1 + \cos\theta_1)T_{\text{sp}}
\]

(96)

The required damper rotation angle was then represented by

\[
\theta_1 = \tan^{-1}\left(\frac{T_{\text{sp}} - T_{\text{rock}}}{T_{\text{out}} - T_{\text{sp}}}\right)
\]

(97)

The analog signals representing the temperatures sensed by the three thermocouples identified as \(T_{\text{out}}\), \(T_{\text{rock}}\), and \(T_{\text{sp}}\) in Figure 34 were gated through the multiplexer and stored as binary form in one of the RAM's. A vertically positioned damper was selected to represent step zero of the stepping motor. Damper angle was calculated with the software developed for the system by solving Equation 97 for \(\theta_1\). The difference between current calculated angle and the previous angle was converted into steps required to move from the previous position to the new desired position. The backward or forward movement of dampers then provide optimum air mixture for the given temperature condition.

The damper control algorithm was tested with and without Advance
Micro Device (AMD) arithmetic chip. Both methods required a degree-step table as shown in Table 9. The step indicators (S.I in Table 9) represent the number of steps which are required to move from the original position to a specific angle $\theta$. A theta and correspondent step indicator are located after the processing of incoming temperatures. By evaluating the difference between current step indicator and previous step indicator, the steps required to advance to the next damper position can be found.

The step calculation is straightforward using AMD arithmetic chip's trigonometry function. Since the microprocessor does not have the capability to handle fraction number directly, a special treatment is required to perform this calculation in the absence of an arithmetic chip. Table 9 can be grouped into three parts according to the characteristics of $\tan \theta$. Group I can be converted to distinguishable integer by inverting $T_{sp}-T_{rock}$ and $T_{out}-T_{sp}$. The step indicator can be located by matching the above number and $1/\tan \theta$. The swapping of numerator and denominator in Equation 96 was done by swapping the microprocessor registers that contain the temperature information.

The inversion approach did not provide enough information for step separation in group II. Another treatment was applied to obtain the required steps. The $\tan \theta$ values were multiply by ten using consecutive addition in microprocessor operations. The integer part was then used to correlate the step indicator. In some cases, $\tan \theta$ has the same integer part with the others. The first indicator was chosen in the case of two $\tan \theta$ have the same integer part. The middle indicator was selected in the case of three $\tan \theta$ have the same integer part. The integer part in the group III can be used directly to match the step indicator. The maximum error in locating
the step indicator is one (1.8 degree in full step operation) in all three cases and was considered acceptable for this type of control.

Table 9. Lookup table for damper operation.

<table>
<thead>
<tr>
<th>S.I.</th>
<th>$\theta$</th>
<th>$\tan \theta$</th>
<th>$\tan^{-1} \theta$</th>
<th>$10\tan \theta$</th>
<th>Integer</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
<td>0</td>
<td>-</td>
<td>31</td>
<td>31</td>
</tr>
<tr>
<td>1</td>
<td>1.8</td>
<td>0.031</td>
<td>31.82</td>
<td>31</td>
<td>31</td>
</tr>
<tr>
<td>2</td>
<td>3.6</td>
<td>0.063</td>
<td>15.89</td>
<td>15</td>
<td>15</td>
</tr>
<tr>
<td>3</td>
<td>5.4</td>
<td>0.095</td>
<td>10.58</td>
<td>10</td>
<td>10</td>
</tr>
<tr>
<td>4</td>
<td>7.2</td>
<td>0.126</td>
<td>7.92</td>
<td>7</td>
<td>7</td>
</tr>
<tr>
<td>5</td>
<td>9</td>
<td>0.158</td>
<td>6.31</td>
<td>6</td>
<td>6</td>
</tr>
<tr>
<td>6</td>
<td>10.8</td>
<td>0.191</td>
<td>5.24</td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td>7</td>
<td>12.6</td>
<td>0.224</td>
<td>4.47</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td>8</td>
<td>14.4</td>
<td>0.257</td>
<td>3.89</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>9</td>
<td>16.2</td>
<td>0.291</td>
<td>2.91</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>10</td>
<td>18</td>
<td>0.325</td>
<td>3.25</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>11</td>
<td>19.8</td>
<td>0.360</td>
<td>3.60</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>12</td>
<td>21.6</td>
<td>0.396</td>
<td>3.96</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>13</td>
<td>23.4</td>
<td>0.433</td>
<td>4.33</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td>14</td>
<td>25.2</td>
<td>0.471</td>
<td>4.71</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td>15</td>
<td>27</td>
<td>0.510</td>
<td>5.10</td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td>16</td>
<td>28.8</td>
<td>0.550</td>
<td>5.50</td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td>17</td>
<td>30.6</td>
<td>0.591</td>
<td>5.91</td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td>18</td>
<td>32.4</td>
<td>0.635</td>
<td>6.35</td>
<td>6</td>
<td>6</td>
</tr>
<tr>
<td>19</td>
<td>34.2</td>
<td>0.680</td>
<td>6.80</td>
<td>6</td>
<td>6</td>
</tr>
<tr>
<td>20</td>
<td>36</td>
<td>0.727</td>
<td>7.27</td>
<td>7</td>
<td>7</td>
</tr>
<tr>
<td>21</td>
<td>37.8</td>
<td>0.776</td>
<td>7.76</td>
<td>7</td>
<td>7</td>
</tr>
<tr>
<td>22</td>
<td>39.6</td>
<td>0.827</td>
<td>8.27</td>
<td>8</td>
<td>8</td>
</tr>
<tr>
<td>23</td>
<td>41.4</td>
<td>0.882</td>
<td>8.82</td>
<td>8</td>
<td>8</td>
</tr>
<tr>
<td>24</td>
<td>43.2</td>
<td>0.939</td>
<td>9.39</td>
<td>9</td>
<td>9</td>
</tr>
<tr>
<td>25</td>
<td>45</td>
<td>1.000</td>
<td>10.00</td>
<td>9</td>
<td>9</td>
</tr>
<tr>
<td>26</td>
<td>46.8</td>
<td>1.065</td>
<td>10.65</td>
<td>10</td>
<td>10</td>
</tr>
<tr>
<td>27</td>
<td>48.6</td>
<td>1.134</td>
<td>11.34</td>
<td>11</td>
<td>11</td>
</tr>
<tr>
<td>28</td>
<td>50.4</td>
<td>1.209</td>
<td>12.09</td>
<td>12</td>
<td>12</td>
</tr>
<tr>
<td>29</td>
<td>52.2</td>
<td>1.289</td>
<td>12.89</td>
<td>12</td>
<td>12</td>
</tr>
<tr>
<td>30</td>
<td>54</td>
<td>1.376</td>
<td>13.76</td>
<td>13</td>
<td>13</td>
</tr>
<tr>
<td>31</td>
<td>55.8</td>
<td>1.471</td>
<td>14.71</td>
<td>14</td>
<td>14</td>
</tr>
<tr>
<td>32</td>
<td>57.6</td>
<td>1.576</td>
<td>15.76</td>
<td>15</td>
<td>15</td>
</tr>
<tr>
<td>33</td>
<td>59.4</td>
<td>1.691</td>
<td>16.91</td>
<td>16</td>
<td>16</td>
</tr>
<tr>
<td>34</td>
<td>61.2</td>
<td>1.819</td>
<td>18.19</td>
<td>18</td>
<td>18</td>
</tr>
<tr>
<td>35</td>
<td>63</td>
<td>1.963</td>
<td>19.63</td>
<td>19</td>
<td>19</td>
</tr>
<tr>
<td>36</td>
<td>64.8</td>
<td>2.125</td>
<td>21.25</td>
<td>21</td>
<td>21</td>
</tr>
<tr>
<td>37</td>
<td>66.6</td>
<td>2.311</td>
<td>23.11</td>
<td>23</td>
<td>23</td>
</tr>
<tr>
<td>38</td>
<td>68.4</td>
<td>2.526</td>
<td>25.26</td>
<td>25</td>
<td>25</td>
</tr>
<tr>
<td>39</td>
<td>70.2</td>
<td>2.778</td>
<td>27.78</td>
<td>27</td>
<td>27</td>
</tr>
<tr>
<td>40</td>
<td>72</td>
<td>3.078</td>
<td>30.78</td>
<td>30</td>
<td>30</td>
</tr>
<tr>
<td>41</td>
<td>73.8</td>
<td>3.442</td>
<td>34.42</td>
<td>34</td>
<td>34</td>
</tr>
<tr>
<td>42</td>
<td>75.6</td>
<td>3.895</td>
<td>38.95</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>43</td>
<td>77.4</td>
<td>4.474</td>
<td>44.74</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td>44</td>
<td>79.2</td>
<td>5.242</td>
<td>52.42</td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td>45</td>
<td>81</td>
<td>6.314</td>
<td>63.14</td>
<td>6</td>
<td>6</td>
</tr>
<tr>
<td>46</td>
<td>82.8</td>
<td>7.916</td>
<td>79.16</td>
<td>7</td>
<td>7</td>
</tr>
<tr>
<td>47</td>
<td>84.6</td>
<td>10.579</td>
<td>105.79</td>
<td>10</td>
<td>10</td>
</tr>
<tr>
<td>48</td>
<td>86.4</td>
<td>15.894</td>
<td>158.94</td>
<td>15</td>
<td>15</td>
</tr>
<tr>
<td>49</td>
<td>88.2</td>
<td>31.819</td>
<td>318.19</td>
<td>31</td>
<td>31</td>
</tr>
</tbody>
</table>
RESULTS AND DISCUSSION

The rock bed thermal storage unit was operated during August and September of 1984. The operation period corresponded to common commercial broiler brooding period number four and five in Table 3. Ambient conditions experienced during the test period are comparable to the 1982 and 1983 weather data used in the simulation.

Development Stage Modification

Close inspection of the initial test results for the rock bed thermal storage unit revealed that two adjustments were needed. A portion of the air passing through the rock bed was bypassing the rock through the small space left as an air film between the insulation blanket and outside plywood. Four wooden blocks were installed along each side of the box to force the air to flow into the gravel section. Another minor air leakage was corrected with additional joint sealant. The inlet temperature monitoring thermocouples were installed too close to the fan motor. The radiative heat from motor affects the termocouple, and therefore bias the temperature reading. A 1.5 degree temperature difference was found between the ambient air and the sampling point. A plywood shade was installed to prevent direct sunshine on the inlet and the inlet temperature sensing thermocouples were relocated away from motor.

Overall Rock Bed Thermal Storage Module Performance

The rock bed thermal storage module was tested under three different operational modes. Overall system performance evaluated using the continuous operation mode from August 24 to August 30 (Figure 35), indicated that a four hour thermal lag could be readily achieved when ambient air temperatures experienced a 15 to 20 degree Celsius diurnal temperature variation. The recorded data show that
Figure 35. Thermal lag of rockbed outlet temperature
Figure 36. Time-temperature rate of change of rockbed system
the interior points maintained a constant temperature time rate of change from one time step to the next (Figure 36). The interior point temperature time rate of change varied only when a heat front reached that location. An air temperature change at the rock bed inlet induces a heat or cooling front to move through the rock bed. The slow rate at which front moved was verified by the thermal lag observed under high ambient temperature conditions. The average heat front transport rate ranged between .3 m/h to 0.45 m/h providing sufficient sensibly cooled air to condition the broiler house space during hot weather period. The daily average temperatures during the test operation period were closely related to the temperatures used for heat load simulations (Table 5). The observed peak cooling load occurred during broods number four or five agreed with the simulation results from the broiler house ventilation model.

The charge and discharge operation was stopped on August 29 at 0800 am PDT after the storage unit was fully charged. A full charge of the test module was judged as when the temperature reading from each thermocouple layer reached the possible average minimum and started to increase. Internal thermal stability was then investigated during the five hour shut off period (0800 to 1300). Recorded data indicated that a average of .5 degree C temperature fluctuation was experienced throughout the entire rock bed (Figure 37). The results presented two facts about the test module. Firstly, the module was able to retain the internal thermal condition under various external conditions. Secondly, the insulation level applied to the test module was adequate enough to minimize thermal leakage to the environment.

Assuming the maximum allowable temperature for a four week old broiler is 26 degree Celsius (79°F), the ventilation fan was turned on when the ambient temperature reached this limit. The fan started forcing the air into the rock bed. The shape of outlet temperature
Figure 37. Internal thermal stability test

Figure 38. Thermal lag operation test
curve does not follow the original discharge curve during this discharge cycle due to the different inlet conditions (Figure 38). Peak temperature of the discharge curve was lower than that of a test module if it was operated without disturbance. The outlet air retained the temperature curve as part of a complete undisturbed discharge temperature curve with additional six hours thermal lag and peak temperature decreasing. The time shift of the temperature curve was affected by changing the restart time or the discharge rate. Longer thermal lag can be obtained if a decrease in ventilation rate or a delay in restarting the system is permitted. Average thermal lag is expected to be longer than the experimental case since the system will not always be operated under the peak cooling load. An averaged ten to fifteen degree Celsius temperature difference between cooled air and ambient air was observed. The estimated twelve degree temperature difference used to design the rock bed volume proved to be adequate.

Sudden environmental temperature variations create operation scheduling problems for broiler house ventilation system. Uniformly conditioned air can be provided to the broiler house by utilizing the rock bed thermal storage unit as a thermal filtering device. The rock bed module responded to a series of dynamic ambient temperature change as shown in Figure 39. The ambient, mid-point, and outlet temperatures were monitored every 15 minutes. The plotted temperature curves show that the filtering process was completed within the first portion of the rock bed. Further investigation was made on a fully charged unit with temperature induced thermal buoyancy. Under one hour operation with .269 m/s air charging velocity, the first layer (.3m from the air inlet) rock responded with 1 to 1.5 degree C (8 percent) to the 15 to 20 degree C ambient temperature change. The second (3m to .6m from air inlet) and farther layer elements responded with a rate less than 4 percent.
Figure 39. Smoothing effect under inlet temperature variation
The results show that the test module is highly capable for smoothing the environmental temperature variation.

The ability to maintain the internal thermal condition under high temperature outside the rock bed storage unit was further tested during September 15 to September 16, 1984 (Figure 40). The fully charged rock bed was exposed to an ambient temperature variation between 5 degrees to 17 degrees Celsius. The first layer of thermocouple readings show the largest temperature variation correspondent to the ambient temperature change was less than 3 degree. The second and third layer temperature data showed a correspondent .5 degree variation. The results presented the designed rock bed will be able to conserve original thermal stratification during normal summer weather conditions for at least 24 hours. A fully charged rock bed thermal storage unit was found capable of providing 5 to 6 hours peak ventilation requirement after 24 hours thermal buoyancy energy balance. The rock bed operation should be scheduled so that available low ambient temperature can be used to charge rock bed whenever possible. The above experiment results indicates the rock bed can be charged to any thermal state and kept for later ventilation uses. This ability to preserve the energy for a long period of time is useful in the broiler house summer ventilation.

**Computer Simulation**

The thermocouple arrangement restricted the availability of initial temperature distribution. Only six point’s initial temperature were available directly from recorded data while more than six nodes were preferred in the simulation. Part of the recorded temperature data were plotted first as shown in Figure 41. The recorded data show that between any two temperature front change, the temperature distribution for a specific point tends to be linear.
Figure 40. Effects of natural convection inside rockbed
Figure 41. Adequacy test for using interpolated initial temperature
It is therefore adequate to estimate initial temperature between two known points.

A series analyses were performed for each of the system parameters listed in Table 7. Each parameter contributed partial influence in shifting the simulated temperature distribution curve backward or forward. The basic shape of the simulated temperature curve was mainly controlled by the governing partial differential equation derived in the mathematical modeling. The analyses showed that the system performance is not highly sensitive to some of the parameters, however, careful evaluation of system parameters is desired in obtaining a precise time-temperature relationship.

Simulated results and recorded data were plotted in Figure 42. The simulation predicted the time of inflection points (highest and lowest point in the curve) occurred. Accuracy was limited to 20 minute time step due to sampling interval set for the recorders. The simulation presented an 0.5 to 1.5 °C in predicting the outlet temperatures. The maximum differences 1.5 °C between simulated and recorded data occurred when abrupt or steeper ambient temperature change took place. Better results were obtained when the environment temperature experienced a smooth change (Figure 43).

The simulation result shown in Figure 42,43 can be used to describe the phenomenon which took place during each thermal charge and discharge cycle. The simulated curve matches the recorded rock bed outlet temperature curve until a non-smooth change in the ambient temperature occurred. The test module responded to the temperature change slower than the mathematically modeled rock bed. Possible reasons for the small simulation error were analyzed as:

1. Test module

The size of the inlet is small compared to the cross sectional area of the rock bed. The distance between the distribution fan and rock pack face should be large enough
Figure 42. Simulated vs. recorded temperature (rapid change)
Figure 43. Simulated vs. recorded temperature (mild change)
to provide evenly distributed air into the rock bed. The designed inlet area provided a space slightly less than .30 meter which could cause the development of radial temperature gradients.

2. Mathematical Model

Assumptions made for the rock bed thermal performance model included neglecting the effect of radial conduction. Associated temperature gradient were not included in the governing partial differential equations. Yagi et al. (1960) studied the steady state axial temperature gradient and concluded the change rate of effective axial thermal conductivity is about two to eight time faster than the radial effective thermal conductivity. Ambient air distributed into the rock bed thermal storage unit first interacts with those gravels located at the center of the rock bed. Slower interaction rates between incoming air and outer portion of the rock bed could develop a small temperature gradient through out the same layer. The slightly differences between estimated highest outlet temperatures and recorded data could come from radial heat conduction. A portion of energy increase was absorbed by the outer part gravel pack in the same layer during the stage of temperature increase. The same reason also accounts for the lower estimated minimum outlet temperatures compare to the real data. The test module outlet temperature distribution and heat front within the rock rock can be closely predicted by the simulation model with an error of less than 6 percent.
Circuit Construction and Operation Overview

All the circuits used in the project were constructed on breadboards with color coded jumper wires. The circuits built on the breadboards required more space than a standard etching PC board but the color coded jumper provided easy circuit function organization which was convenient in the hardware debug and modification.

During the experimental stage, the assembled microprocessor control system had revealed the following facts that most of them were related to the hardware limitation.

Timing problem associated with integrated tested circuit was experienced. The sequences needed to complete a temperature sensing procedure include channel selection, analog signal transmission, A/D conversion digital signal output to other external devices.

The original designed temperature sensing circuits was simplified as the functional block shown in Figure 34. for easy identification. A logic 0 signal sending out from PIA is connected to the corresponding control pins in A/D converter and multiplexer. The A/D converter interprets the logic 0 at control pin as a start conversion command and starts to convert the analog voltage signal that currently available at the the analog input port. The same logic 0 signal is sent to multiplexer which enable the channel selection. Average 720 ns propagation time is required to transmit analog signal from selected channel through multiplexer to the A/D converter. The analog signal converted by A/D converter is therefore different from the analog signal that channel through multiplexer if consider the associated time in both chips. The sensed temperature equivalent voltage was therefore not processed under this condition.

The cost associated with a additional 15 volt power supply rejected the possibility to adopt a sample and hold circuit. Satisfactory operation was obtained through the use of two NAND gates
connected in series between the PIA and A/D converter start conversion control pin to provide the required time delay and control pulses.

**Stepping Motor Control Circuit**

The stepping motor control was done by both hardware and software bit pattern generation method. The hardware bit pattern generation requires two bits of control signal from CPU to trigger the operation. Two bit signal transfer can be completed with one byte data. The operation of the stepping motor is therefore a procedure of calculating required steps and sending out control byte contains motor rotation direction and steps. The simplicity of the hardware bit pattern generation is useful since the tested microprocessor control module has a limited less than one kilobyte memory available for the control program. Certain disadvantage such as missing steps have found to be associated with the hardware bit pattern generation method. The hardware bit pattern generation method is accomplished by pre-arranged gating operation. The more electronic gate an information signal passes the more chance occur that a noise signal will occur. Noise introduces possible missing or extra step of motor operation which lead to false damper operation.

Software bit pattern generation provides additional flexibility in the motor control. The motor rotation can be switched between half step (0.9 degree/step) and full step (1.8 degree/step) by indexing the program pointer to appropriate lookup table. Exact bit pattern is obtained from table and sent to stepping motor directly through PIA which eliminates the possible noise interference. Two bit pattern lookup tables take a portion of available memory space and make the integrated operation from temperature sensing, A/D conversion, and motor control difficult to perform. Separate test on
bit pattern generation showed reliable result with software method which encourages the motor control of this kind to be used if the system can add on more memory.

**Damper Operation**

The damper displacement was designed so the stepping motor rotation was related to damper directly. The test of the rotation angle was done by a set of LED index and a circular paper marked with tics for every 1.8 degree. The only disadvantage for this simple angle display device was the highly concentration required for the operator. Even though the angle displacement is not a factor to be monitored after the motor control is fully tested, a circuit adopted from Lou (1976) with some modifications was built for easier, clear, and fast reading.

The microprocessor was successful in controlling the stepping motor used to adjust the mixture of outside and sensibly cooled air needed to maintain the set-point temperature in the conditioned space. The calculated damper angle required for the appropriate air mixture did not necessarily fall directly on one of the fixed rotation angle of the stepping motor. A maximum error of 2.2 percent in positioning the damper was experienced under full step operation. Higher resolution (.9 degree/step) can be achieved with the half step operation and provides more accurate air flow control for the ventilation system if necessary.
SUMMARY AND CONCLUSIONS

An rock bed thermal storage unit utilized the river run gravels as the storage medium for the broiler house summer ventilation was designed, constructed, and tested. The design concept was based on the utilization of large diurnal difference between daily high and low temperatures under summer weather condition of Corvallis, Oregon. The size of the rock bed was obtained by running a broiler house ventilation model under historic Oregon weather data. Peak ventilation requirement was used as the sizing reference for the rock bed. Insulations was provided under the consideration of efficiency and expense. The rock bed test was focused on: 1) Continuous operation for instantaneous system response. 2) Discrete type operation for test of long term energy storage capability.

Finite difference method was applied to the two phase coupled governing partial differential equations of the rock bed. High flexibility was incorporated into the computer simulation program written for the mathematical model. The initial conditions can be provided by either last state temperature distribution from other program results, measured discrete temperature distribution or automatically generated by the program. The temperature input can be a continuous function from any collector panel or discrete weather data. Program was expected to predict the internal temperature distribution and outlet air temperature under different ambient temperature changes and operation schedules.

Initial control experiments were also made for the broiler house summer ventilation that utilized the experimental rock bed build for the project. A microcomputer incorporated Motorolla MC 6808 microprocessor was assembled and tested to interface with the control components build for the stepping motor and temperature sensing devices. Control components included stepping motor bit pattern generation circuit, thermocouple to microprocessor interface.
circuits, and basic external interface I/O circuits.

The conclusions about the system performance came from compiled results and experiment debug experiences was categorized as:

1) The expense to build the rock bed thermal storage unit is approximately 95 dollars excludes the expense of ventilation fan.

2) The rock bed thermal storage unit is capable to provide at least 90 percent of the broiler house peak summer ventilation requirements under normal weather conditions.

3) Ten rock bed storage units with dimension of 2.4 m by 1.8m by 1.2m high can store enough energy to provide the sensible cooled ventilation air for 8000 mature broilers.

4) The system is capable to maintain a ten to fifteen degree temperature difference between ambient and conditioned air.

5) A five hour delay of the ambient air temperature can be easily achieved if the system is continuously operated for a diurnal cycle.

6) More than 7 hour temperature delay is available to the broiler house ventilation with a preset conditioned space temperature 26 degree C.

7) The fully charged rock bed thermal storage unit is capable to maintain internal thermal distribution for more than 24 hours.

8) The numerical solution to the rock bed thermal storage governing partial differential equations can provide a .5 to 2 degree accuracy in predicting the rock bed outlet temperature.
9) The response time to any discrete or continuous temperature changes at the rock bed inlet can be predicted accurately by the numerical simulation.

10) The numerical solution can provide valuable references to the design of any ventilation system that utilizes the gravel as the energy storage medium.

11) Sophisticated industrial stepping motor control is not necessary for the low speed damper operation.

12) Ten of the stepping motor control circuits and damper set up are required to cover the ventilated space for 8000 mature broilers.

13) The stepping motor, motor control circuit, and accessories cost 40 dollars approximately.

14) The use of microprocessor development system is recommended for sophisticated ventilation control algorithms.

15) Single chip microcomputer and erasable programmable read only memory (EPROM) should be used to provide more flexible and less expensive ventilation controllers.

16) The use of thermocouple for the temperature sensing devices create a lot of extra difficulties due to the required cold junction compensation.

17) Temperature sensor with on-chip compensation should be used to simply design tasks and limit some possible errors.

18) The number of temperature sensing circuits required is depended upon the ventilation schedule. Each temperature sensing circuit cost about 6 dollars.
19) The printed circuit board (PC board) technique with a small initial cost should be used to simplify the unnecessary wiring and preventing the possible shorted circuit problem.

20) The possible signal noises from the long thermocouple wires should be prevented if the control circuits will be used in the commercial broiler house.

The implementation of rockbed thermal storage module, microprocessor control ventilation system, and simulation program present a fairly satisfactory overall performance. For the later continuing researches, more efforts should be made to improve the computer simulation accuracy. Microcomputer control systems that incorporate sophisticated control algorithms and better electronic devices should be explored.
BIBLIOGRAPHY


Broiler Industry. 1980 March.


APPENDICES
APPENDICES

APPENDIX A. Broiler House Ventilation Model

PROGRAM BROILER(INPUT, OUTPUT, TAPE5, TAPE6, TAPE7, TAPE8)
DIMENSION VRM(49,4), VRH(49,4), EXNH3(49,4), QSUP(49,4),
DIMENSION QSUP1(49,4), WVPEx2(49,4)
DIMENSION WVPEx(49,4), VENTNH3(49,4), VR(49,4),
DIMENSION QSUP2(49,4)
READ(6,*)RWALL, RPER, RROOF, RFOUND, AWALL, PER, AROOF, NUMBIRD
URPOOL=0
TANPOO=0
DO 100 J=1,4
  READ(5,7)TO, PHIOUT
  7 FORMAT(F2.0,1X,F4.2)
CALL PSYCHRO(TO, PHIOUT, HO, WO, VO, HFGO)
DO 150 I=1,49
  PHIIN=.6
  IF(I.LE.7)TI=82
  IF(I.GT.7.AND.I.LE.14)TI=79
  IF(I.GT.14.AND.I.LE.21)TI=76
  IF(I.GT.21.AND.I.LE.28)THEN
    TI=73
    PHIIN=.7
  ENDIF
  IF(I.GT.28)THEN
    TI=70
    PHIIN=.7
  ENDIF
  TIA=TI+459.69
  CALL PSYCHRO(TI, PHIIN, HI, WI, VI, HFGI)
  RHPRIME=((14.696*WO)/(.6219+WO))/EXP(54.6329-12301.688/TIA-&5.16923*ALOG(TIA))
  CALL PSYCHR0(TI, RHPRIME, HI, A1, A2, A3)
  WT=39.8744+2.50313*I+1.04057*I**2-.0127234*I**3+.000073311*I**4
  FACTOR=NUMBIRD*WT/(60*454)
  IF(WT.LE.112)GO TO 10
  IF(WT.LE.664)GO TO 20
  IF(WT.LE.886)GO TO 30
  GO TO 40
  10 QL=(3.3E-5*WT**3-.0123*WT**2+1.47*WT-33.9)*FACTOR
  GO TO 50
  20 QL=(-1.23E-7*WT**3+.0002*WT**2-.111*WT+32.9)*FACTOR
  GO TO 50
  30 QL=(-.004054*WT+14.06189)*FACTOR
  GO TO 50
  40 QL=(3.69E-12*WT**4-1.9E-8*WT**3+3.752E-5*WT**2-.0382*WT+25.8)
  &*FACTOR
  QS=(15.9-.0143*WT+.496E-6*WT**2+.02E-9*WT**3-.292*WT-83.3)*FACTOR
  GO TO 80
  50 IF(WT.LE.618)GO TO 60
  IF(WT.LE.886)GO TO 70
  QS=(5*A1OG10(WT)+.375E-10*WT**4-7.76E-7*WT**3+.637E-4*WT**2-
  &.292*WT-.83.3)*FACTOR
  GO TO 80
  60 QS=(27.1438-.0974*WT+.2174E-3*WT**2-.2191E-6*WT**3+.7841E-10*
  &WT**4)*FACTOR
  70 WVP=QL/HFGI
  80 WVP=QL/HFGI
H2OMA = WVP / (WI - W0)  
QV = (HIP - HO) * H2OMA  
DELT = TI - TO  
AFOUND = 2 * PER  
AOVERR = AWALL / RWALL + PER / RPER + AROOF / RROOF + AFOUND / RFOUND  
QB = AOVERR * DELT / 60  
QM = 0  
QSUP(I, J) = QV + QB - QM - QS

c
QVH = QS + QM - QB  
HEATMA = QVH / (HIP - HO)  
VRH(I, J) = HEATMA * VI  
VRM(I, J) = H2OMA * VI  
IF (VRM(I, J) .GT. VRH(I, J)) THEN  
VENT = VRM(I, J) / 2  
H2OMA1 = VENT / VI  
QV1 = (HIP - HO) * H2OMA1  
QSUP1(I, J) = QV1 + QB - QM - QS  
WVPEX(I, J) = H2OMA1 * (WI - W0)
ELSE
QSUP1(I, J) = 0  
WVPEX(I, J) = 0  
VR(I, J) = VRH(I, J)  
QSUP2(I, J) = 0  
WVPEX2(I, J) = 0
GO TO 150
ENDIF
QLAM1 = WVPEX(I, J) * HFGI  
IF (QLAM1 .LT. QSUP1(I, J)) GO TO 815
XMA = H2OMA1
805 XMA = XMA + 0.1  
WVPEX2(I, J) = (H2OMA - XMA) * (WI - W0)  
QV2 = XMA * (HIP - HO)  
QSUP2(I, J) = QV2 + QB - QM
DIFF = WVPEX2(I, J) * HFGI - QSUP2(I, J)  
IF (ABS (DIFF) .LT. 5.0) GO TO 810
GO TO 805
810 VR(I, J) = XMA * VI  
GO TO 150
815 VR(I, J) = VRM(I, J) / 2  
QSUP2(I, J) = QSUP1(I, J)  
WVPEX2(I, J) = WVPEX(I, J)
150 CONTINUE
IF (J .EQ. 1) N = 50  
IF (J .EQ. 2) N = 77  
IF (J .EQ. 3) N = 63  
IF (J .EQ. 4) N = 49
DO 125 I = 1, N  
WT = 39.8744 + 2.50313 * I + 1.04057 * I ** 2 - .0127234 * I ** 3 + .00073311 * I ** 4  
IF (I .LE. 49) THEN  
URPROD = 3 * NUMBIRD * WT * 1.16 / (1000 * 454)
ELSE
URPROD = 0.0
ENDIF
URPOOL = URPOOL + URPROD  
IF (J .EQ. 1 .AND. I .LE. 49) THEN  
A = (0.0357 * I) ** 8  
PH = (2.7 * A) / (1 + A) + 5.8
ELSE
PH = 8.5
ENDIF
RATEPH = 1.34 * PH - 7.2
RATE9 = 1.3490 - 7.2
FACTPH = RATEPH / RATE9
T = 70
IF (I .LE. 28) T = 73
IF (I .LE. 21) T = 76
IF (I .LE. 14) T = 79
IF (I .LE. 7) T = 82
IF (I .GT. 49 .AND. J .EQ. 1) T = 52.9
IF (I .GT. 49 .AND. J .EQ. 2) T = 40.9
IF (I .GT. 49 .AND. J .EQ. 3) T = 42.5
TC = (T - 32) / 1.8
RATE = EXP (0.165 * (TC - 10) + 1.8)
RATE35 = EXP (0.165 * (35 - 10) + 1.8)
FACTOR = RATE / RATE35
FACTORS = FACTOR * FACTPH
TAN = FACTORS * 0.2 * URPOOL / 3
URPOOL = URPOOL - (TAN * 3)
TANPOOL = TANPOOL + TAN
PKA = 10.0819 - 0.0361195 * TC + 0.000105238 * TC ** 2
XKA = 1 / 10 ** PKA
HPLUS = 1 / 10 ** PH
FRAC = XKA / (HPLUS + XKA)
VOL = FRAC * TANPOOL
TANPOOL = TANPOOL - VOL
IF (I .GT. 49) THEN
VRH(I,J) = 0.0
VRM(I,J) = 0.0
VENTNH3(I,J) = 0.0
EXNH3(I,J) = 0.0
GO TO 125
ELSE
VENTNH3(I,J) = VOL / (24 * 60 * 1.0455E-6)
XREQ = VENTNH3(I,J) - VR(I,J)
IF (XREQ .LT. 0) XREQ = 0
EXNH3(I,J) = XREQ * 1.0455E-6
ENDIF
VRM(I,J) = VRM(I,J) / 35.29
VRH(I,J) = VRH(I,J) / 35.29
VENTNH3(I,J) = VENTNH3(I,J) / 35.29
VR(I,J) = VR(I,J) / 35.29
QSUP(I,J) = QSUP(I,J) * 17.6
QSUP2(I,J) = QSUP2(I,J) * 17.6
WVPEX2(I,J) = WVPEX2(I,J) / 2.2
EXNH3(I,J) = EXNH3(I,J) / 2.2
125 CONTINUE
100 CONTINUE
WRITE(7,400) (J, (I, VRM(I,J), VRH(I,J), VENTNH3(I,J), QSUP(I,J), &I = 1, 49), J = 1, 4)
400 FORMAT (/4('BROOD #', I2, //, 5X, 'DAY', 8X, 'VRM(CMM)', 5X, 'VRH(CMM)', &6X, 'VRN(CMM)', 5X, 'QSUP(W)', //, 49(5X, I2, 5X, F9.1, 5X, F9.1, &5X, F9.1, 5X, F9.1, /) ) )
WRITE(8, 500) (J, (I, VR(I,J), WVPEX2(I,J), QSUP2(I,J), EXNH3(I,J), &I = 1, 49), J = 1, 4)
500 FORMAT (4('BROOD #', I2, //, 5X, 'DAY', 6X, 'VR(CMM)', 3X, &'EXH20(KG/MIN)', 3X, 'QSUP(W)', 6X, 'EXNH3-N(KG/MIN)', &//, 49(5X, I2, 3X, F9.1, 5X, F8.4, 7X, F8.2, 7X, F10.6, / ) ) )
900 STOP
END
*PSYCHROMETRIC EVALUATION PROGRAM
*
* SUBROUTINE PSYCHRO(TDB, RH, H, W, VSA, HFG)
*
TDBAB=TDB+459.69
IF(TDB.GE.32.0)THEN
A=54.6329-12301.688/TDBAB-.56983*ALOG(TDBAB)
PS=EXP(A)
PV=RH*PS
HFG=1075.8965-.56983*(TDBAB-491.69)
ELSE
A=23.3924-11286.6489/TDBAB-.46057*ALOG(TDBAB)
PS=EXP(A)
PV=RH*PS
HFG=1220.844-.05077*(TDBAB-459.69)
ENDIF
W=.6219*PV/(14.696-PV)
VSA=53.35*TDBAB/(144*(14.696-PV))
CALL ITERATE(PV,TDBAB,TDPAB,TWBAB)
TDP=TDPAB-459.69
TWB=TWBAB-459.69
IF(TDP.GE.32.0)THEN
H=.2405*TDB+W*(.448*TDBAB-.01783*TDPAB+864.7168)
ELSE
H=.2405*TDB+W*(.448*TDBAB-.01377*TDPAB+862.3629)
ENDIF
RETURN
END

*ITERATE*

SUBROUTINE ITERATE(P,DB,DP,WB)
G1=200
G2=579.69
I=0
10 I=I+1
IF(I.EQ.1.AND.DB.LT.491.69)THEN
J=1
F1=FN2(P,G1)
F2=FN2(P,G2)
ELSEIF(I.EQ.1.AND.DB.GE.491.69)THEN
J=2
F1=FN1(P,G1)
F2=FN1(P,G2)
ELSEIF(I.EQ.2.AND.DB.GE.491.69)THEN
J=3
F1=FN3(P,G1,DP)
F2=FN3(P,G2,DP)
ELSEIF(I.EQ.2.AND.DB.LT.491.69)THEN
GO TO 40
ENDIF
GO TO 50
*40 J=4
F1=FN4(P,G1,DP)
F2=FN4(P,G2,DP)
*50 F3=G2-((G2-G1)/(F2-F1))*F2
IF(J.EQ.1)F4=FN2(P,F3)
IF(J.EQ.2)F4=FN1(P,F3)
IF(J.EQ.3)F4=FN3(P,F3,DP)
IF(J.EQ.4)F4=FN4(P,F3,DP)
IF(ABS(F4).LE.0.00001)THEN
GO TO 75
ELSEIF(F4*F2.GT.0.0)THEN
F1=F1/2
G2=F3
F2 = F4
GO TO 50
ELSE
  G1 = G2
  G2 = F3
  F1 = F2
  F2 = F4
  GO TO 50
ENDIF

75 IF (J.EQ.1 .OR. J.EQ.2) THEN
  DP = F3
  GO TO 10
ENDIF

IF (J.EQ.3 .AND. F3.LT.489.69) GO TO 40
IF (J.EQ.3 .OR. J.EQ.4) WB = F3
RETURN
END

*FN1
*FUNCTION FN1(P1, T1)
FN1 = 54.6329 - 12301.688 / T1 - 5.16923 * ALOG(T1) - ALOG(P1)
RETURN
END

*FN2
*FUNCTION FN2(P1, T1)
FN2 = 23.3924 - 11286.6489 / T1 - .46057 * ALOG(T1) - ALOG(P1)
RETURN
END

*FN3
*FUNCTION FN3(P1, T1, T2)
FN3 = (.2405 * (EXP(54.6329 - 12301.688 / T1 - 5.16923 * ALOG(T1)) - 14.696) & *(1 + 1.5577 * P1 / 14.696) * (T1 - T2) /.62194 * (1075.8965 - .56983 * & (T1 - 491.69))) + P1 * EXP(54.6329 - 12301.688 / T1 - 5.16923 * ALOG(T1))
RETURN
END

*FN4
*FUNCTION FN4(P1, T1, T2)
FN4 = (.2405 * (EXP(23.3924 - 11286.6489 / T1 - .46057 * ALOG(T1)) - 14.696 & *(1 + 1.5577 * P1 / 14.696) * (T1 - T2) /.62194 * (1220.844 - .05077 * & (T1 - 459.69))) + P1 * EXP(23.3924 - 11286.6489 / T1 - .46057 * ALOG(T1))
RETURN
END
APPENDIX B. Ventilation Requirement Calculation

* * *

PROGRAM FOR CALCULATE THE VENTILATION RATE

PROGRAM VENT(INPUT,OUTPUT,TAPE5,TAPE6)

IMPLICIT REAL(M)

OPEN (UNIT=5,FILE='TEMP',STATUS='OLD')

PRINT *, 'STRUCTURE DIMENSION WIDTH,LENGTH,& WALL HEIGHT(FT)'
READ *, WD, LG, WH
PRINT *, 'INPUT THE PORTION OF THE FOUNDATION ABOVE GROUND(FT)'
READ *, ABOVE
PRINT *, 'ROOF SLOPE(1:3/12 2:4/12 3:5/12)'
READ *, SL
IF(SL.EQ.1) SLOPE=3./12.
IF(SL.EQ.2) SLOPE=4./12.
IF(SL.EQ.3) SLOPE=5./12.
SO=(SLOPE*WD/2.)**2.
RL=SQRT(SO+WD*WD/4.)
PERI=2.*(WD+LG)
AC=WD*LG
AW=2.*(LG+WD)*WH
IF(SL.EQ.1) SLOPE=3./12.
IF(SL.EQ.2) SLOPE=4./12.
IF(SL.EQ.3) SLOPE=5./12.
SO=(SLOPE*WD/2.)**2.
RL=SQRT(SO+WD*WD/4.)
PERI=2.*(WD+LG)
AR=RL*LG*2.
AC=WD*LG
AW=2.*(LG+WD)*WH
AF=PERI*ABOVE
PRINT *, 'DESIGN INSIDE TEMP.& RELATIVE HUMIDITY(DECIMAL)'
READ *, TI, RHI
PRINT *, 'HEAT TRAN.COEF. OF WALL,ROOF,CEILING (BTU/HR-FT"-HR)'
READ *, UW, UR, UC
PRINT *, 'HEAT TRAN.COEF. OF PERIMITER & FOUNDATION (BTU/HR-FT"-HR)'
READ *, UP, UF
PRINT *, 'LIVESTOCK: SENSIBLE & LATENT HEAT PROD. (BTU/HR-HEAD)'
READ *, QS, QL
PRINT *, 'ANY HEAT GAIN FROM LIGHT OR MOTOR ?'
READ *, QM
PRINT *, 'HOW MANY HEAD IN STRUCTURE ?'
READ *, HEAD
WRITE(6,5)
WRITE(6,15) TI, RHI
WRITE(6,25)
COEF=AR/AC
URC=(UR*UC)/(UR+UC/COEF)
QS=QS*HEAD/60.
QL=QL*HEAD/60.
CALL PSYCHRO(TI, RHI, TWB, TDP, HI, WI, VI, HFGI)
DO 100 J=1,11
READ(5,*) RHO, TO
CALL PSYCHRO(TO, RHO, TWB, TDP, HO, VO, HO, VFGO)
DT=TI-TO
A=((14.696*WO)/(.6219+WO))
TIA=TI+459.69
RHP=A/EXP(54.6329-12301.688/TIA-5.16923*ALOG(TIA))
CALL PSYCHRO(TI, RHP, TWB, TDP, HIP, A1, A2, A3)
QC=AC*DT*URC/60.
QW=AW*DT*UW/60.
QP=AF*DT*UF/60.
QB=QC+QP+QP+QW
QM=QM/60.
VAPOR=QL/HFGI
MAH2O = VAPOR / (WI - WO)
QVM = (HIP - HO) * MAH2O
QSUPM = QVM + QB - QS - QM
QVHEAT = QS + QM - QB
MAHEAT = QVHEAT / (HIP - HO)
VRHEAT = MAHEAT * VI
VRMOIT = MAH2O * VI
WRITE (6, 35) TO, VRHEAT, VRMOIT

100 CONTINUE
5 FORMAT (///, 1X, 'DESIGN CONDITION AS FOLLOW:', //)
15 FORMAT (1X, 'INSIDE TEMP. = ', F4.0, ', RH = ', F4.2, //)
25 FORMAT (1X, 'TEMP(OUT)', 4X, 'TEMP(CFM)', 6X, 'MOISTURE(CFM)', //)
35 FORMAT (4X, F4.0, 8X, F5.0, 10X, F5.0)
STOP
END
APPENDIX C. Ventilation Requirement Calculation

* *
HEATING LOAD CALCULATION PROGRAM
* *
PROGRAM HEATMDM(INPUT,OUTPUT,TAP5,TAP6,TAP9)
* *
COMMON TDATA(5,49,24)
DIMENSION CLTDW(4,24),CLTDR(24),CLF(24),AREA(6),U(5)
DIMENSION CLTDWC(5,12,24),QSOLAR(12,5),QSENS(50),Q Equip(24)
DIMENSION CLTDRC(5,12,24),QSUP(49,24),VRH(49,24),VRM(49,24)
DIMENSION QHEAT(49,24),WVPD(49),CLFCUR(5,24),SHGF(12,5)
DIMENSION TOUTM(5,49),RH(49)
INTEGER D,H,ORT,HR,DAY
REAL K,LAMP,LM(12,5)
CHARACTER*10 DATE,TIME
WRITE(6,1) DATE( ),TIME( )
OPEN (UNIT=5,FILE='CLTD',STATUS='OLD')
READ(5,*) ((CLTDW(I,J),J=1,24),I=1,4)
READ(5,*) ((LM(I,J),J=1,5),I=1,12)
READ(5,*) (CLTDR(I),I=1,24)
READ(5,*) ((CLFCUR(I,J),J=1,24),I=1,4)
READ(5,*) ((SHGF(I,J),J=1,4),I=1,12)
READ(5,*) (CLF(I),I=1,24)
READ(5,*) ((TDATA(NB,DAY,HR),HR=1,24),DAY=1,49),NB=3,5)
READ(5,*) ((TOUTM(NB,DAY),NB=3,5),DAY=1,49)
READ(5,*) (RH(I),I=1,49)
REWIND (UNIT=5)
READ *, (AREA(I),I=1,6)
PRINT *, 'OVERALL HEAT TRANSFER COEFFICIENT 1-4(WALL) 5:ROOF'
READ *, (U(I),I=1,5)
PRINT *, 'NO. OF LAMP WATTS & DURATION FACTOR'
READ *, LAMP,W,FS
PRINT *, 'WHICH BROODING,'
READ *, NB
PRINT *, 'CALCULATION PERIOD FROM: TO ?'
READ *, LPLB,LPUB
PRINT *, 'BUILDING COLOR FACTOR'
READ *, CK
PRINT *, 'ROOF & CEILING CORRECTION FACTOR'
READ *, F
PRINT *, 'ROLLER CURTAIN SHADING COEFFICIENT'
READ *, SC
PRINT *, 'BUILDING ORIENTATION 1: NORTH-SOUTH, 2: EAST-WEST'
READ *, ORT
PRINT *, 'DO YOU NEED DETAILS OF AIR?(YES=1 NO=0)'
READ *, ITER
PRINT *, 'DO YOU NEED PLOTTING OUTPUT?(YES=1 NO=0)'
READ *, ISCALE
PRINT *, 'HOW MANY BIRDS IN THE BUILDING?'
READ *, BIRD
M=(NB*2)-1
WRITE(6,5)
IF(ORT.EQ.1) THEN
WRITE(6,15)
ELSE
WRITE(6,25)
ENDIF
WRITE(6,35) AREA(5)
WRITE(6,45) LAMP,W
WRITE(6,55) NB
DO 400 N=1,49
IF(ITER.EQ.0) WRITE(6,65) N
IF(ITER.EQ.0) WRITE(6,75)
IF(NB.EQ.1.AND.N.EQ.30) M=M+1
IF(NB.EQ.2.AND.N.EQ.26) M=M+1
IF(NB.EQ.3.AND.N.EQ.24) M=M+1
IF(NB.EQ.4.AND.N.EQ.28) M=M+1
IF(NB.EQ.5.AND.N.EQ.26) M=M+1
QBUILD=0.
RHIN=0.6
IF(N.LT.7) TI=82
IF(N.GT.7.AND.N.LE.14) TI=79
IF(N.GT.14.AND.N.LE.21) TI=76
RHIN=0.7
TI=73.
ENDIF
IF(N.GT.28) THEN
TI=70.
RHIN=0.7
ENDIF
RHOUT=RH(N)
TINSD=TI
DO 300 H=LPLB,LPUB
TOUT=TDATA(NB,N,H)*9./5.+32.
TM=TOUTM(NB,N)*9./5.+32.
QBUILD=0.
DO 200 D=1,5
IF(D.EQ.5) THEN
CLTDRC(D,M,H)=(CLTDR(H)+LM(M,D))*CK+78.-TINSD+(TM-85)
QSOlar(M,D)=U(D)*AREA(D)*CLTDRC(D,M,H)*F
ELSE
CLTDWC(D,M,H)=(CLTDW(D,H)+LM(M,D))*CK+78.-TINSD+(TM-85)
QSOlar(M,D)=U(D)*AREA(D)*CLTDWC(D,M,H)
ENDIF
QBUILD=QBUILD+QSOlar(M,D)
300 CONTINUE
DO 100 J=ORT,ORT+2
QRADIA=SC*SHGF(M,J)*AREA(6)*CLFCUR(ORT,H)
100 QBUILD=QBUILD+QRADIA
IF(N.LE.22) THEN
QSENSI(N)=(-.76*(N**3.)-10.*N*N+347.*N-511.)*BIRD/1000
ELSE
A=-0.00000558659*(N**3.)-0.0000000696725*(N**4.)
WT=0.138492+0.0103605*N+0.00152572*N*N+A
B=1.23608*WT*WT-0.228763*WT*WT*WT+0.0231126*(WT**4.)
QSENSI(N)=(14.2393-4.94177*WT+B)*BIRD
ENDIF
NH=H-7
QEQUIP(H)=3.412*LAMP*W*FS*CLF(NH)
IF(ITER.EQ.0) THEN
COEF=-0.00193599*N*N*N+0.0000195998*N*N*N*N
WVP=(3.05428-0.196998*N+0.0598057*N*N+COEF)*BIRD/1000
QVENTH=QSENSI(N)+QEQUIP(H)+QBUILD
WRITE(6,85) H,TOUT,WVP,QSENSI(N),QEQUIP(H),QBUILD,QVENTH
ELSE
CALL PSYCHRO(TOUT,RHOUT,HO,WO,VO,HFGO)
TIA=TI+59.69
CALL PSYCHRO(TI,RHIN,HI,HI,HI)
OR=EXP(54.6329-12301.688/TIA-5.16923*ALOG(TIA))
RHPRIME=((14.696*WO)/(0.6219+WO))/OR
CALL PSYCHRO(TI,RHPRIME,HIP,DM,DO)
COEF=-0.00193599*N*N*N+0.0000195998*N*N*N*N
WVP=(3.05428-0.196998*N+0.0598057*N*N+COEF)*BIRD/1000
H2OMA=WVP/(WI-WO)
QVENT = (HIP-HO)*H2OMA
QVENTH = QSENSI(N) + QEQUIP(H) + QBUILD
HEATMA = QVENTH/(HIP-HO)
QHEAT(N, H) = QVENTH
IF(ISCALE.EQ.0) THEN
WRITE(6, 85) H, TOUT, WVP, QSENSI(N), QEQUIP(H), QBUILD, QVENTH
ENDIF
* PRINT *, 'DAYS: ', N, ' HOUR: ', H
* PRINT *, 'WVP= ', WVP
* PRINT *, 'H2OMA= ', H2OMA
* PRINT *, 'QVENT= ', QVENT
* PRINT *, 'QVENTH= ', QVENTH
* PRINT *, 'HEATMA= ', HEATMA
* PRINT *, 'QBUILDING = ', QBUILD
* PRINT *, 'QSENSIBLE = ', QSENSI(N)
* PRINT *, 'QEQUIPMENT = ', QEQUIP(H)
VRH(N, H) = HEATMA*VI/60./25.
VRM(N, H) = H2OMA*VI/60./25.
* PRINT *, 'VRH= ', VRH(N, H), ' VRM= ', VRM(N, H)
* PRINT *, 'HIP= ', HIP, ' HO= ', HO, ' HI= ', HI
ENDIF
WRITE(9, 85) QVENTH
300 CONTINUE
WVPD(N) = WVP
400 CONTINUE
IF(ISCALE.EQ.1) CALL SCALE(NB, WVPD, QHEAT, VRH, VRM, LPLB, LPUB)
1 FORMAT(//,1X,'PROGRAM EXECUTING AT : ',2(1X,A10),/)
5 FORMAT(1X,'SYSTEM MANAGEMENT SUMMARY :',/)
15 FORMAT(1X,'ORIENTATION : FACING SOUTH')
25 FORMAT(1X,'ORIENTATION : FACING EAST')
35 FORMAT(1X,'FLOOR AREA : ',F8.1, ' FT*FT')
45 FORMAT(1X,'LAMP: NO.',F5.0,'*',F4.0,' WATTS',/)
55 FORMAT(1X,'NO. ',12,' BROOD ')
65 FORMAT(//,1X,'AGE (DAYS): ',13,/) 75
FORMAT(3X,'TIME',2X,'TEMP(OT)',4X,'MOISTURE',2X,'SENSHEAT',2X,'EQUI
LIPHEAT',2X,'BUILDHEAT',2X,'TOTAL (BTU/HR)',/) 85 FORMAT(4X,I2,4X,F6.2,5X,F6.2,3X,F8.0,3X,F8.0,2X,F9.0,4X,F10.0)
STOP
END
* SCALING FUNCTION SUBROUTINE
* SUBROUTINE SCALE(NB, WVPD, QHEAT, VRH, VRM, LPLB, LPUB)
* COMMON TDATA(5,49,24)
DIMENSION WVPD(49), QHEAT(49, 24), VRH(49, 24), VRM(49, 24)
CHARACTER*1 BLANK, DOT, STAR, PLUS, MINUS
CHARACTER*81 LINE(0:80)
INTEGER H
BLANK=''
DOT='.'
STAR='*'  
PLUS='+'
MINUS='-'+
DO 1500 N=1,49
WRITE(6,1025) N
WRITE(6,1005) N
DO 1400 H=LPLB,LPUB
TOUT=TDATA(NB,N,H)*9./5.+32.
DO 1000 I=0,80
1000 LINE(I)=BLANK
LINE(0)=LINE(80)=DOT
IF(H.EQ.LPLB.OR.H.EQ.LPUB) THEN
DO 1100 I=0,80
1100 LINE(I)=DOT
   ENDIF
   MMOIS=VRM(N,H)/100.+.5
   MHEAT=VRH(N,H)/100.+.5
   IF(MMOIS.LT.0) MMOIS=0
   IF(MHEAT.LT.0) MHEAT=0
   IF(MMOIS.GT.80) MMOIS=80
   IF(MHEAT.GT.80) MHEAT=80
   LINE(MMOIS)=STAR
   LINE(MHEAT)=DOT
   WRITE(6,1015) H,TOUT,VRM(N,H),VRH(N,H),LINE
1400 CONTINUE
1500 CONTINUE
1005 FORMAT(3X,'TIME',2X,'TEMP(OT)',2X,'MRATE(CFM)',2X,'HRATE(CFM)')
1015 FORMAT(4X,I2,4X,F6.2,3X,F9.0,3X,F9.0,2X,81A1)
1025 FORMAT(//,1X,'AGE (DAYS): ',13,/)RETURN
END

*PSYCHROMETRIC EVALUATION PROGRAM*

SUBROUTINE PSYCHRO(TDB,RH,H,W,VSA,HFG)
   TDBAB=TDB+459.69
   IF(TDB,GE.32.0)THEN
      A=54.6329-12301.688/TDBAB-5.16923*ALOG(TDBAB)
      PS=EXP(A)
      PV=RH*PS
      HFG=1075.8965-.56983*(TDBAB-491.69)
   ELSE
      A=23.3924-11286.6489/TDBAB-.46057*ALOG(TDBAB)
      PS=EXP(A)
      PV=RH*PS
      HFG=1220.844-.05077*(TDBAB-459.69)
   ENDIF
   W=.6219*PV/(14.696-PV)
   VSA=53.35*TDBAB/(144*(14.696-PV))
   CALL ITERATE(PV,TDBAB,TDPAB,TWBAB)
   TDP=TDPAB-459.69
   TWB=TWBAB-459.69
   IF(TDP,GE.32.0)THEN
      H=.2405*TDB+W*(.448*TDBAB-.01783*TDPAB+864.7168)
   ELSE
      H=.2405*TDB+W*(.448*TDBAB-.01377*TDPAB+862.3629)
   ENDIF
   PRINT*, 'TDB', 'RH', 'TWB', 'TPD', 'H', 'W', 'VSA', 'HFG'
   PRINT*,TDB,RH,TWB,TDP,H,W,VSA,HFGRETURN
END

*ITERATE*

SUBROUTINE ITERATE(P,DB,DP,WB)
   G1=200
   G2=579.69
   I=0
10   I=I+1
   IF(I.EQ.1.AND.DB.LT.491.69)THEN
      J=1
      F1=FN2(P,G1)
      F2=FN2(P,G2)
   ELSEIF(I.EQ.1.AND.DB.GE.491.69)THEN
      J=2
   ENDIF
F1=FN1(P,G1)  
F2=FN1(P,G2)  
ELSEIF(I.EQ.2.AND.DB.GE.491.69)THEN  
  J=3  
  F1=FN3(P,G1,DB)  
  F2=FN3(P,G2,DB)  
ELSEIF(I.EQ.2.AND.DB.LT.491.69)THEN  
  GO TO 40  
  ENDIF  
  GO TO 50  
  *  
  40  
  J=4  
  F1=FN4(P,G1,DB)  
  F2=FN4(P,G2,DB)  
  *  
  50  
  F3=G2-((G2-G1)/(F2-F1))*F2  
  IF(J.EQ.1)F4=FN2(P,F3)  
  IF(J.EQ.2)F4=FN1(P,F3)  
  IF(J.EQ.3)F4=FN3(P,F3,DB)  
  IF(J.EQ.4)F4=FN4(P,F3,DB)  
  IF(ABS(F4).LE.0.00001)THEN  
    GO TO 75  
  ELSEIF(F4*F2.GT.0.0)THEN  
    F1=F1/2  
    G2=F3  
    F2=F4  
    GO TO 50  
  ELSE  
    G1=G2  
    G2=F3  
    F1=F2  
    F2=F4  
    GO TO 50  
  ENDIF  
  75  
  IF(J.EQ.1.OR.J.EQ.2)THEN  
    DP=F3  
    GO TO 10  
  ENDIF  
  IF(J.EQ.3.AND.F3.LT.489.69)GO TO 40  
  IF(J.EQ.3.OR.J.EQ.4)WB=F3  
  RETURN  
END  
*  
*FN1  
*  
FUNCTION FN1(P1,T1)  
FN1=54.6329-12301.688/T1-5.16923*ALOG(T1)-ALOG(P1)  
RETURN  
END  
*  
*FN2  
*  
FUNCTION FN2(P1,T1)  
FN2=23.3924-11286.6489/T1-.46057*ALOG(T1)-ALOG(P1)  
RETURN  
END  
*  
*FN3  
*  
FUNCTION FN3(P1,T1,T2)  
FN3=.2405*(EXP(54.6329-12301.688/T1-5.16923*ALOG(T1))-14.696)  
&*(1+.15577*P1/14.696)*(T1-T2)/(.62194*(1075.8965-.56983*  
&(T1-491.69)))+P1-EXP(54.6329-12301.688/T1-5.16923*ALOG(T1))
FUNCTION FN4(P1,T1,T2)
FN4=(.2405*(EXP(23.3924-11286.6489/T1-.46057*ALOG(T1))-14.696
&)*(1+.15577*P1/14.696)*(T1-T2))/(.62194*(1220.844-.05077*
&(T1-459.69)))+P1-EXP(23.3924-11286.6489/T1-.46057*ALOG(T1))
RETURN
END
APPENDIX D. Rockbed Performance Simulation Program

* ROCKBED TEMPERATURE PROFILE PREDICT PROGRAM
* PROGRAM ROCKMOD(INPUT,OUTPUT,TAPE6,TAPE7,TAPE8)
* DIMENSION
A(150),B(150),C(150),F(150),AR(150),R(150),OT(50),Z(150)
DIMENSION TROLD(150),TIME(0:2000),TINL(2000),T(150),TROCK(150)
DIMENSION ORIGT(50)
CHARACTER*10 DATE
REAL K1,K2
DATA PI/3.14159264/
PRINT *,'NODES,TIME INCREM,AMBTEMP,COLLECTIME=?'
READ *, N,DT,TAMB,TIMEC
PRINT *,'AIR(VEL,DENSI,SPEC.HEAT)ROCK(DIA,SPEC.HEAT,DENSITY=?'
READ *, U,RHOA,CAIR,DROCK,CROCK,RHOR
PRINT *,'VOID FRACTION. PRINTOUT INTERVAL=?'
READ *, VF,IP
PRINT *, 'BED LENGTH (METER)=?' 
READ *, BEDLG
PRINT *, 'INLET TEMP. USING COLLECTOR OR DATA?(COLLECT=1,DATA=2)'
READ *, INPUT
IF(INPUT.EQ.2) THEN
ITIMEC=1+TIMEC
PRINT *,'STARTING FROM WHEN?'
READ *, ISTART
PRINT *, 'INPUT THE TEMPERATURE FOR EACH HOUR'
READ *, (OT(IFLAG),IFLAG=ISTART,ISTART+ITIMEC)
ENDIF
PRINT *, 'IS THIS A DISCHARGING PROCESS?(YES=1 NO=0)'
READ *, IDCHARG
IF(IDCHARG.EQ.1) THEN
PRINT *, 'TAPE8(OUTLET) AVAILABLE?(YES=1,NO=0)' 
READ *, ATAPE
IF(ATAPE.EQ.1) THEN
OPEN(UNIT=8,FILE='OUTLET')
READ(8,*) (ORIGT(I),I=1,N,-1)
REWIND (UNIT=8)
ELSE
PRINT *, 'THE INITIAL CONDITION FOR EACH NODE=?'
READ *, (ORIGT(I),I=1,N)
ENDIF
ENDIF
GV=U*RHOA*VF
PRINT *, 'DO YOU NEED 3-DIMENSION PLOTTING?(YES=1,NO=0)'
READ *, I3D
IF(I3D.EQ.1) OPEN(UNIT=7,FILE='D3PLOT')
CPD=BEDLG*GV*GV/(RHOA*DROCK)
PD=CPD*[(21.+1750*1.81*0.00001/(GV*DROCK))]
TBEDLG=RHOA*CAIR*VF*U*TIMEC*3600./(RHOR*CROCK*(1.-VF))
M=3600.*TIMEC/DT
DX=BEDLG/FLOAT(N)
HV=700.*((GV/DROCK)**0.76)
WRITE(6,1) DATE( )
WRITE(6,5)
WRITE(6,15) TIMEC
WRITE(6,25) BEDLG
WRITE(6,35) TAMG
WRITE(6,45) GV
WRITE(6,55) U
WRITE(6,65) HV
WRITE(6,75)  CROCK
WRITE(6,85)  CAIR
WRITE(6,105) RHR
WRITE(6,115) RHOA
WRITE(6,125) DROCK
WRITE(6,135) VF
WRITE(6,145) N
WRITE(6,155) DX
WRITE(6,165) DT
WRITE(6,185) TBEDLG
WRITE(6,195) ISTART
IF(IDCHARG.EQ.1) WRITE(6,205)
IF(IDCHARG.EQ.0) WRITE(6,215)
WRITE(6,225) PD
K1=HV/(RHOA*CAIR*VF)
K2=HV/(RHR*CROCK*(1.-VF))
ACOE=(U*DT)/(2.*DX)
Bcoe=K1*DT
CCOE=K1*DT/(1.+K2*DT)
D=(K1*K2*DT*DT)/(1.+K2*DT)
E=1.+Bcoe-D
Fcoe=1.+Bcoe-D+2.*ACOE
G=2.*ACOE
T(time(0)=0.
IF(INPUT.EQ.2) THEN
IFLAG=1
IT=ISTART
DO 10 I=1,M+1
TIME(I)=TIME(I-1)+DT
IF((DT*IT).LT.(3600*IFLAG)) THEN
TINL(I)=OT(IT)
ELSE
IFLAG=IFLAG+1
IT=IT+1
TINL(I)=OT(IT)
ENDIF
10 CONTINUE
ELSE
DO 50 I=1,M+1
TIME(I)=TIME(I-1)+DT
50 TINL(I)=-15.3*COS(PI*TIME(I)/(3600*4.))+36.4
ENDIF
DO 100 I=1,N
A(I)=E
C(I)=ACOE
IF(I.EQ.N) THEN
B(I)=-G
A(I)=Fcoe
ELSE
ENDIF
IF(I.GT.1.AND.I.LT.N) B(I)=-ACOE
100 CONTINUE
IF(IDCHARG.EQ.1) THEN
F(I)=ACOE*TINL(I)+ORIGT(I)*(1.+CCOE)
DO 110 I=2,N
110 F(I)=ORIGT(I)*(1.+CCOE)
ELSE
CAL=TAMB*(1.+CCOE)
DO 150 I=1,N
IF(I.EQ.1) THEN
F(I)=ACOE*TINL(I)+CAL
ELSE
ENDIF
150 CONTINUE
ELSE
F(I)=CAL
ENDIF

CONTINUE

ENDIF
AR(I)=A(I)
R(I)=C(I)/AR(I)
DO 200 I=2,N-1
AR(I)=A(I)-B(I)*R(I-1)
200 R(I)=C(I)/AR(I)
AR(N)=A(N)-B(N)*R(N-1)
IF(IDCHARG.EQ.1) THEN
DO 210 I=1,N
210 TROLD(I)=ORIGT(I)
ELSE
DO 250 I=1,N
250 TROLD(I)=TAMB
ENDIF
DO 700 D=1,M
X=0.
Z(1)=F(1)/AR(1)
DO 300 I=2,N
300 Z(I)=(F(I)-B(I)*Z(I-1))/AR(I)
T(N)=Z(N)
DO 400 I=N-1,1,-1
400 T(I)=Z(I)-R(I)*T(I+1)
DO 500 I=1,N
500 TROLD(I)=TROCK(I)
DO 600 I=1,N
IF(I.EQ.1) THEN
F(I)=ACOE*TINL(L+1)+CCOE*TROCK(I)+T(I)
ELSE
F(I)=CCOE*TROCK(I)+T(I)
ENDIF
600 CONTINUE
IF((L/IP*IP).EQ.L) CALL
SCALE(T,TROCK,N,L,TIME,TINL,DX,M,IP,I3D)
700 CONTINUE
IF(I3D.EQ.1) THEN
ENDFILE (UNIT=7)
CLOSE (UNIT=7)
REWIND (UNIT=7)
ENDIF

FORMAT(///,1X,'UPDATE VERSION OF ROCKMOD AT ',(1X,A10),/)
FORMAT(///,1X,'SYSTEM & OPERATION SCHEDULE:',/)
15 FORMAT(1X,'SOLAR COLLECTOR COLLECT TIME--',F7.3,' HOURS')
25 FORMAT(1X,'BEDLG:STORAGE ROCKBED LENGTH--',F7.4,' METER')
35 FORMAT(1X,'TAMB:STARTING AMBIENT TEMP--',F7.3,' DEG C')
45 FORMAT(1X,'GV: MASS FLOW RATE----------------',F7.5,' KG/S-SQM')
55 FORMAT(1X,'U: AIR FLOW SPEED-----------------',F7.5,' M/SEC')
65 FORMAT(1X,'HV: HEAT TRAN. COEF.------------',F7.2,' W/CUM -C')
75 FORMAT(///,1X,'MATERIALS SUMMARY:',/)
85 FORMAT(1X,'CROCK: SPECIFIC HEAT OF ROCK--',F7.1,' JOULE/KG-C')
95 FORMAT(1X,'CAIR: SPECIFIC HEAT OF AIR--',F7.1,' JOULE/KG-C')
105 FORMAT(1X,'RHOR: DENSITY OF ROCK---------',F7.1,' KG/CUB M')
115 FORMAT(1X,'RHOA: DENSITY OF AIR--------',F7.4,' KG/CUB M')
125 FORMAT(1X,'DROCK: ROCK DIAMETER-----------',F7.5,' METER')
135 FORMAT(1X,'VF: VOID FRACTION------------',F7.4)
145 FORMAT(///,1X,'CALCULATION PARAMETERS:',/)
155 FORMAT(1X,'N: NUMBER OF NODES USED------',I7)
165 FORMAT(1X,'DX: SPACE INCREMENT----------',F7.3,' METER')
175 FORMAT(1X,'DT: TIME INCREMENT----------',F7.1,' SEC')
SUBROUTINE SCALE(T,TROCK,N,L,TIME,TINL,DX,M,IP,I3D)
DIMENSION T(150),TROCK(150),TIME(0:2000),TINL(2000)
CHARACTER*53 LINE(0:52)
CHARACTER*1 BLANK,DOT,STAR,PLUS,MINUS
IF((M-L).LT.IP.AND.IDCHARG.EQ.0) THEN
OPEN (UNIT=8,FILE='OUTLET')
ENDIF
BLANK=' '
DOT='.'
STAR='*'
PLUS='=' + '
MINUS=' - '
WRITE(6,305) TIME(L),TINL(L)
WRITE(6,315)
WRITE(6,325)
X=0.
DO 1200 J=1,N
IF(I3D.EQ.1) THEN
WRITE(7,*) TROCK(J)
ENDIF
X=X+DX
DO 1000 I=0,52
1000 LINE(I)=BLANK
LINE(0)=LINE(52)=DOT
IF(J.EQ.1.OR.J.EQ.N) THEN
DO 1100 I=0,52
1100 LINE(I)=DOT
LINE(5)=LINE(10)=LINE(15)=LINE(20)=LINE(25)=PLUS
LINE(30)=LINE(35)=LINE(40)=LINE(45)=LINE(50)=PLUS
ENDIF
MA=T(J)+.5
MR=TROCK(J)+.5
LINE(MR)=STAR
LINE(MA)=DOT
IF((J/5*5).EQ.J.AND.J.NE.N) LINE(0)=MINUS
WRITE(6,335) J,X,T(J),TROCK(J),LINE
IF((M-L).LT.IP) THEN
WRITE(8,*) TROCK(J)
IF(J.EQ.N) THEN
ENDFILE(UNIT=8)
ENDIF
CLOSE (UNIT=8)
REWIND (UNIT=8)
ENDIF
ENDIF
CONTINUE
1200 CONTINUE
STOP
END

* SCALING FUNCTION PROGRAM *

*