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A Rockbed Regenerator for a
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A ROCKBED REGENERATOR FOR A FARROWING ROOM

By

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ABSTRACT

A ROCKBED REGENERATOR FOR A FARROWING ROOM

By

Yi Chen

Two rockbeds of 0.64 m^2 were constructed for a farrowing room to recover heat from exhaust air and preheat incoming air during winter days. Rockbed depth, cycle time and air flow rates were tested to evaluate their effects on performance of the regenerative ventilation system. An orthogonal design was used to form a half-fraction factorial experiment.

On an average a heat effectiveness of 0.60 and a moisture removal factor of 0.43 were achieved. Cycle time affected the heat recovery effectiveness and the moisture removal factor significantly. Rockbed depth was of little importance. Interaction between two air flow rates was strong. Essentially equal mass flow rates of the room exhaust and make-up air, 0.4 m rockbed depth and a heat recovery effectiveness about 0.5 were recommended.

Balance equations of air mass, moisture and energy in animal housing and design criteria for regenerative ventilation systems were proposed.

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1 INTRODUCTION

1.1 Background

A controlled environment for confined animal housing is necessary not only to protect livestock or poultry from cold and hot weather but also to provide them with nearly optimum conditions for high production (Esmay, 1978). Temperature, humidity and fresh air in confined animal structures are the most important environmental factors. Continuous mechanical ventilation is a common, effective means to remove water vapor, carbon dioxide, ammonia, odors, dust and air-borne disease organisms, as well as sensible heat, and to supply fresh air for confined animal houses. However, under cold weather conditions, supplemental heat may be required to maintain an appropriate indoor air temperature. This is especially important for baby animals and young stock, because they can not produce sufficient heat to regulate their body temperatures in cold buildings. The heat required depends on climate degree days, density and size of animals housed, building insulation, and the ventilation rate used.

Various types of heating devices, such as gas, oil or water heaters, electric heaters, brooder lamps, electric pads and so on, are used to provide supplemental heat in animal houses through winter months. Most of them consume fossil fuel (gas, oil, or coal) directly or indirectly. Since both the uncertainty of supply and the increasing cost of conventional fossil fuel have risen in recent years, there has been an increasing interest in searching for alternative heat energy sources.

The use of heat exchangers to recover a portion of the heat from the exhaust ventilation air of livestock or poultry buildings to preheat the cold incoming air is an energy saving concept. It can reduce or, even, eliminate the need for supplemental heat. Early research work on shell-and-multitube exchangers of parallel-flow and counterflow was reported by Giese and Downing (1950), and Giese and Ibrahim (1950). Giese and Bond (1952), and Ogilvie (1966) reported their counterflow plate-type exchangers. Turnbull (1965) tested a perpendicular flow plate-type exchanger. Larkin et al. (1975 and 1977) operated thermosiphon heat exchangers. All those above have the common problem of complexity, corrosion, cost and maintenance.

In agriculture, rockbed regenerators have been studied since 1970's. Witz et al. (1974 and 1979) reported their work on a rock sink system with rotation reversible fans. Bon et al. (1981) developed a computer program to simulate the performance of the rock sink system above. Lampman (1978) constructed a rockbed system with multiduct-dampers. Moysey et al. (1980) examined an experimental rockbed unit in laboratory conditions. Parker et al. (1978 and 1981) studied pressure drop and heat transfer in crushed limestone and developed a finite difference algorithm to predict the temperature distribution in the rockbed. Chandra et al. (1977) applied dimensional analyses in their study of pressure drop in an experimental rock chamber. Due to the advantages of simple design and construction, high heat capacity, low cost and long life, the rockbed heat exchanger has interested many agricultural engineers, researchers and farmers.

Freezing and dust clogging are common problems dealing with rockbed regenerators. Due to lake influence East Lansing seldom experiences prolonged periods of extreme cold during the winter. From 1940 through 1969 there were an average of only 6 days, on which the daily minimum temperature was -17.8 °C or below. The recorded lowest temperature was -27.2 °C in February of 1959 (NOAA, 1973). Therefore, ice clogging might not be a serious problem when a rockbed regenerator is operated in this area.

Since the existing design procedure associated with balance equations of moisture and sensible heat for conventional ventilation systems is not applicable for regenerative ventilation systems, new design criteria based on new balance equations of mass, moisture and energy are needed.

1.2 Objectives

The objectives of this thesis study were to:

1. Investigate extensively the effects of rockbed design parameters on heat recovery and moisture removal under conditions in a swine farrowing house,

2. Investigate air temperature profiles at various locations within the rockbed and in the ventilation system,

3. Formulate balance equations of mass, moisture and energy and propose criteria for design and evaluation of ventilation systems with heat exchangers.

2 LITERATURE REVIEW AND ANALYTICAL APPROACHES

2.1 Previous Studies in Fields and Laboratories

Rockbeds are alternate-type heat exchangers, through which two working gases do not pass simultaneously. During a charge period while warm gas is flowing through a rockbed, some heat is transferred from the gas to the rocks. During a consequent discharge period when cold gas passes through the precharged rockbed, some heat stored in the rocks is transferred to the gas. Phase changes and mass transfer may also occur associated with the heat transfer.

A rockbed regenerator may consist of two identical rockbeds installed vertically into a ventilation system. One rockbed is charged by downward warm air while the other is discharged by upward cold air. After a time interval the air flow directions are reversed so that the precharged rockbed is discharged whereas the predischarged one is charged. A cycle time includes two half cycles, charge and discharge for each rockbed. There are many ways to reverse the flow directions, such as reversing two fan rotation directions, switching damper systems to change air flow paths and so on.

Witz et al. (1974 and 1979) developed a double rock sink system for a beef barn since 1971. Two insulated rock sinks were located separately at different sides out of the building to prevent short circuiting between the exhaust and incoming air. The ceiling and attic space of the barn was divided into two sections. Each air path made up of one sink, one fan and one section of the ceiling and attic space. The ceiling was perforated. The rotation directions of two fans were reversed at 1 to 15 minute intervals. Reversing the fan rotation caused extra dynamic stress on the motors and fans, and required special blade design to get similar air flow rates in both directions. The area of each sink was approximately 1.04 m^2. The depth of the sinks increased from 223 mm up, as the rock size increased from 19 to 107 mm in diameter. An efficiency of 33 % was calculated on the basis of temperature differences. The room temperature was maintained at about 4.4°C without supplemental heat when the outside temperature was -28.8°C. The use of salt controlled the frost problem when the outside temperature was down to -25°C for 4 to 5 days. For longer periods of one week or more with the outside temperature below -26°C, operating the fans in the exhaust direction for about three hours removed the ice accumulation from the rockbeds. They suggested using small rock size and increasing the cross section area to improve rockbed performance and reduce freezing problems, associated with the use of salt. A cycle time of 10 to 15 min was suggested. They also recommended locating both the inlet and outlet vents on the same side of the building. This was to prevent cold air flowing across the building due to wind when both fans were off.

Lampman (1978) studied a rockbed system in a swine barn. Two rockbeds of 1.22 m square were filled to 0.305 m deep with rocks of 50 to 100 mm of size. The rockbeds were located separately on the same side out of the barn. Both the exhaust fan and the make-up fan were connected to each rockbed through two-way ducts. A single damper motor, controlled by a timer, drove four dampers. Each damper controlled one branch duct, respectively. The shutter system was switched every 10 minutes to exchange the air flow directions through the rockbeds. The room temperature could be maintained above 10°C when holding 300 pigs in the coldest weather of -40°C. To minimize freezing, the exhaust flow rate was 0.85 to 1.4 m^3/s whereas the make-up flow rate was 0.47 to 0.59 m^3/s. Skirts around the bottom of the rockbeds were thought to reduce the effect of windchill. Dust fouling became a severe problem when there was no water condensation on the rocks during periods of low dew point temperatures and warm weather, since condensed water flushed out accumulated dust. Metal mesh filters were used. However, daily maintenance was required. The freezing problem was overcome by combined efforts of adding salt and applying an unidirectional warm air flow for over 1 hour. A heat effectiveness of 28 to 34 % was obtained based on mass flow rates and enthalpy.

Moysey et al. (1980) tested an experimental rockbed regenerator of 0.3 m square in an environmentally controlled room consisting of two compartments. They stated that increasing the superficial flow velocity from 0.31 to 0.58 m/s or the bed thickness from 0.2 to 0.4 m, or reducing the cycle time from 12 to 8 min had significant effect on increasing the system efficiency. The effects of relative humidity in the range of 33 to 78 % and rock size from 35 to 50 mm were of less importance. They mentioned that reducing both the air flow rate and the cycle time resulted in increasing the efficiency. This implied some interaction between these two factors. They computed the heat exchanger efficiency as 21.9 to 50.9 % based on the enthalpy differences or 27.3 to 61.2 % based on the sensible heat differences. Dust clogging and ice accumulation were also observed. They recommended the cycle time of less than 10 min and the superficial airflow velocity of 0.42 m/s or less. They pointed out that using small rocks and deep beds could achieve better efficiencies. However, selecting rock size and bed depth should deal with reducing problems of high pressure drop and dust accumulation.

2.2 Analytical Approaches

The theoretical analysis for rockbed heat exchangers lags behind the accumulation of experimental results. Some mathematical correlations are empirical and semiempirical.

2.2.1 Fundamental Assumptions

Parker et al. (1978) summarized the following basic assumptions in studying the characteristics of both heat transfer and pressure drop in rockbeds:

1. Air temperature, rock temperature and static pressure are uniform at any specific cross section of the rockbed.

2. Thermal and fluid-mechanical properties are constant for both air and the stone.

3. Rocks are uniform in size and shape, and are uniformly but randomly packed in the rockbeds.

4. There is no temperature or pressure gradient perpendicular to the air flow direction, i.e. a "plug flow".

5. There is no edge effect on either the flow or the temperature.

6. There is no temperature gradient within individual stones.

7. There is no heat conduction parallel to the flow direction in either the air or the stone.

8. There is no heat or mass transfer to the surrounding environment.

Carefully screening stones could reduce the error due to non-uniform size, shape and properties. Turbulent flow provides quite uniform air conditions. Good insulation and water proof around the rockbed reduce the losses of heat and water.

Parker et al. (1978) stated that for a larger rockbed randomly packed with uniform rocks, a reasonable approximation of plug flow could be achieved on a statistical basis. Therefore, no heat or mass flow normal to the air flow direction was considered.

Rose (1949) reported that the wall effect might be neglected if the ratios of the diameter and the depth of the rockbed to the particle diameter were at least 50:1 and 20:1, respectively. Close (1965) concluded that the rock pile conductivity in the flow direction was negligible during charging and discharging.

2.2.2 Biot Number

With high rock conductivity, low heat transfer coefficient between air and rock surface, and small rock size, the real temperature gradient within rocks may be ignored. Whether the lumped system approach is reasonable for a given rock size could be estimated by calculating the Biot number (Moyers, 1970):

$$Bi = 0.5H * d / k \qquad (2.1)$$

where H - area heat transfer coefficient of the flow to the rock, $W/m^2 \cdot K$,

k - rock conductivity, $W/m \cdot K$,

d - equivalent spherical diameter of the rock, m.

Moyers (1971) showed that an error of less than 5 % resulted from the lumped system assumption if Bi was less than 0.1. Parker et al. (1980) estimated the Bi value for crushed stone of 25.4 mm in diameter would be at most 0.02

under the condition of normal rockbed flow rates.

The area heat transfer coefficient, H, can be calculated with the following equatiion (Duffie, 1980):

$$H = 108.3\alpha(d/\alpha)(G/d)^{0.7}/(1 - \varepsilon) \qquad (2.2)$$

for $60 < Re < 480$

where d - equivalent spherical diameter, m,

ε - void fraction of the rockbed, decimal,

α - shape factor of the rocks, decimal.

G - mass velocity of the fluid, $kg/m^3 \cdot s$.

$$G = \rho * v \qquad (2.3)$$

where ρ - fluid mass density, kg/m^3,

v - flow velocity, m/s.

Re is the particle Reynold's number

$$Re = G * d/\mu \qquad (2.4)$$

where μ - fluid absolute viscosity, $kg/m \cdot s$.

The void fraction ε, also known as 'porosity', is defined as the ratio of the void volume Voi to the container volome Vo filled with the sample rock pebbles.

$$\varepsilon = Voi / Vo \qquad (2.5)$$

The void volume is measured with the water replacement method, i.e. filling water into the sample rock container to replace the air in the void.

The equivalent spherical diameter (Duffie, 1980) is the diameter of a sphere particle having the same volume as the average particle volume in the rock sample. It can be calculated from

$$d = [1.91Vo(1 - \varepsilon)/ n]^{1/3} \qquad (2.6)$$

where n is the number of rocks in the sample.

The shape factor α is the ratio of the surface area of the pebble in the sample to the surface area of the equivalent sphere. It is difficult to evaluate. Duffie (1980) recommended that for crushed gravel α varies linearly from about 2.5 to about 1.5 as the pebble diameter increases from 5 to 50 mm, and for smooth river gravel α is approximately equal to 1.5 independent of the rock size. McCorquodale et al. (1977) measured an α value of 1.92 for crushed dolomite of 15.6 mm. No information was found for crushed limestone.

2.2.3 Heat Transfer in the Rockbed

Schumann (1929) developed the first analytical model to describe the temperature change rates for an incompressible fluid and a solid with respect to time and the location in a packed bed. Furnas (1932) expanded the fluid to gases.

$$\partial T_g/\partial X = H(T_r - T_g)/(C_g * v_g)$$

$$\partial T_r/\partial t = -H(T_r - T_g)/[C_r(1 - \varepsilon)] \qquad (2.7)$$

where X - distance along the flow direction, m,

H - area heat transfer coefficient, $W/m^2 \cdot k$,

v - flow velocity, m/s,

t - time, s,

T - temperature, K,

ε - void fraction,

C - volumetric heat capacity, $J/m^3 \cdot K$,

g - subscript for gases,

r - subscript for rock.

The negative sign in equation (2.7) indicates that the rock temperature decreases while discharging. Schumann presented a series of analytical solution curves, which, however, were of limited applications. Furnas (1932) observed that the area heat transfer coeffecient H is affected by fluid velocity, temperature, rock size and void fraction. It is somewhat difficult to prove Schumann's correlations by measuring the continuously changing temperature of both the fluid and the solid within a porous bed directly and simultaneously. In recent years computer programs with applied numerical methods have been developed to simulate the complicated transient thermal response of the rock storage. These, however, are very time-consuming. Due to the complexity of the heat transfer process and lack of exact knowledge of the this mechanism, the accuracy and adequacy of the analytical equations and numerical approaches are still being examined by many investigators.

2.2.4 Pressure Drop through Rockbeds

The pressure drop through a packed bed is due to both viscous and inertia drag (Ergun, 1952). Many researchers have worked on this problem.

Dunkle et al. (1976) recommended the following relationship:

$$\Delta P = (21 + 1750/Re)[L*G^2/(\rho*d)] \qquad (2.8)$$

Parker et al. (1978) stated that Dunkle's equation agreed closely with the behavior of compacted beds of crushed limestone, however, a loosely-filled rockbed exhibited a lower pressure drop than predicted from (2.8).

Duffie (1980) quoted Shewen's equation:

$$\Delta P = (4.24 + 166B/Re)[L*G^2*B/(\rho*d)] \qquad (2.9)$$

where B is a coefficient dealing with the void fraction and the shape factor α.

$$B = \alpha(1 - \varepsilon)/ \varepsilon^{1.5} \qquad (2.10)$$

Brownell et al. (1947) introduced another correlation between a modified Reynolds' number and a modified friction coefficient

$$\left\{ \begin{array}{l} Re = G*d*\varepsilon^{-m}/\mu \\ \\ f = 2d*\rho*\Delta P*\varepsilon^{n}/(L*G^2) \end{array} \right. \qquad (2.11)$$

where the exponents of both m and n depend upon the porosity ε and the shape factor α.

Leva (1959) proposed a pressure drop correlation.

$$\Delta P = [2L*G^2*f_m/(\rho*d)][(1 - \varepsilon)/\alpha]^{3-n}\varepsilon^{-3} \qquad (2.12)$$

where f_m was a modified friction coefficient, and a function of the particle Reynold's number. The exponent n in (2.12) referred to the fluid state factor, which varied from 1 for the laminar flow to 2 for the turbulent flow. In most turbulent flow cases an average n value of 1.9 was assumed.

These correlations above, as well as others not presented in this thesis, have respective limited applications. Each of them is a compromise leading to a final generalized correlation.

2.3 Heat Effectiveness and Moisture Removal Factor

2.3.1 Heat Recovery Factor and Effectiveness

In the previous sections the study of the microscopic process and mechanism of heat, momentum and mass transfer in the porous rockbed was reviewed. Besides that, the rockbeds as well as other heat exchangers have been studied from a macro viewpoint by examining their overall efficiency.

The term - "efficiency" - has different definitions in the literature reviewed. Some confusion arises when various results are compared. It is, therefore, desirable to define the efficiency in an appropriate way.

Since the air flows alternate in rockbeds, the air state parameters at various locations within the rockbed system fluctuate with respect to time. For analysis of long term operation, it is convenient to describe the states of the exhaust air, the incoming air, the outside air, etc. by their mean values over the given time period.

Fig. 2.1 shows a block diagram of an animal building with a rockbed regenerator. Consider the building subsystem. The building exhaust air is indicated with mass flow rate $\dot{M}_e$, enthalpy E_e and humidity ratio h_e. The building incoming air - M_i, E_i and h_i. The possible leakage

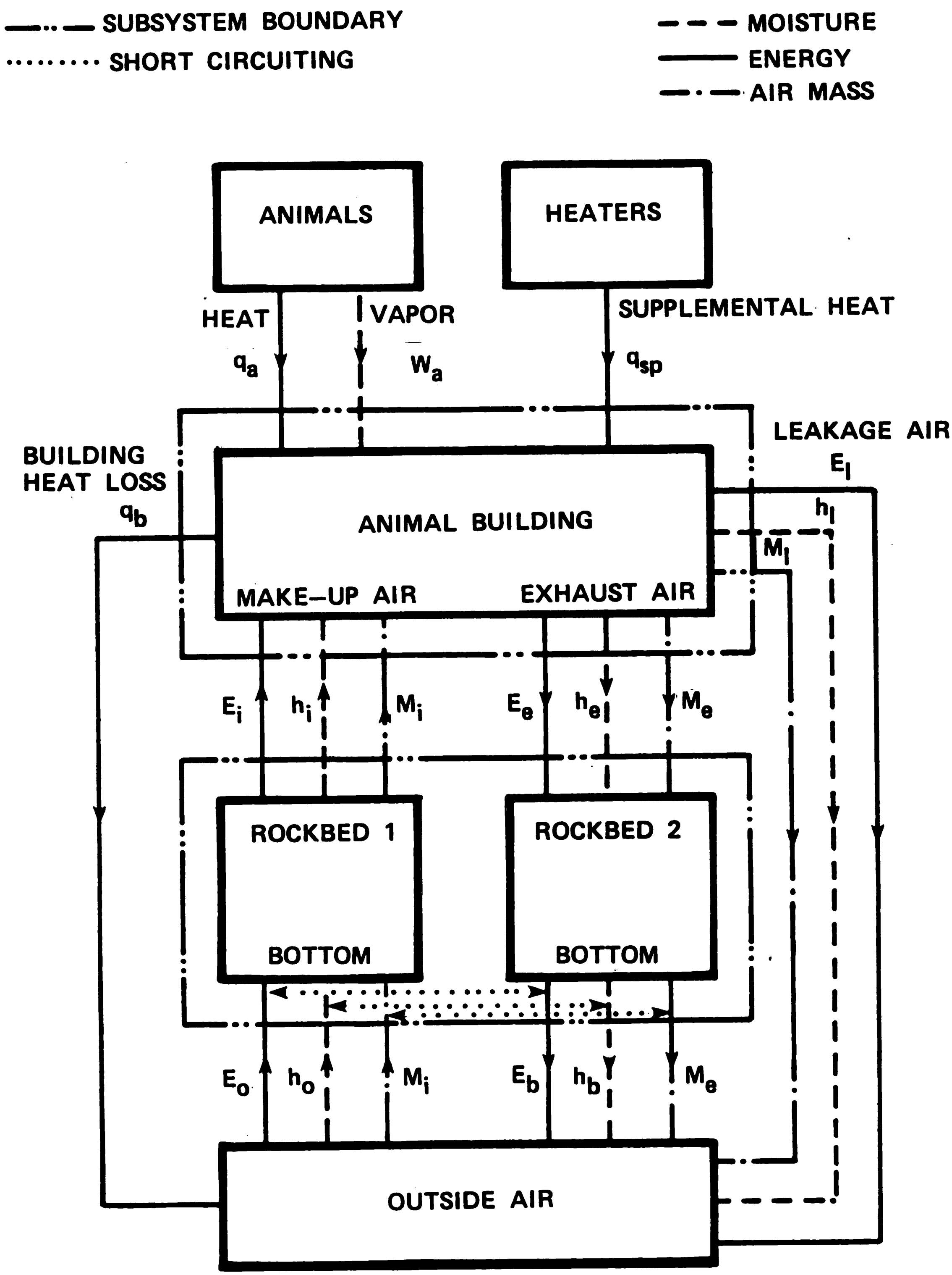


FIGURE 2.1 BLOCK DIAGRAM OF AN ANIMAL BUILDING SYSTEM WITH ROCKBEDS

air is represented with M_l, E_l and h_l. Both the exhaust and the incoming air are mechanically-forced while the air leakage is natural due to mass and pressure difference betwwen the building and the atmosphere. The building conductive heat loss is q_b. The building supplemental heat rate - q_{sp}. The total heat production rate by animals - q_a, which includes both sensible and latent heat. The vapor production rate in the building - W_a, which is the sum of the rate of vapor released by animals and the rate of evaporation from urine, feces and other wet surfaces (Bond et al., 1959).

Within the rockbed subsystem boundary, including ducts, there are two pairs of inputs and outputs. The building exhaust air is an input. The cooled exhaust air leaving the rockbed bottom is its corresponding output, indicated with M_e, E_b and h_b. The outside cold air, represented with M_i, E_o and h_o, is the other input. The building incoming air is its corresponding output.

The product of the incoming mass flow rate and the enthalpy difference between the incoming air and the outside air represents the actually recovered energy rate. Likewise, the recoverable energy rate is the product of the exhaust mass flow rate and the enthalpy difference between the exhaust air and the outside air. The "heat recovery factor" Rf for the rockbed subsystem is here defined as the ratio of the recovered energy rate to the recoverable energy rate for a given time period.

$$Rf = M_i(E_i - E_o)/[M_e(E_e - E_o)] \quad (2.13)$$

where M - mass rate of the air flow, kg dry air/s,

E - air enthalpy, kJ/kg dry air,

i - subscript for the building incoming air,

e - subscript for the building exhaust air,

o - subscript for the air outside the building.

Enthalpy is used here because there is some transfer between sensible heat and latent heat when condensation and evaporation of water occur on the rock surfaces.

A mass flow rate M is the product of the volumetric flow rate and its density. The building incoming mass rate M_i is not equal to the exhaust mass rate M_e, when volumetric rates, Q_i and Q_e, are assumed equal due to temperature difference. If M_i is larger than M_e, there should be air leakage outgoing to preserve a mass balance. The leakage air carries unrecoverable heat.

In order to reflect the inherant heat recovery efficiency of the building subsystem, the "heat recovery effectiveness" η is defined as the ratio of the actual recovered energy rate to the maximum recoverable energy rate for the system. On the mean value basis, the maximum recoverable energy rate is the product of the total mass flow rate leaving the building and the enthalpy difference between the outgoing air and the outside air. The total outgoing flow mass rate equals either M_e or M_i, whichever is larger. It is denoted by M_{max}. Then, η is described as

$$\eta = M_i(E_i - E_o)/[M_{max}(E_e - E_o)] \quad (2.14)$$

Heat exchanger effectiveness has been introduced in many heat transfer textbooks. In these textbooks, only the simultaneous-type heat exchangers, such as plate-type, shell-and-tube type and so on, have generally been discussed. Sokhansanj et al. (1980) introduced the term of the heat recovery factor for a simultaneous-type heat regenerator based on the sensible heat differences. If there are some phase changes in fluids, e.g. evaporation or condensation water in air, occuring in these exchangers, latent heat should be taken into consideration. The enthalpy term seems quite appropriate to calculate efficiency for these cases.

2.3.2 Moisture Removal Factor

As a portion of energy is recovered and sent back to the building, an amount of water vapor may also return back to the building. To evaluate the moisture removal from the building subsystem due to the effect of the heat exchanger the "moisture removal factor" is defined here as

$$\beta = 1 - \{ M_i(h_i - h_o)/[M_{max}(h_e - h_o)] \} \quad (2.15)$$

where $M_i(h_i-h_o)$ - rate of moisture retured to the building

$M_{max}(h_e-h_o)$ - maximum removable moisture rate.

If short circuiting occurs between adjacent rockbed chambers, the cooled exhaust air leaves one rockbed bottom and, then, enters the other bed bottom. Thus, the air state at the bottom of rockbeds should be used instead of the outside air state in calculation.

3 EXPERIMENT

3.1 Conditions and Environment

A field study on a rockbed heat exchanger was carried out in the west room of the farrowing unit at Michigan State University Swine Research Center.

The farrowing unit of M. S. U. Swine Research Center consisted of two rooms of 14.5 x 7.6 x 1.7 m with a shed roof of an 1:5 slope (Fig. 3.1). Sixteen farrowing crates were linked into two rows in the west room. As a whole, 75 % of the floor is slotted. An underfloor deep pit accumulates manure before it is pumped out about every two months. The farrowing unit was not well insulated. Its exposure factor, EF, was 13.2 W/C per animal unit for 16 sows and litters. This EF value was much higher than that of 0.5 to 2.6 W/C per animal unit recommended in ASAE D270.4 (1979). A perforated duct channel was constructed as an air distributor for a solar collector. However, the solar heating system was inoperative during this study, due to most cloud days. A conventional ventilation system, consisting of winter, spring-fall and summer fans, was not used during this study. A gas heater was the main heating device in the room.

Baby pigs should be kept in a warm, dry and draft-free

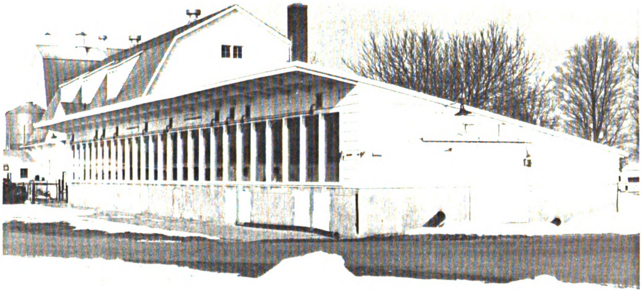


FIGURE 3.1 FARROWING UNIT OF M.S.U. SWINE RESEARCH CENTER

environment (MWPS-1, 1980). In practice, the indoor air temperature is maintained between 18 to 27°C (Curtis, 1978), plus zone heating in the piglet creep areas (MWPS-1, 1980). Indoor air relative humidity of 40 to 80 % (Sainsbury, 1974), and air movement velocities below 0.25 m/s (Sainsbury, 1979) are desired.

3.2 Experimental Facility

A regenerative ventilation system was constructed on the north side out of the west farrowing room (Fig. 3.2). Two rockbed chambers were adjacent to each other under a common frame of 1.9 x 1.7 x 2.4 m with a 1:4 gable roof. The structure illustrated in Fig. 3.3 was of 9.5 mm plywood sheathing applied over 100 mm square post framing. The inner side-wall and ceiling were of 12.7 mm plywood. The two chambers were insulated with styrofoam of 1.8 $m^2 \cdot C/W$ thermoresistance over the attic and the four side walls. Two layers of 0.1 mm polyethylene film was used over the insulation material to provide a vapor barrier. Frames, made of 25.4 x 25.4 x 4.8 mm angle steel and 25.4 x 3.2 mm flat iron with 25.4 mm flattened expanded sheet metals, supported the rockbeds. The rockbed bottom was 0.9 m above the ground. Plywood skirts of 0.3 m wide were around the bottom of the support frames. A 254 mm square opening was on the upper west side of each chamber as an entrance for the building exhaust air into the rockbed chamber. Likewise, an opening of the same size was on the upper east

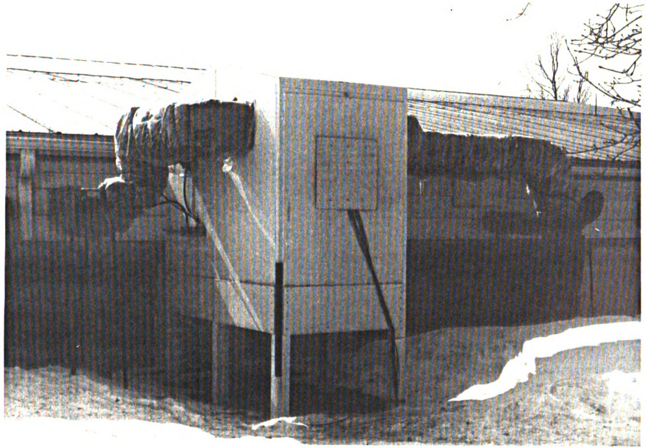


FIGURE 3.2 VENTILATION SYSTEM WITH ROCKBED REGENERATOR

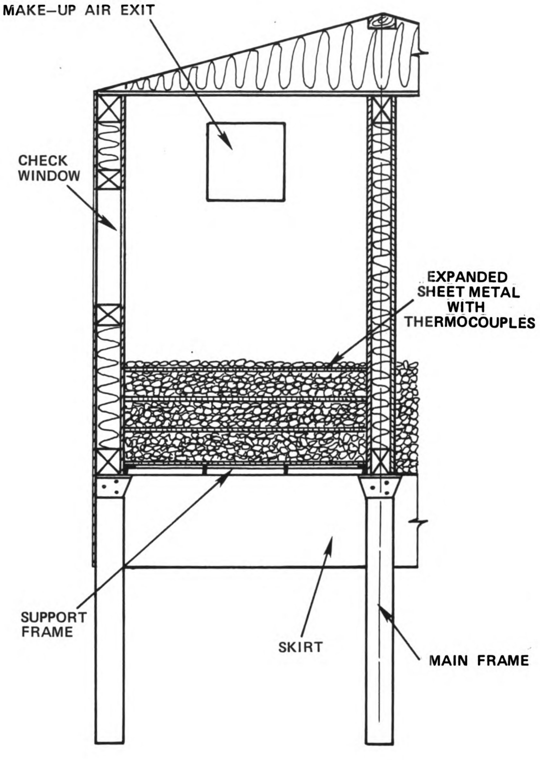


FIGURE 3.3 WEST SIDE VIEW OF NORTH ROCKBED CHAMBER

side of each chamber as an exit for the warmed building make-up air out the chamber.

Limestone rocks with an average size of 16 mm were screened with a #4 mesh. The pebbles, not passing through the screen, were packed into the rockbed randomly.

Fig. 3.4 shows the schematic plan view of the regenerative ventilation system. Both the exhaust and the make-up fans were blowers directly driven by totally-enclosed 0.186 kW motors with double speeds of 1140/1725 rpm. Two switches were used to change fan speeds manually. A 254 mm square two-way duct and the exhaust blower connected a 152 mm room outlet pipe and the air entrances of the chambers. Likewise, the make-up blower and a 254 mm square two-way duct linked with the air exits of two chambers and a room inlet pipe of 254 mm diameter. In the farrowing room, a 254 mm vinyl coated flexible hose along the east wall transported the incoming air from the room inlet to the solar collector distributor (Fig. 3.5).

There were four shutters in the 254 mm square ducts. Every shutter was controlled by a damper motor (Fig. 3.6). Opening and closing these dampers were arranged so that the building exhaust air flowed downwards through one rockbed while the make-up air flowed upwards through the other rockbed. A clock-operated timer switched on and off these four dampers to alternate cycles. Fan speed switches and the timer were mounted in a control box (Fig. 3.7).

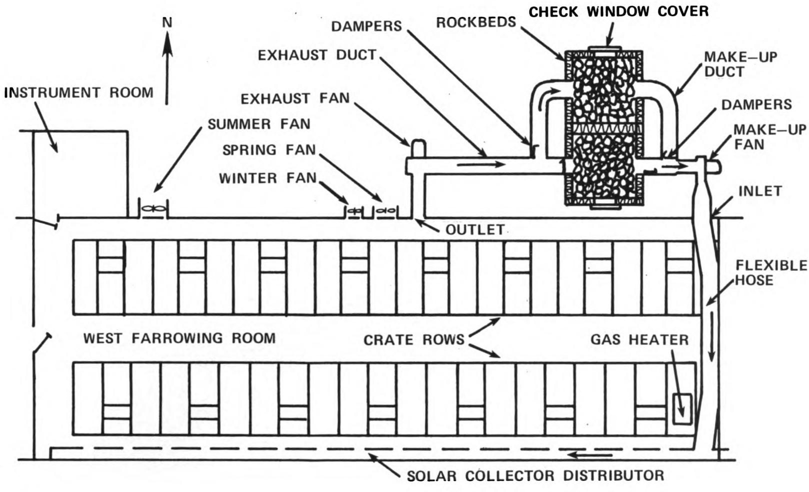


FIGURE 3.4 SCHEMATIC PLAN VIEW OF WEST FARROWING ROOM WITH ROCKBED REGENERATOR

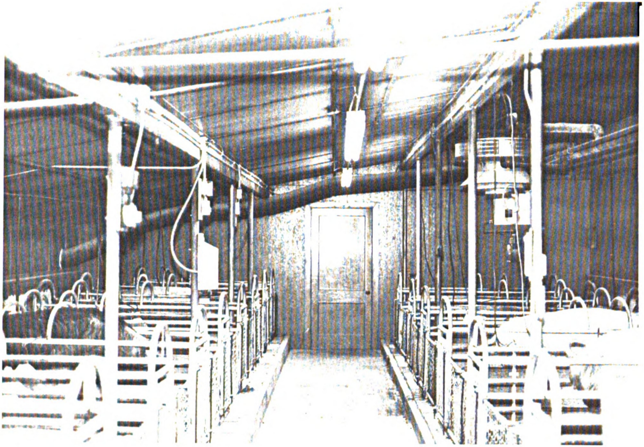


FIGURE 3.5 FLEXIBLE HOSE CONNECTING INLET AND AIR DISTRIBUTOR

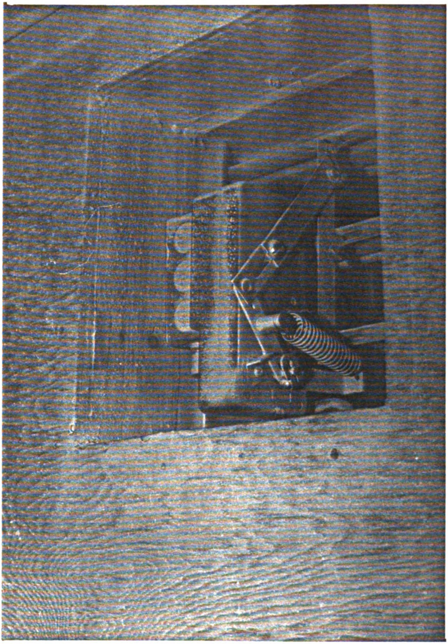


FIGURE 3.6 DAMPER AND MOTOR

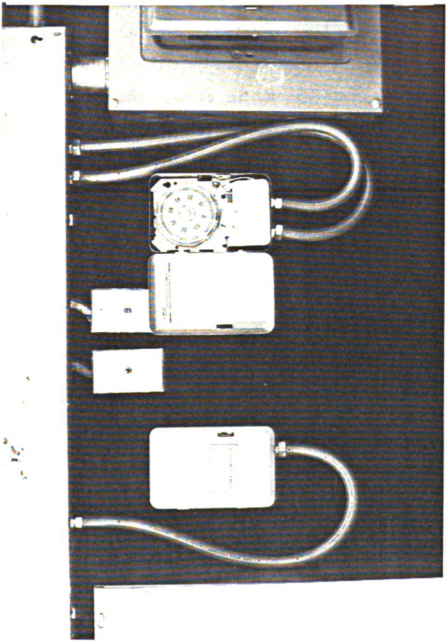


FIGURE 3.7 TIMER AND FAN SPEED SWITCHERS IN CONTROL BOX

3.3 Method of Study

The rockbed regeneraor was continuously operated while the farrowing room was full with 16 sows and litters from February through April in 1982. Measurements were taken from February 10th through March 4th.

Four design parameters of the rockbed regenerator were tested to evaluate their effects on heat recovery and moisture removal for the building subsystem. The ranges of the four parameters studied were

1. Rockbed depth: 0.30, 0.41 and 0.51 m,
2. Cycle time: 2, 4, 6, 8, 10 and 12 min,
3. Exhaust fan speed: 1140 and 1725 rpm,
4. Make-up fan speed: 1140 and 1725 rpm.

It is desired to test flow rates of both the exhaust and the make-up air. However, these flow rates were dependent, affected by the fan speeds, the rockbed depth and the resistance in the ducts. So the two independent factors - the fan speeds - were used. This was a factorial experiment of four factors.

The following data were measured in this study :

1. Dry- and wet-bulb temperatures at the farrowing room outlet,

2. Dry- and wet-bulb temperatures at the farrowing room inlet,

3. Dry- and wet-bulb temperatures at the bottom of the north sample rockbed chamber,

4. Dry- and wet-bulb temperatures of the outside air,

5. Dry-bulb temperatures at the exhaust air entrance and the make-up air exit in the sample rockbed chamber,

6. Temperatures at each 100 mm depth level within the sample rockbed,

7. Static air pressure above and below the sample rockbed,

8. Air flow velocity in the farrowing room outlet and inlet pipes.

The experiment was scheduled according to an orthogonal table. For each treatment combination of four factors, the rockbed system was first operated about 2 hours to obtain a steady state. During this period, only air temperatures at various locations were recorded at an interval of several minutes depending on the cycle time used. Then, during a second period about 40 min the recording interval was set to 10 sec and measurements of humidity, velocities and static pressure were taken. During the night, the temperatures were monitored at 5 to 15 min interval.

3.4 Measurement and Instrumentation

The following instruments were used to carry out measurements, data acquisition and calibration:

1. WEATHERtronics Hot Wire Anemometer Model 2440,
2. Copper-Constantan Thermocouple Wires TQAN #24,
3. ATKINS Thermocouple Psychrometer Model 90023,
4. DWYER Manometer Model 25,

5. KAYE Digistrip II Multipoint Recorder,
6. DIGI-CAL AN6520 Calibrator.

3.4.1 Dry Bulb Temperature Measurement

Dry-bulb temperatures were sensed by thermocouples at 16 locations. The thermal electromotive forces in microvolts from these sensors were converted to temperatures and printed out on chart papers in digital form by the multichannel recorder. Table 3.1 shows the recorder channel allocation.

Three or five pairs of thermocouple wires were mounted separately on each piece of 12.7 mm flattened expanded sheet metals. The thermojunction of these PVC-coated wires did not touch the metal material. One thermocouple was at the center of each sheet. Others were placed 100 to 380 mm away around the center. Thermocouples were also mounted on the expanded sheet metal at the bottom of the sample rockbed in th same way. One metal sheet with mounted thermocouples was placed in the center of the rockbed plane at each 100 mm thickness of stones. The sheet on the rockbed top was covered with an approximately 20 mm thick layer of rocks. Eleven recording channels were allocated to these thermocouples. At least, the center thermocouple of each sheet was connected to a recording channel. Table 3.2 shows the channel allocation among layers.

The Digistrip II recorder had an accuracy of +(0.003 %

reading + 0.3 °C) with Copper-Constantan thermocouples for temperature monitoring between -219 to 402°C. Its scanning speed was 8 channels per second. It required an ambient environment of 0 to 50°C and 0 to 95 % RH with no condensation. The recorder was installed in a small cabin (Fig. 3.8) next to the farrowing room and about 15 m away from the rockbed. The recorder was calibrated with a DIGI-CAL AN6520 calibrator.

3.4.2 Humidity Measurement

Air humidity can be determined by measuring the dry- and wet-bulb temperatures of the air steam. Wet-bulb depression is the temperature difference between the dry-bulb and the wet-bulb. The ATKINS psychrometer used was a hand-held microvolt thermometer with LCD digital display. Two fine-wire K-type thermocouples sensed the dry-bulb and wet-bulb temperature, respectively. The readout was the dry-bulb temperature and the wet-bulb depression. Its pistol-shape made it easy to measure air humidity in a duct through a small hole.

The ATKINS psychrometer was accurate to within 0.5 % reading for the range of 0 to 100°C and 0 to 100 % RH. It was calibrated with the DIGI-CAL AN6520 calibrator. Incorrect readout of the wet-bulb depressions occurred when the wet-bulb temperature dropped below -5°C. The measured data were compared with current local daily climate data on three-hour basis as a check.

Table 3.1 Channel Allocation on Digistrip II Recorder

Channel Number	Thermocouple Locations
1 - 11	Bottom to Top in Rockbed
12	Rockbed Chamber Entrance
13	Rockbed Chamber Exit
14	Room Outlet Pipe
15	Room Inlet Pipe
16	Room Outside

Table 3.2 Channel Number Allocation within the Rockbed

Layer Level	Rockbed Depth (m)		
	.30	.41	.51
0	# 1 - 2	# 1 - 2	# 1 - 2
1	# 3 - 4	# 3 - 4	# 3 - 4
2	# 5 - 7	# 5 - 6	# 5 - 6
3	# 8 - 11	# 7 - 8	# 7 - 8
4		# 9 - 11	# 9 - 10
5			# 11

Note : Level 0 - Rockbed Bottom.

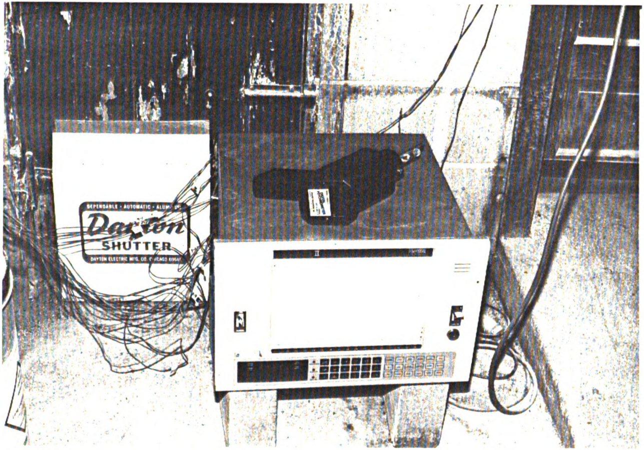


FIGURE 3.8 DATA LOGGER AND PSYCHROMETER

3.4.3 Air Flow Measurement

The volumetric rate of an air flow in a pipe is the product of the inside section area of the pipe and the mean normal velocity of the air flow through the area. The mean normal velocity of air flow was taken as an average of 10 velocities measured in 5 concentric ring sections of equal area. The velocity measurements were taken several times for each combination of rockbed depth and fan speeds.

The hot wire anemometer had an accuracy of 3 % full scale for both 0 to 5 m/s and 0 to 40 m/s.

3.4.4 Static Pressure Measurement

A simple DWYER inclined manometer was used in this study. The manometer could handle 0 to 0.75kPa.

There was a tubular hole on the check window cover of the sample rockbed chamber. One of two tubular probes of the manometer was injected into the upper space above the rockbed with the other probe open to the atmosphere. The reading on the manometer scale was the static pressure difference between the atmosphere and the space above the sample chamber. When the air flow direction was reversed the two probes were also exchanged in order to read positive pressure differences. Measured static pressure data were grouped and averaged for 12 combinations of rockbed depth and two fan speeds.

4 ORTHOGONAL EXPERIMENT DESIGN

4.1 Introduction

The major analysis of this study was through a factorial experiment of $6 \times 3 \times 2^2$. A full factorial would call for 72 treatment combinations, and the complete analysis of the experiment would ideally require several replicates of the full factorial. It would be very time-consuming, costy and, even, inpractical. An orthogonal design was, therefore, applied in this study. It involved a half-fractional factorial experiment to estimate the main effects of the four factors. According to its results one or more of the following analysis could be performed :

1. Determining the main effects and significance of these factors tested.

2. Identifying the factors which have appreciable effects for further close study; and releasing the remaining factors with little effects, or replacing them with new factors.

3. Selecting the optimum combination of design parameters within the ranges tested, if all interactions of factors, suppressed in residual with errors, were negligible.

4. Predicting the necessity of replicating a full

factorial experiment in further study, if interactions were competitive with main effects of these factors tested.

Orthogonal designs are based on orthogonal arrays of strength two, which form orthogonal main effect plains (John, 1971). Orthogonal designs can be expressed in matrix form, which are named for "orthogonal tables" (Wang et al., 1979). In an orthogonal table the rows identify the treatment combinations; the columns - the factors; and the elements - the levels of the respective factor at the respective run.

The major properties of the orthogonal tables are summarized as follows (CAAMS, 1978) :

1. Every level of any individual factor appears for the same number of times in its column.

2. Every treatment combination of any factor pair appears for the same number of times in respective column pair.

3. Permutation of rows, columns or level notations in a column does not affect the orthogonality of the table.

4. If an orthogonal design is a fraction of a factorial experiment, the remaining fractional factorial also forms an orthogonal array.

5. Many orthogonal tables can handle some low-order interactions while some can not handle any interactions.

6. Total number of factors and interactions tested should not exceed the number of columns in an orthogonal table.

4.2 Orthogonal Table - $OA_{36}(6 \times 3 \times 2^2)$

Table 4.1 shows the orthogonal table constructed for this study. The table is designated by $OA_{36}(6 \times 3 \times 2^2)$, where the subscript of 36 indicates the row dimension, the expression of 6 x 3 x 2 - the number of levels for individual factors and the superscript of 2 - the number of factors having the same respective number of levels.

It is noticed that :

1. This orthogonal table was one-half fraction of the full factorial experiment.

2. All of four factors in this table were independent and controllable.

3. All levels of each factor were equally spaced over its respective range.

4. This table could not estimate any interactions. All interactions suppressed each other in the residual.

5. It was desired to conduct these runs in a randomized order, although the table was organized in standard order form. In this study other factors varied in random ways, except the factor of rockbed depth changed in an increasing order for some convenience.

Table 4.1 Orthogonal Table $OA_{36}(6 \times 3 \times 2^2)$

Run	Factors			
	Rockbed Depth (m)	Cycle Time (min)	Exhaust Fan Speed (rpm)	Make-up Fan Speed (rpm)
01	0.30	2	1725	1140
02	0.30	2	1140	1725
03	0.30	4	1725	1725
04	0.30	4	1140	1140
05	0.30	6	1725	1140
06	0.30	6	1140	1725
07	0.30	8	1725	1725
08	0.30	8	1140	1140
09	0.30	10	1725	1140
10	0.30	10	1140	1725
11	0.30	12	1725	1725
12	0.30	12	1140	1140
13	0.41	2	1725	1725
14	0.41	2	1140	1140
15	0.41	4	1725	1140
16	0.41	4	1140	1725
17	0.41	6	1725	1725
18	0.41	6	1140	1140
19	0.41	8	1725	1140
20	0.41	8	1140	1725
21	0.41	10	1725	1725
22	0.41	10	1140	1140
23	0.41	12	1725	1140
24	0.41	12	1140	1725
25	0.51	2	1725	1140
26	0.51	2	1140	1725
27	0.51	4	1725	1725
28	0.51	4	1140	1140
29	0.51	6	1725	1140
30	0.51	6	1140	1725
31	0.51	8	1725	1725
32	0.51	8	1140	1140
33	0.51	10	1725	1140
34	0.51	10	1140	1725
35	0.51	12	1725	1725
36	0.51	12	1140	1140

5 RESULTS AND DISCUSSION

During the test period the outside climate temperature averaged -5.8 °C and fluctuated between -19.7 and 6.1°C. The atmospheric relative humidity varied from 56 to 100 % with an average of 82 %. The room temperature was maintained between 17 and 26 °C. The room relative humidity averaged 47 % with a range of 29 to 57 %.

5.1 Basic Character of This Study

5.1.1 Rock Equivalent Diameter and Void Ratio

The equivalent spherical diameter of rocks was 15.8 mm and the void ratio α was 0.507. The values were obtained with 3 samples by using the water replacement method. The ratio of the rockbed depth and the side length to the rock equivalent spherical diameter were from 19 to 32, and 51, respectively. Therefore, no edge effect was assumed in this study, based on Rose's results mentioned in Section 2.2.1.

5.1.2 Air FLow Rates

The volumetric flow rates (Table 5.1) varied as the rockbed depth and the fan speeds changed, and were also subject to the resistance in the exhaust and the make-up ducts. The flow rates at low speed for both fans provided

Table 5.1 Air Flow Rates (m^3/s)

Rockbed Depth (m)	Exhaust Fan Speed		Make-up Fan Speed	
	1725 rpm	1140 rpm	1725 rpm	1140 rpm
.30	.167	.111	.184	.125
.41	.162	.110	.172	.118
.51	.159	.108	.165	.113

Table 5.2 Comparison of Static Pressure Drop

Rockbed Depth (m)	Air Flow Rate (m^3/s)	Pressure Drop Measured (Pa)	Pressure Drop Calculated (Pa)
.30	.184	37	53
	.167	35	44
	.125	17	26
	.111	16	22
.41	.172	50	63
	.162	45	56
	.118	19	32
	.110	17	28
.51	.165	60	73
	.159	55	67
	.113	24	38
	.108	21	35

ventilation below the minimum winter ventilation rate of 0.142 m^3/s for 16 sows and litters, suggested in MWPS-1 (1980). However, some air leaked out from the west farrowing room through the underfloor channel due to a pressure difference between the two farrowing rooms when a fan was operated at a high speed in the east farrowing room.

The superficial velocity of the air flow was from 0.17 to 0.28 m/s with respect to the rockbed cross section of 0.64 m^2. These values were below 0.42 m/s as suggested by Moysey (1980). The mass flow rate of the exhaust air varied from 0.12 to 0.19 kg/s while that of the make-up air was between 0.13 and 0.22 kg/s. These measurements showed that the mass flow rate of the room make-up air was slightly higher than that of the exhaust air with both fans at the same speed, either 1725 or 1140 rpm.

5.1.3 Static Pressure Drop

Table 5.2 shows a comparison between the meassured static pressure drop and the calculated values from Shewen's equation (2.10). In the calculation, a shape factor α of 2.0, the void ratio ε of 0.507 and the equivalent diameter of 0.0158 m were used. It was seen that the measured static pressure drop was lower than that calculated. Flattened expanded metal sheets put into the rockbed might be a reason for this pressure drop reduction, since these metal sheets modified the bulk rockbed structure and air flow was redistributed within the sample rockbed. These tested values were in the range of 50 to 100 Pa recommended by Parker et al. (1981).

5.1.4 Reynold's Number and Biot Number

The particle Reynold's number was between 168 and 310 in this study, which was in the range of 60 to 480. When a shape factor α of 2.0 was assumed, the convection heat transfer coefficient H was of 10 to 15 $W/m^2 \cdot K$, calculated with equation (2.2). The specific heat of limestone is 2.15 $W/m \cdot K$. The resulting Biot number was from 0.037 to 0.055, which was less than 0.1. Hence, a lumped system approach was appropriate (Section 2.2.2).

5.2 Results of the Orthogonal Table

Data processing was carried out at Albert H. Case Computer Center, College of Engineering, M. S. U. Recorded data were processed with a Fortran program, which also formed graphs from analyzed data. A Fortran computer model by Lerew (1972) was used to find psychrometric properties of the moist air. The air properties were applied to calculate the heat recovery effectiveness η, the moisture removal factor β and the heat recovery factor Rf. Some short circuiting occurred in the study, since two adjacent rockbeds were not blocked up well from their bottom. Resultant values of η, β and Rf are listed in Table 5.3.

Table 5.3 Orthogonal Table Results

Run	Factors				Results		
	Rockbed Depth (m)	Cycle Time (min)	Exhaust Fan Speed (rpm)	Make-up Fan Speed (rpm)	η	β	Rf
01	0.30	2	1725	1140	.36	.67	.36
02	0.30	2	1140	1725	.43	.60	.73
03	0.30	4	1725	1725	.65	.40	.73
04	0.30	4	1140	1140	.63	.43	.72
05	0.30	6	1725	1140	.56	.49	.56
06	0.30	6	1140	1725	.53	.51	.90
07	0.30	8	1725	1725	.72	.34	.81
08	0.30	8	1140	1140	.68	.40	.77
09	0.30	10	1725	1140	.52	.51	.52
10	0.30	10	1140	1725	.59	.47	1.0
11	0.30	12	1725	1725	.76	.31	.84
12	0.30	12	1140	1140	.74	.31	.84
13	0.41	2	1725	1725	.65	.41	.72
14	0.41	2	1140	1140	.66	.39	.73
15	0.41	4	1725	1140	.50	.53	.50
16	0.41	4	1140	1725	.60	.46	.97
17	0.41	6	1725	1725	.69	.39	.75
18	0.41	6	1140	1140	.63	.49	.70
19	0.41	8	1725	1140	.57	.47	.57
20	0.41	8	1140	1725	.58	.54	.81
21	0.41	10	1725	1725	.78	.26	.84
22	0.41	10	1140	1140	.81	.24	.88
23	0.41	12	1725	1140	.58	.46	.58
24	0.41	12	1140	1725	.63	.42	1.0
25	0.51	2	1725	1140	.50	.52	.50
26	0.51	2	1140	1725	.59	.46	.93
27	0.51	4	1725	1725	.74	.31	.79
28	0.51	4	1140	1140	.76	.29	.81
29	0.51	6	1725	1140	.54	.49	.54
30	0.51	6	1140	1725	.58	.46	.92
31	0.51	8	1725	1725	.71	.34	.76
32	0.51	8	1140	1140	.74	.30	.79
33	0.51	10	1725	1140	.47	.57	.47
34	0.51	10	1140	1725	.56	.50	.88
35	0.51	12	1725	1725	.71	.34	.76
36	0.51	12	1140	1140	.70	.36	.76
				Average :	.60	.43	.74

5.2.1 Main Effects

The main effects of each factor are shown graphically in Fig. 5.1. Analysis of variance is tabulated in Table 5.4. All interactions aliased each other with random errors in the residual.

The F-test in Table 5.4 showed that the cycle time was most significant for both the heat recovery effectiveness η and the moisture removal factor β, however, only with 75 % confidence of significance because the interactions were strong. Rockbed depth was of little importance for η, β or Rf within the tested range. Fan speeds had little effect on η or β, but significantly effect on Rf. Cycle time was the third important factor for Rf.

Fig. 5.1 (b) shows a tendency for an increase in η and Rf as the cycle time increased. Large values of η observed were with 10 to 12 min cycle time in this study. This agreed with the range of 10 to 15 min suggested by Witz et al. (1979). In Moyseys' study (1980) both increase and decrease in the efficiency were observed when the cycle time reduced from 12 to 8 min. Results of this study were based on the statistical analysis rather than a single comparison. Fig. 5.1 (a) shows that from 0.4 m up the influence of the rockbed depth on η and β was negligible.

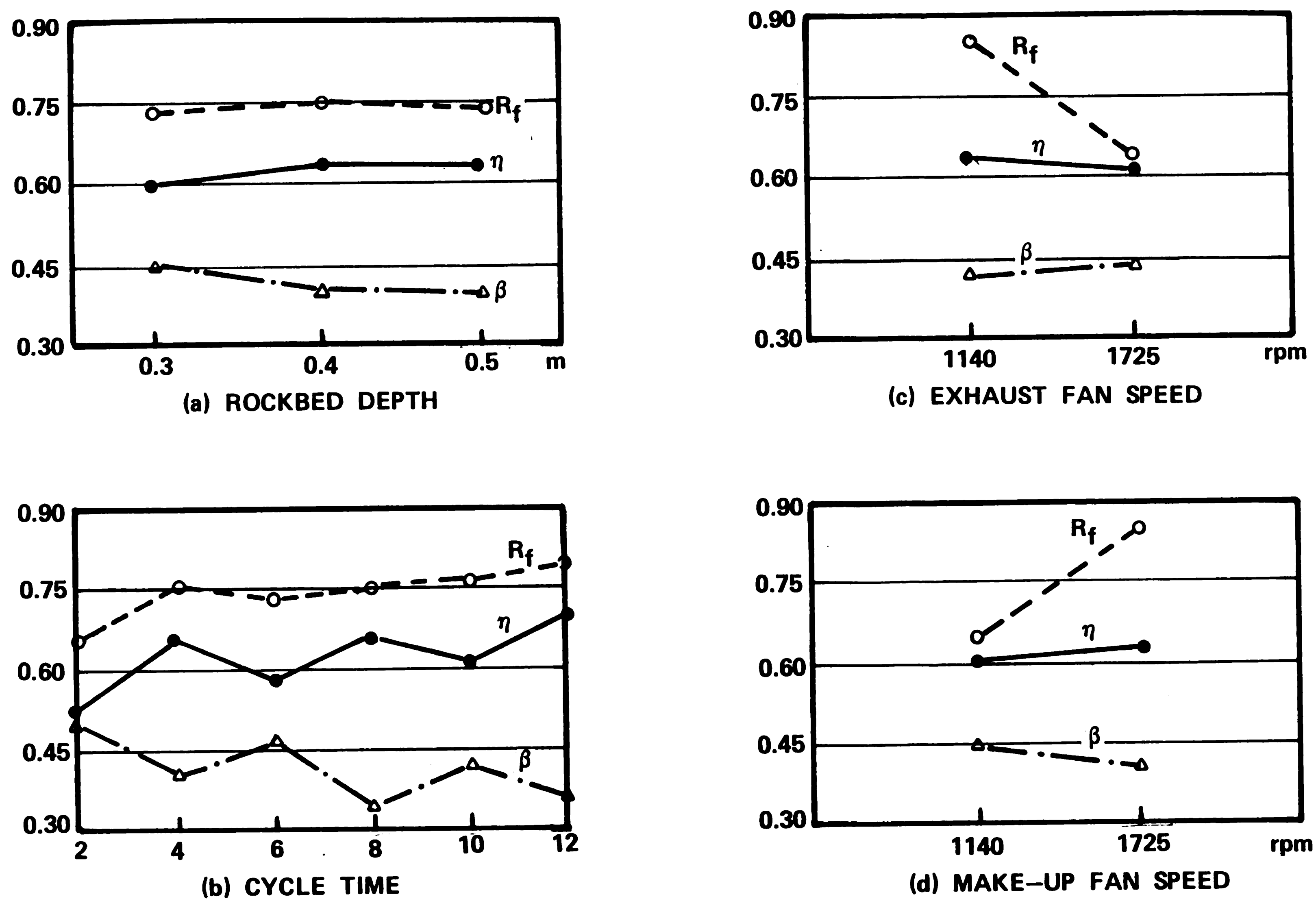


FIGURE 5.1 MAIN EFFECTS OF FACTORS

Table 5.4 Analysis of Variance

Term	Source	SS	d.f.	MS	F-Ratio
	Rockbed Depth	.01119	2	.00559	.49
	Cycle Time	.09174	5	.01825	1.61
η	Exhaust Fan Speed	.00314	1	.00314	.28
	Make-up Fan Speed	.00593	1	.00593	.52
	Residual	.29467	26	.01133	
	Total	.40616	35	.01160	
	Rockbed Depth	.01150	2	.00575	.57
	Cycle Time	.08054	5	.01611	1.60
β	Exhaust Fan Speed	.00116	1	.00116	.11
	Make-up Fan Speed	.00401	1	.00401	.40
	Residual	.26248	26	.01010	
	Total	.35968	35	.01028	
	Rockbed Depth	.00340	2	.00170	.36
	Cycle Time	.06526	5	.01305	2.78
Rf	Exhaust Fan Speed	.35106	1	.35106	74.69
	Make-up Fan Speed	.35264	1	.35264	75.03
	Residual	.12209	26	.00470	
	Total	.89445	35	.02556	

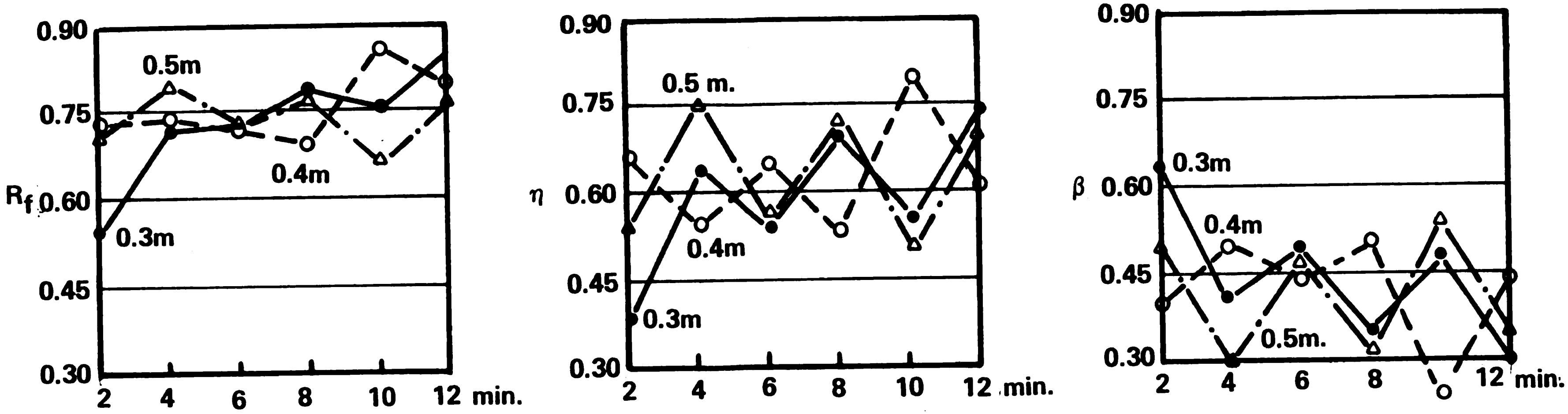
Note : 1. SS - Sum of Square of deviations about a mean
d.f. - Degrees of Freedom,
MS - Mean of Sum of Square of Deviations
2. $F_{0.05}(1,26) = 4.22$, $F_{0.05}(2,26) = 3.37$, $F_{0.05}(5,26) = 2.59$.
3. $F_{0.05}(1,26) = 1.38$, $F_{0.05}(2,26) = 1.46$, $F_{0.05}(5,26) = 1.42$.

5.2.2 Interactions

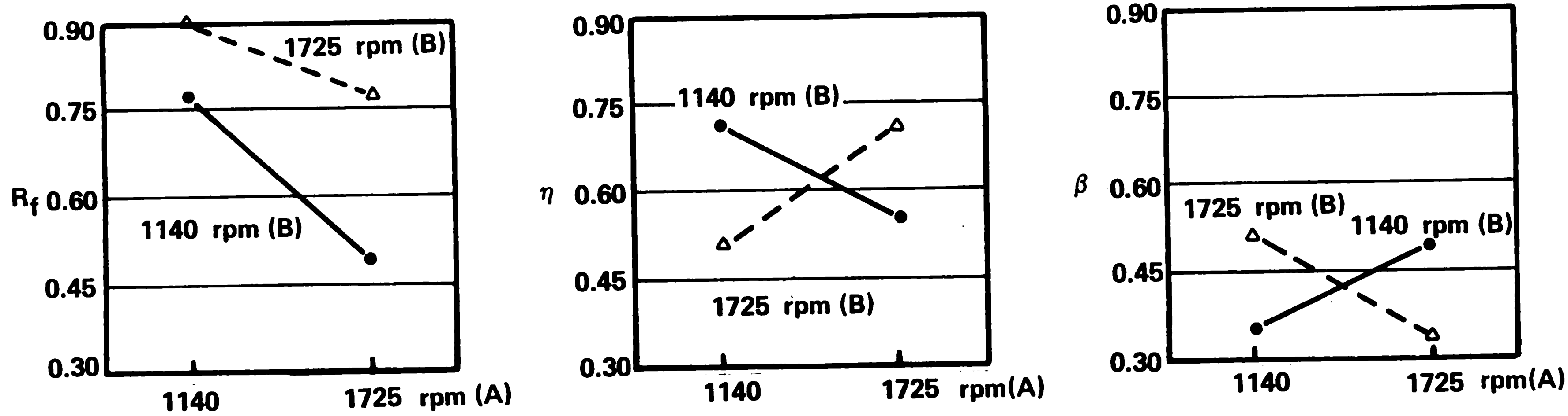
Fig. 5.2 illustrates six two-factor interactions, though it was impossible to study them mathematically.

There was a strong interaction of two fan speeds, i.e. two air flow rates. When two fan speeds differed very much from each other, the heat recovery effectiveness η averaged 0.5 to 0.55 (Fig. 5.2 (b)). When the room exhaust fan was at 1725 rpm and the room make-up fan was at 1140 rpm, the high exhaust flow rate gave much heat to the rocks, however, the low make-up flow rate recovered less heat from the rocks. This lowerred η down to 0.36 through 0.61. In this case, some cold outside air leaked into the room, which required more supplemental heat. When the exhaust fan was at 1140 rpm but the make-up fan was at 1725 rpm, less heat was transferred to the rocks by the low exhaust flow rate, and some warm indoor air leaked out the room. In this case, η dropped to 0.43 through 0.63. When both fans were at 1725 rpm, η was from 0.65 to 0.78. Likewise, in the case of 1140 rpm speed for both fans, the heat recovery effectiveness was from 0.63 to 0.81. This strong negative interaction of the two fan speeds was the major source causing fluctuations in response curves of cycle time in Fig. 5.2 (b) and Fig. 5 (a), (c) and (d).

The interactions of rockbed depth with either the exhaust fan or the make-up speed were negligible on η and β (FIg. 5 (e) and (f)). So was the interaction of the cycle time with either the exhaust fan or

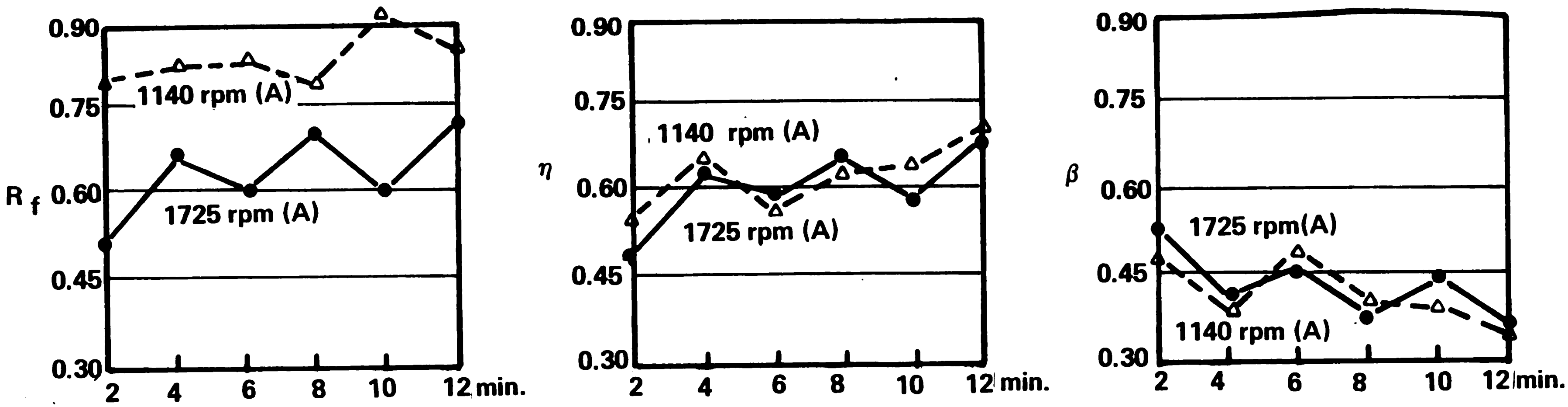


(a) INTERACTION OF CYCLE TIME (min.) WITH ROCKBED DEPTH (m)

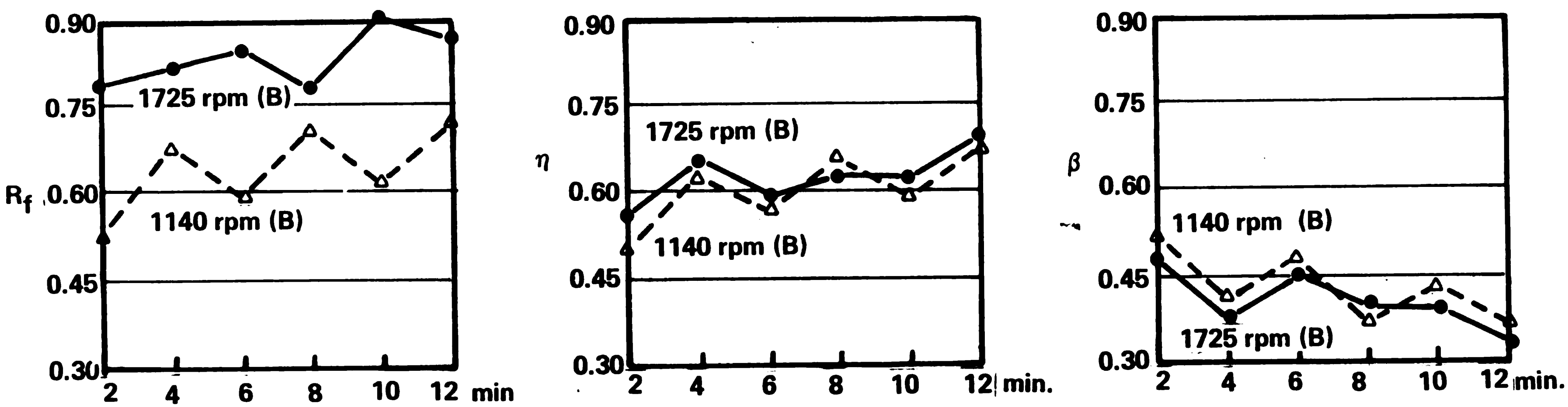


(b) INTERACTION OF EXHAUST FAN SPEED A WITH MAKE–UP FAN SPEED B

FIGURE 5.2 TWO–VARIABLE INTERACTIONS

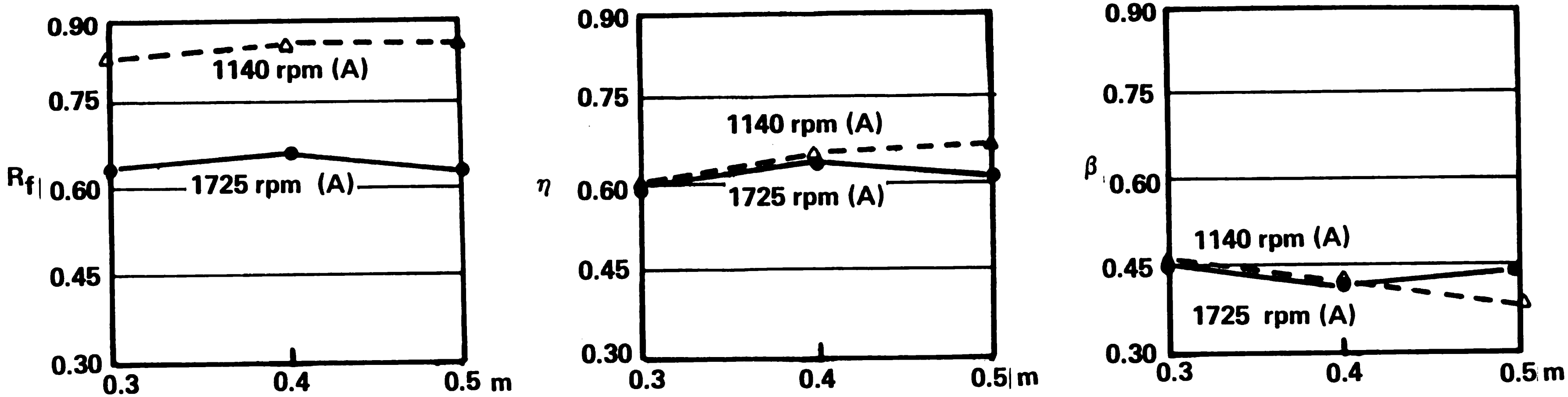


(c) INTERACTION OF CYCLE TIME (min.) WITH EXHAUST FAN SPEED A (rpm)

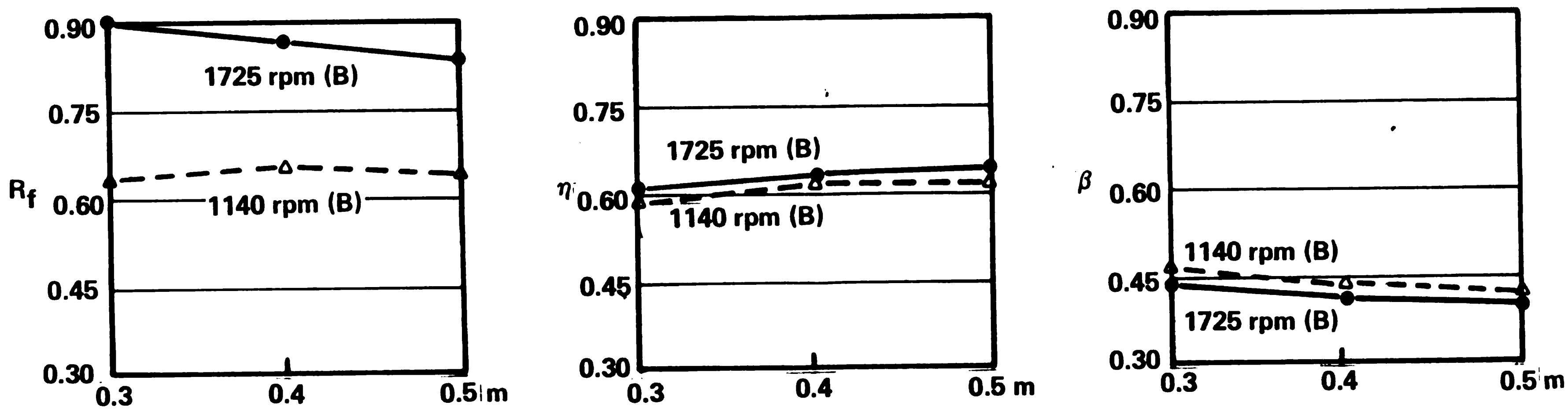


(d) INTERACTION OF CYCLE TIME (min.) WITH MAKE–UP FAN SPEED B (rpm)

FIGURE 5.2 TWO–VARIABLE INTERACTIONS (CONTINUED)



(e) INTERACTION OF ROCKBED DEPTH (m) WITH EXHAUST FAN SPEED A (rpm)



(f) INTERACTION OF ROCKBED DEPTH (m) WITH MAKE-UP FAN SPEED B (rpm)

FIGURE 5.2 TWO-VARIABLE INTERACTIONS (CONTINUED) the make-up fan speed on Rf. The interaction of cycle time with rockbed depth had little effect on η, β and Rf if response curves at 0.3 and 0.51 m were compared with each other.

5.3 Relationship between η and Rf

By definition, the heat recovery effectiveness η had a direct relationship with the heat recovery factor Rf :

$$\eta = (M_e/M_{max})Rf \qquad (5.1)$$

This depended merely upon two mass flow rates, M_e and M_i, as M_{max} was either M_e or M_i whichever was larger. Rf was never less than η, because M_e was always less than or equal to M_{max}. When both fans were at the same speed, either 1140 or 1725 rpm, the make-up mass flow rate M_i was a little larger than the exhaust mass flow rate M_e, so η was not much less than Rf. The resulting values were of 0.65 to 0.78 for η, 0.72 to 0.84 for Rf in the case of 1725 rpm for both fans, and of 0.63 to 0.81 for η, 0.70 to 0.88 for Rf in the case of 1140 rpm for both fans.

When the exhaust fan was at 1725 rpm and the make-up fan was at 1140 rpm, M_e became M_{max}, so η equaled Rf. The low resultant values of 0.36 to 0.61 for both η and Rf were due to insufficient make-up air capable of carrying less heat. Lampman (1978) concluded an exhaust air flow rate of 2 or 2.5 times the make-up flow rate to minimize freezing problems in his investigation. Somehow, it reduced the heat recovery effectiveness. When the exhaust fan was at

1140 rpm and the make-up fan was at 1725 rpm, Rf was from 0.73 to 1.0. This meant that most, even approximately total, heat, given up by the low exhaust flow rate to the rockbed, was reclaimed by the high make-up flow rate. But, in this case η was from 0.43 to 0.63, which was lower than Rf very much. This was because some heat was lost with the leakage air.

5.4 Correlation of η with β

Physically, in this system the more heat was recovered, the less moisture was removed from the room, since more moisture was returned back to the room. In the extreme, if entire mass of the exhaust air was recirculated back to the room, all recoverable heat would be "regenerated" if duct losses were neglected. This case would be undesirable as ventilation must remove moisture, toxic gases and so on from the room. At the other extreme, if the exhaust air was released directly to the atmosphere without passing through the rockbeds, no heat would be recovered, but all removable moisture would be removed. This is the case for conventional ventilation systems. Mathematically, these two circumstances may be expressed as the following, respectively:

$$\beta_0 = 0 \qquad \text{for } \eta_0 = 1 \tag{5.2}$$

$$\beta_1 = 1 \qquad \text{for } \eta_1 = 0 \tag{5.3}$$

A third-order polynomial was expected as the correlation of η with β.

$$\beta = a * \eta^3 + b * \eta^2 + c * \eta + d \qquad (5.4)$$

Transform (5.4) by applying two given conditions in (5.2) and (5.3). A linear model came out.

$$Z = u * \eta + s \qquad (5.5)$$

where

$$Z = (\beta + \eta - 1)/(\eta^2 + \eta)$$

$$u = a$$

$$s = a * \eta_0 + b$$

A single linear regression was conducted. The resultant regressive equation was :

$$Z = -0.51 * \eta + 0.1 \qquad (5.6)$$

with $r = -0.667$ and d.f. = 34

A t-test showed that for 0.1 % significance level with 34 d.f. the absolute value of the correlation coefficient R should be larger than 0.53 (BIAM, 1980). Finally

$$\beta = -0.51 * \eta^3 + 0.61 * \eta^2 - 1.1 * \eta + 1 \qquad (5.7)$$

Fig. 5.3 shows the close relationship between the regression curve with the tested data.

Since the moisture removal factor β was less 1.0, for the renegerative ventilation system a larger exhaust flow rate than that for a conventional ventilation system is required. The performance of both heat recovery and moisture removal should be considered during the design of a regenerative ventilation system. A heat recovery effectiveness about 0.5 would lead to a moisture removal factor about 0.54, which may be desirable for the system.

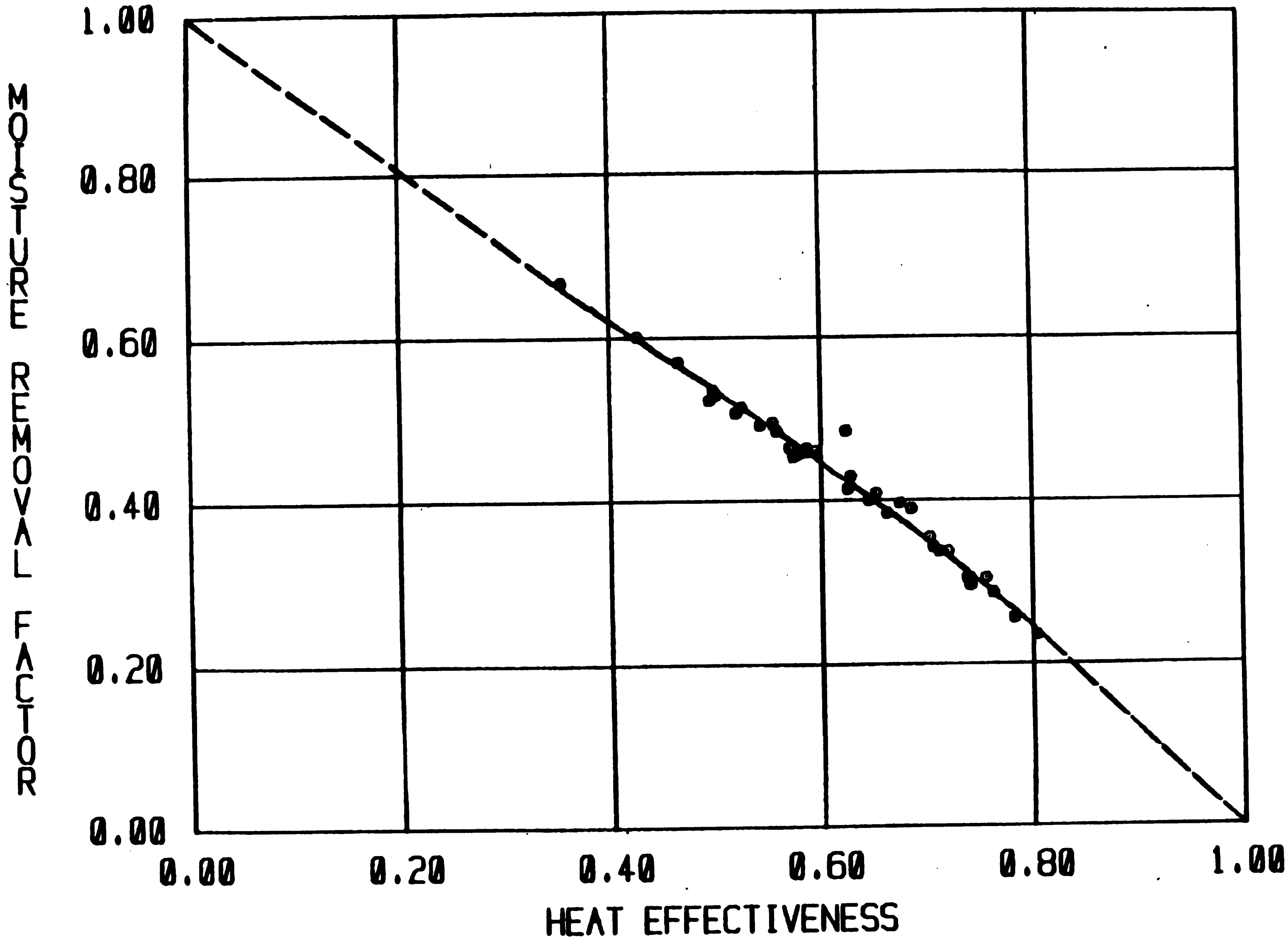


FIGURE 5.3 CORRELATION OF HEAT EFFECTIVENESS AND MOISTURE REMOVAL FACTOR

5.5 Correlation between Enthalpy and Flow Rates

The make-up air enthalpy E_i was a function of the indoor air enthalpy E_e, the outdoor air enthalpy E_o, and the exhaust and make-up air flow rates, Q_e and Q_i, for the given rockbed regenerator structure. A multiregression model was obtained:

$$E_i = 0.96E_e + 0.3E_o + 86.5Q_e - 74.3Q_i - 15.8 \quad (5.8)$$

where Es were in kJ/kg dry air and Qs in m^3/s. This regression model had a coefficient of determination of 0.95.

The mean values for E_e, E_i, E_o, Q_e and Q_i were 60.13, 49.78, 21.9 kJ/kg dry air and 0.136 and 0.146 m^3/s, respectively, in this study. So, E_e was the most significant factor, E_i - the second, and both Q_e and Q_i - of less importance in the ranges tested. A simplified regression model was as follows :

$$E_i = 1.382E_e + 0.143E_o - 30.8 \quad (5.9)$$

with a coefficient of determination of 0.71 and d.f. of 33. A F-test indicated this regression model had 95.5 % confidence of significance.

5.6 Correlation between Humidity and Flow rates

Likewise, a multiregression equation shows the incoming air humidity ratio h_i had a functional relationship with the exhaust and outside air humidity ratioes, h_e and h_o, and flow rates of Q_e and Q_i.

$$h_i = .83h_e + .34h_o + .011Q_e - .01Q_i - .0013 \quad (5.10)$$

where hs were in kg water/kg dry air and Qs - m^3/s.

This model had R^2 of 0.98 and d.f. of 33. Mean values of h_e, h_i and h_o were 0.0079, 0.0062 and 0.0027 kg water/kg dry air, respectively. The first and the second important factors affecting h_i were h_e and h_o, respectively. A simplified regression equation is shown below.

$$h_i = 0.91h_e + 0.26h_o - 0.0017 \qquad (5.11)$$

with $R^2 = 0.90$ and d.f. = 33.

5.7 Air Temperature Profiles

5.7.1 Temperature Profiles within the Rockbed

Fig. 5.4 (a), (b), (c), (d) and (e) present 5 typical air temperature profiles at each 0.1 m depth level within the sample rockbed.

Fig. 5.4 (a) and (b) show the influence of the cycle time on the temperature profiles. In these two trials the 0.51 m depth and 1725 rpm of the two fan speeds were the same, the cycle time was different, 4 and 12 min, respectively, besides the exhaust and the outside temperatures. It is seen that

1. The rockbed was charged and discharged more thoroughly for the 12 min cycle time than 4 min. For example, consider the temperature profiles at the rockbed bottom in FIg. 5.4 (a) and (b). Though the outside temperature was about 6.0 °C lower in the 12 min trial than

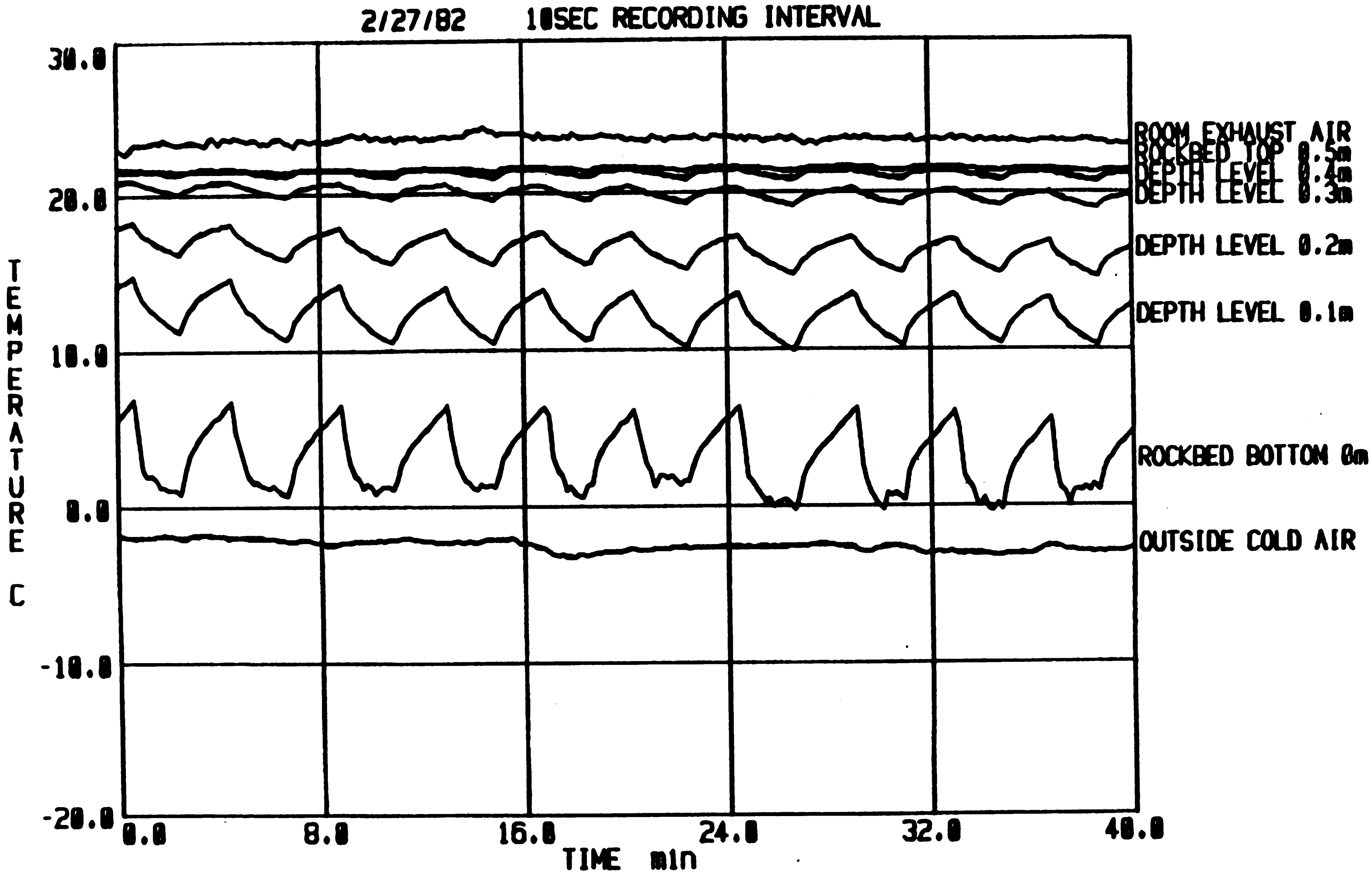


FIGURE 5.4 TEMPERATURE PROFILES WITHIN THE ROCKBED LAYERS

(a) 4min CYCLE, 0.51m DEPTH, .159m^3/s EXHAUST AND .165m^3/s MAKE-UP FLOW RATES

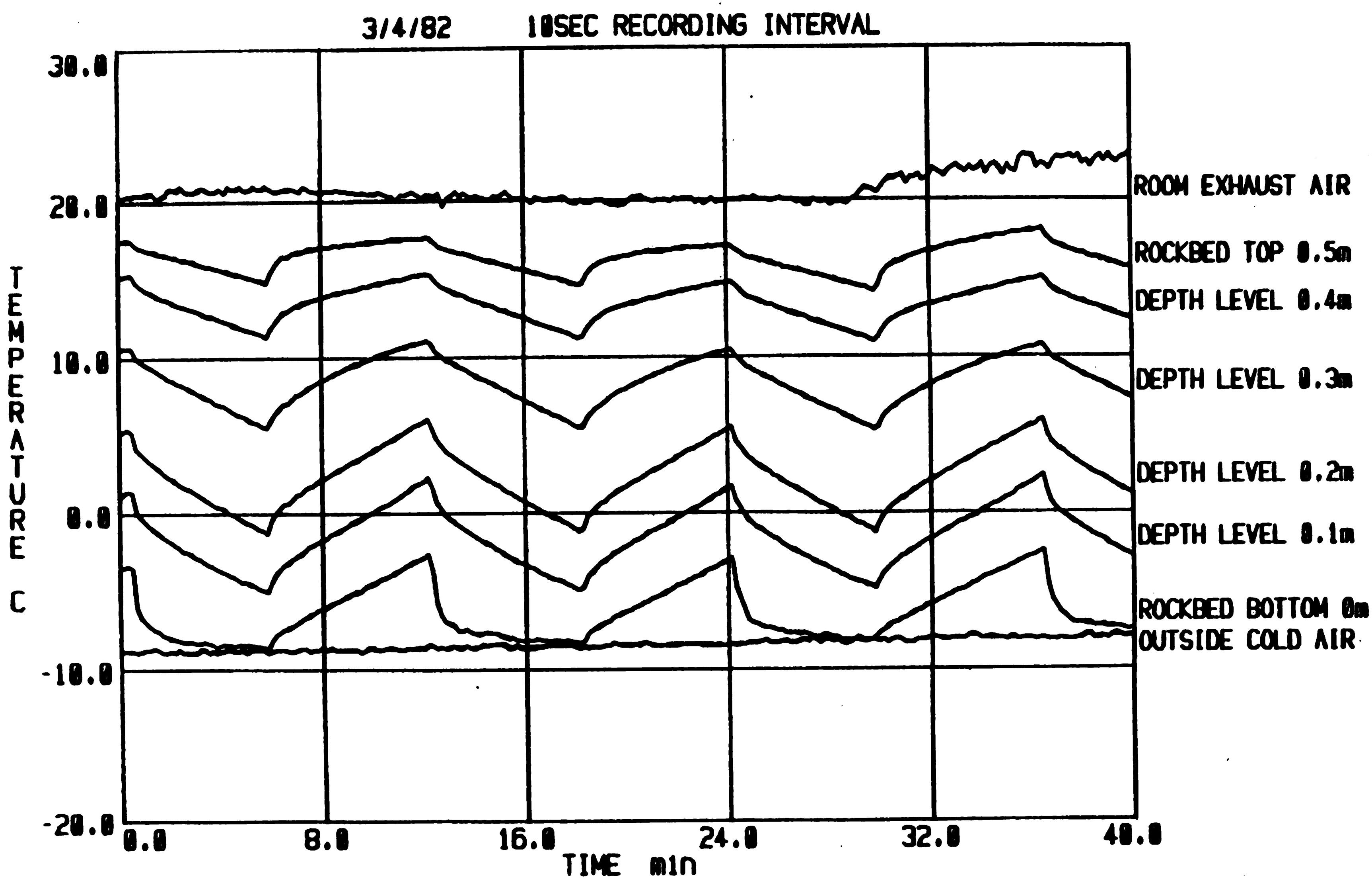


FIGURE 5.4 TEMPERATURE PROFILES WITHIN THE ROCKBED LAYERS (CONTINUED)

(b) 12min CYCLE, 0.51m DEPTH, .159m^3/s EXHAUST AND .165m^3/s MAKE-UP FLOW RATES

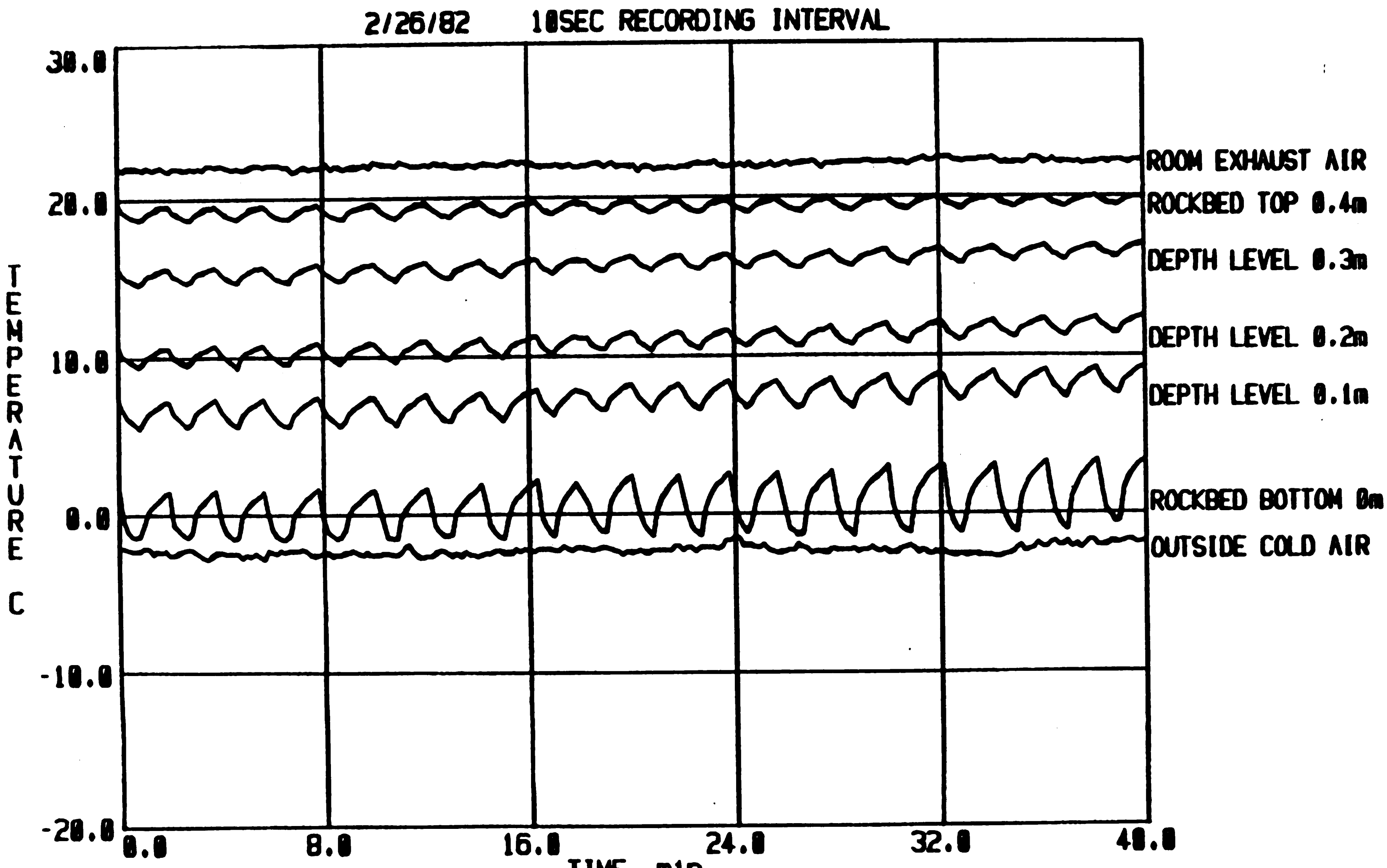


FIGURE 5.4 TEMPERATURE PROFILES WITHIN THE ROCKBED LAYERS (CONTINUED)
(c) 2min CYCLE, 0.41m DEPTH, .110m^3/s EXHAUST AND .118m^3/s MAKE-UP FLOW RATES

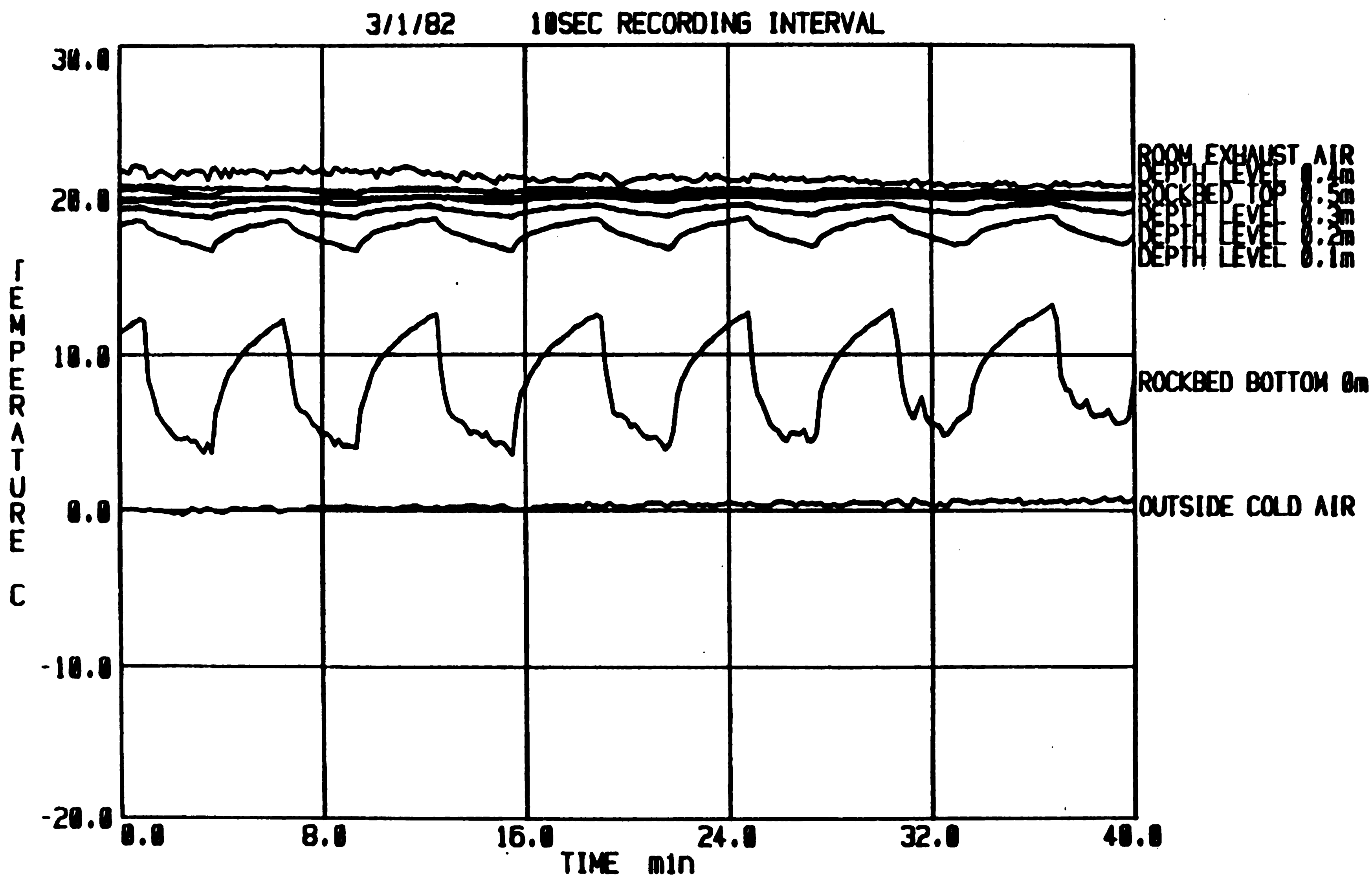


FIGURE 5.4 TEMPERATURE PROFILES WITHIN THE ROCKBED LAYERS (CONTINUED)

(d) 6min CYCLE, 0.51m DEPTH, .159m^3/s EXHAUST AND .113m^3/s MAKE-UP FLOW RATES

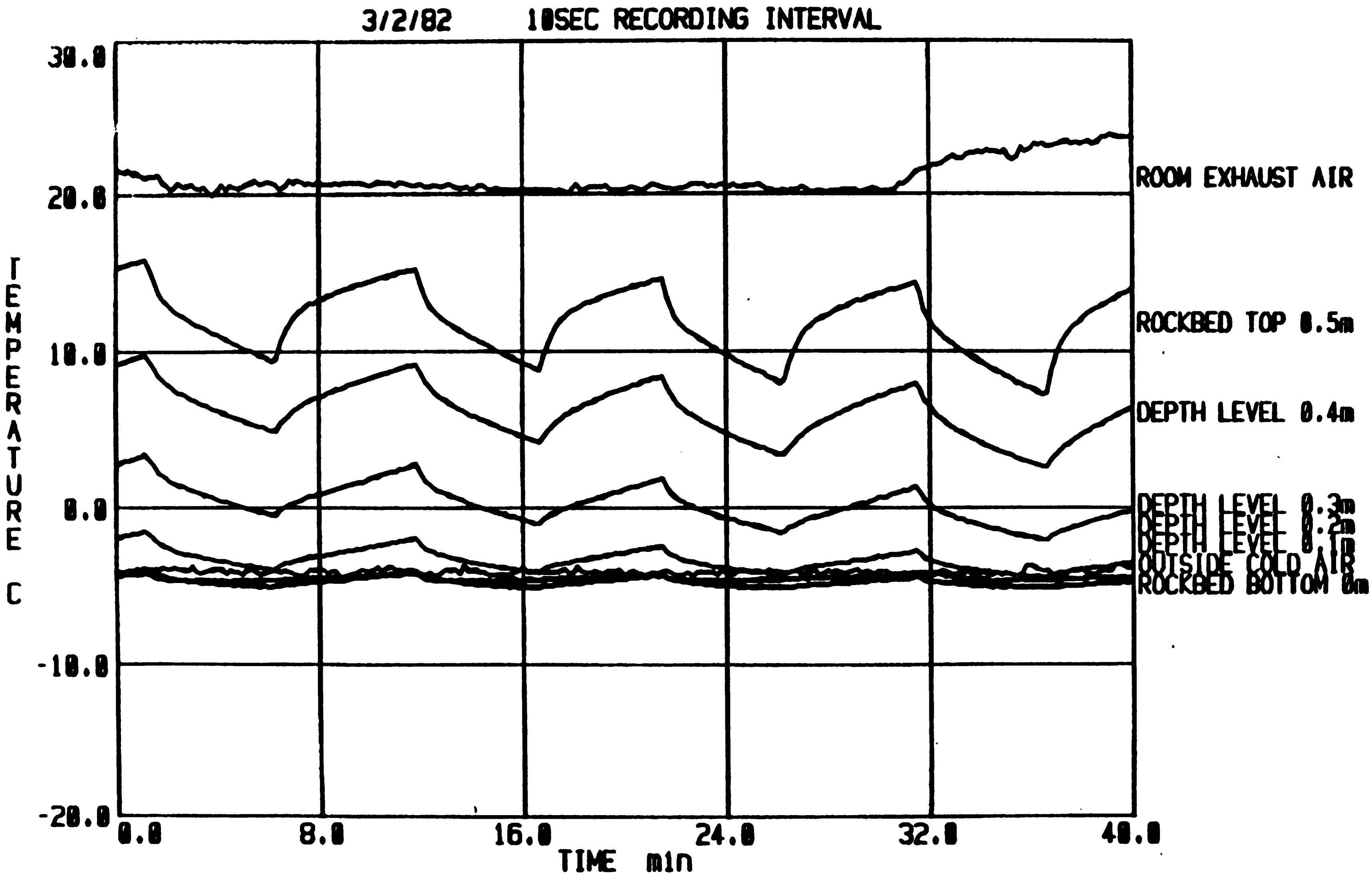


FIGURE 5.4 TEMPERATURE PROFILES WITHIN THE ROCKBED LAYERS (CONTINUED)

(e) 10min CYCLE, 0.51m DEPTH, .108m^3/s EXHAUST AND .165m^3/s MAKE-UP FLOW RATES that in the 4 min trial, the temperature increases during the warming half cycle were approximately equal in the two trials. During the cooling half cycles, the temperature dropped closer to the outside temperature for the 12 min cycle time. This was due to the larger heat capacity, i.e. thermal inertia, of the rockbed.

2. When the outside cold air or the room exhaust warm air was first introduced, there was an abrupt drop or increase in temperature within the rockbed. This abrupt drop was obviously observed at the rockbed bottom and at the 12 min cycle time.

3. The upward and downward segments, corresponding to warming and cooling half cycles, of these profiles were closer to straight lines as the cycle time was above 10 min. This indicated that the rockbed thermal response to the temperature change was not constant with respect to the cycle time. Similar results were seen in other trials regardless of the two fan speeds.

4. The temperature gradient throughout the rockbed depth was more uniform for the longer cycle time. This was because during the longer cycles heat was distributed more uniformly. Similar phenomena were observed for other trials when the two fans were at the same speed, either 1140 or 1725 rpm. Fig. 5.4 (c) shows an example for the two fans at 1140 rpm speed. On the other hand, when the exhaust fan was at 1725 rpm and the make-up fan - 1140 rpm (Fig. 5.4 (d)), the temperature gradient differed very much along the

vertical direction. Since the warm exhaust flow rate was higher than the cool make-up flow rate, the temperatures throughout the rockbed were high. Especially, the mean temperatures at the upper several depth levels were very close to each other, and near to the exhaust air temperature. When the exhaust fan was at 1140 rpm and the make-up fan was at 1725 rpm, the profiles were just opposite to the case above. In the lower depth levels the temperatures were close to the outside cold temperature, and the temperature at the rockbed top was much lower than the exhaust air temperature (Fig. 5.4 (e)). This was caused by the larger make-up flow carrying most stored heat from the rockbed. In Fig. 5.4 (e) the mean temperature at the rockbed bottom was little "lower" than the "outside" temperature was due to wind chilling and position difference between two points monitored.

5.7.2 Average Temperature from the Rockbed Bottom Upwards

Fig. 5.5 presentes 4 graphs, which illustrates the average temperature varied at various depth levels. Fig. 5.5 (a) shows that a positive temperature gradient decreased from the rockbed bottom upwards for both fans at 1725 rpm. Similar results were seen for both fans at 1140 rpm, which is not repeatedly shown here. Fig. 5.5 (b) shows an example of the 1725 rpm exhaust fan speed and the 1140 rpm make-up fan speed. Above 0.1 m the temperature increased very little. From 0.3 m upwards, there was almost

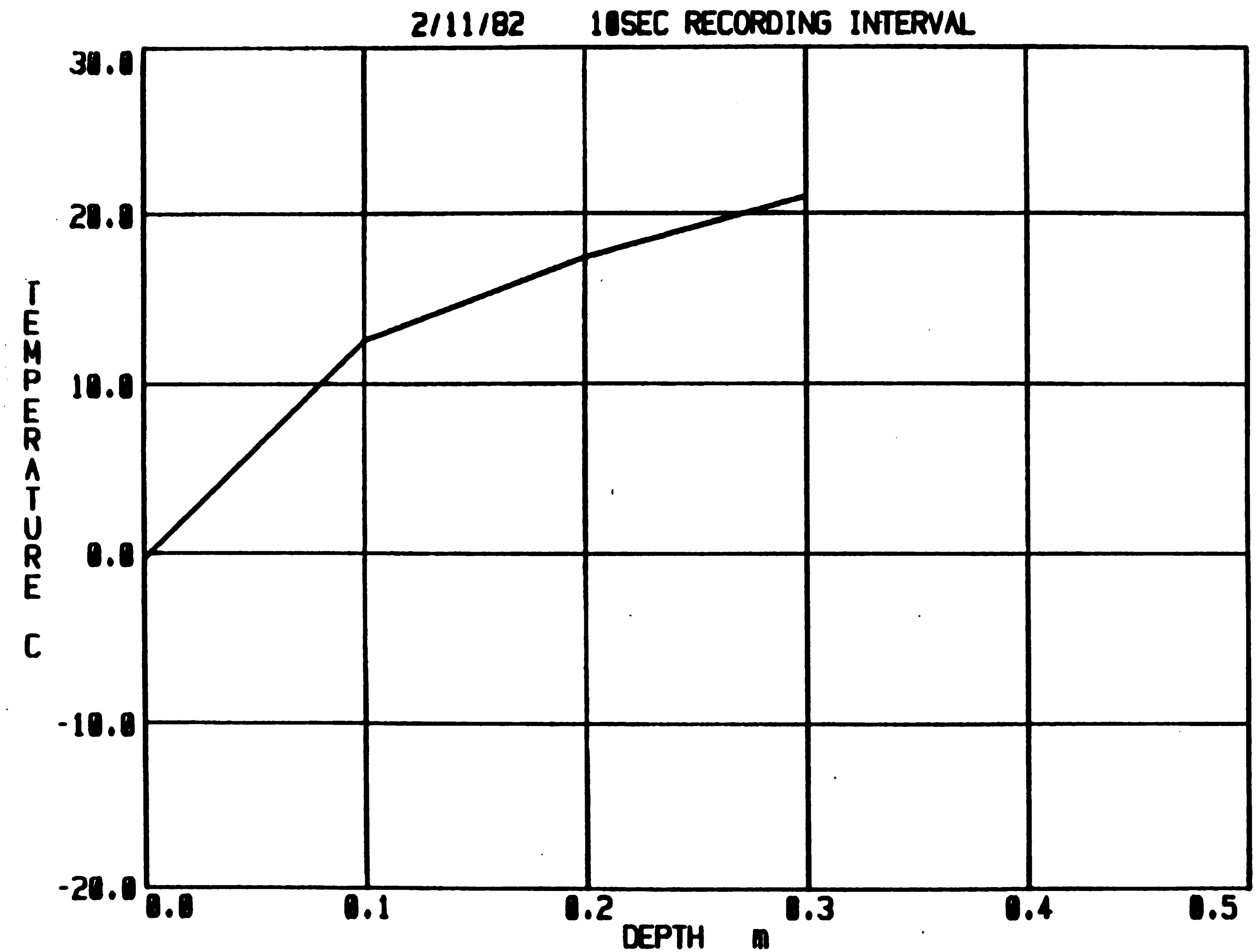


FIGURE 5.5 AVERAGE TEMPERATURE FROM ROCKBED BOTTOM UPWARDS

(a) 4min CYCLE, 0.30m DEPTH, .167m^3/s EXHAUST AND .184m^3/s MAKE-UP FLOW RATES

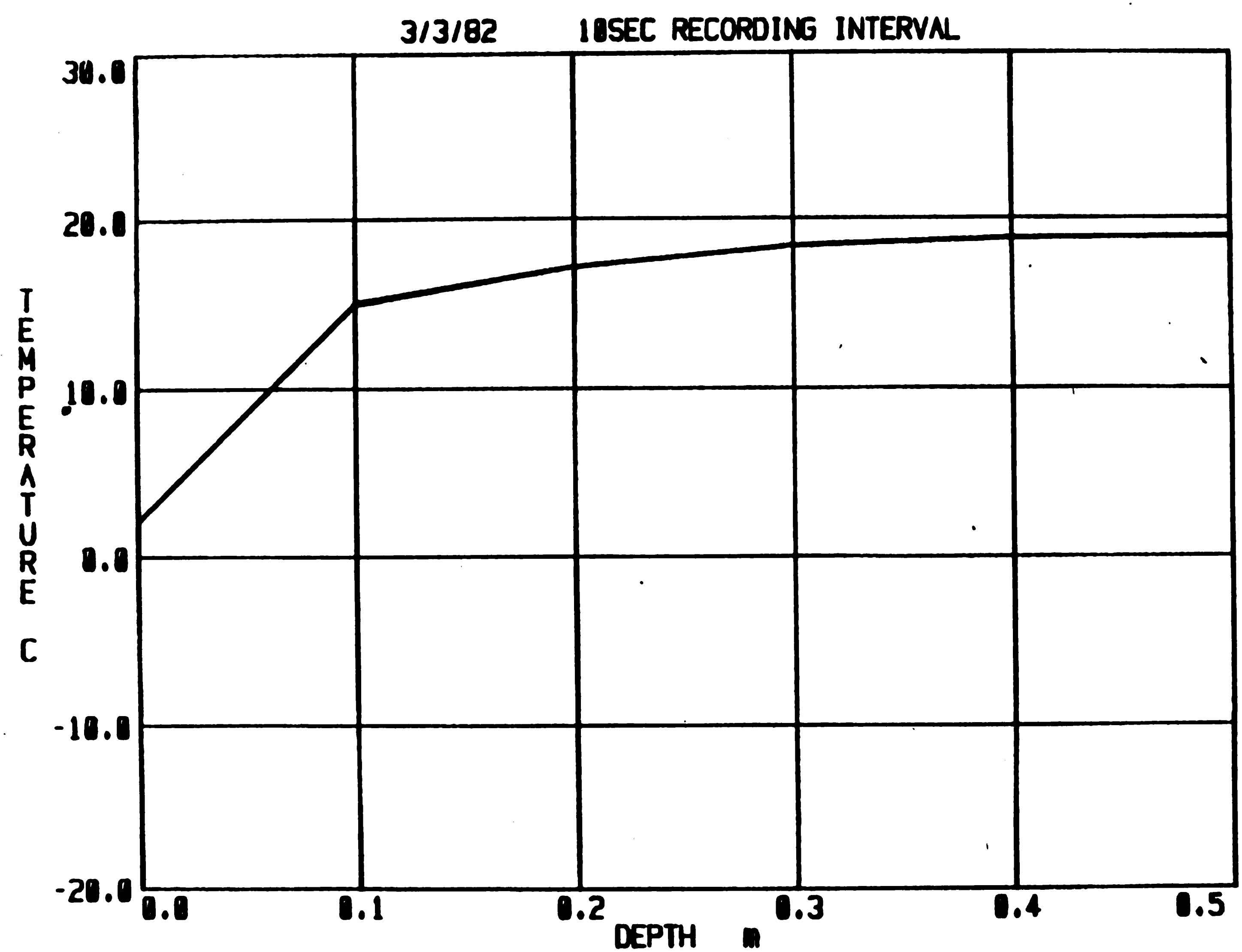


FIGURE 5.5 AVERAGE TEMPERATURE FROM ROCKBED BOTTOM UPWARDS (CONTINUED)
(b) 10min CYCLE, 0.51m DEPTH, .159m^3/s EXHAUST AND .113m^3/s MAKE-UP FLOW RATES

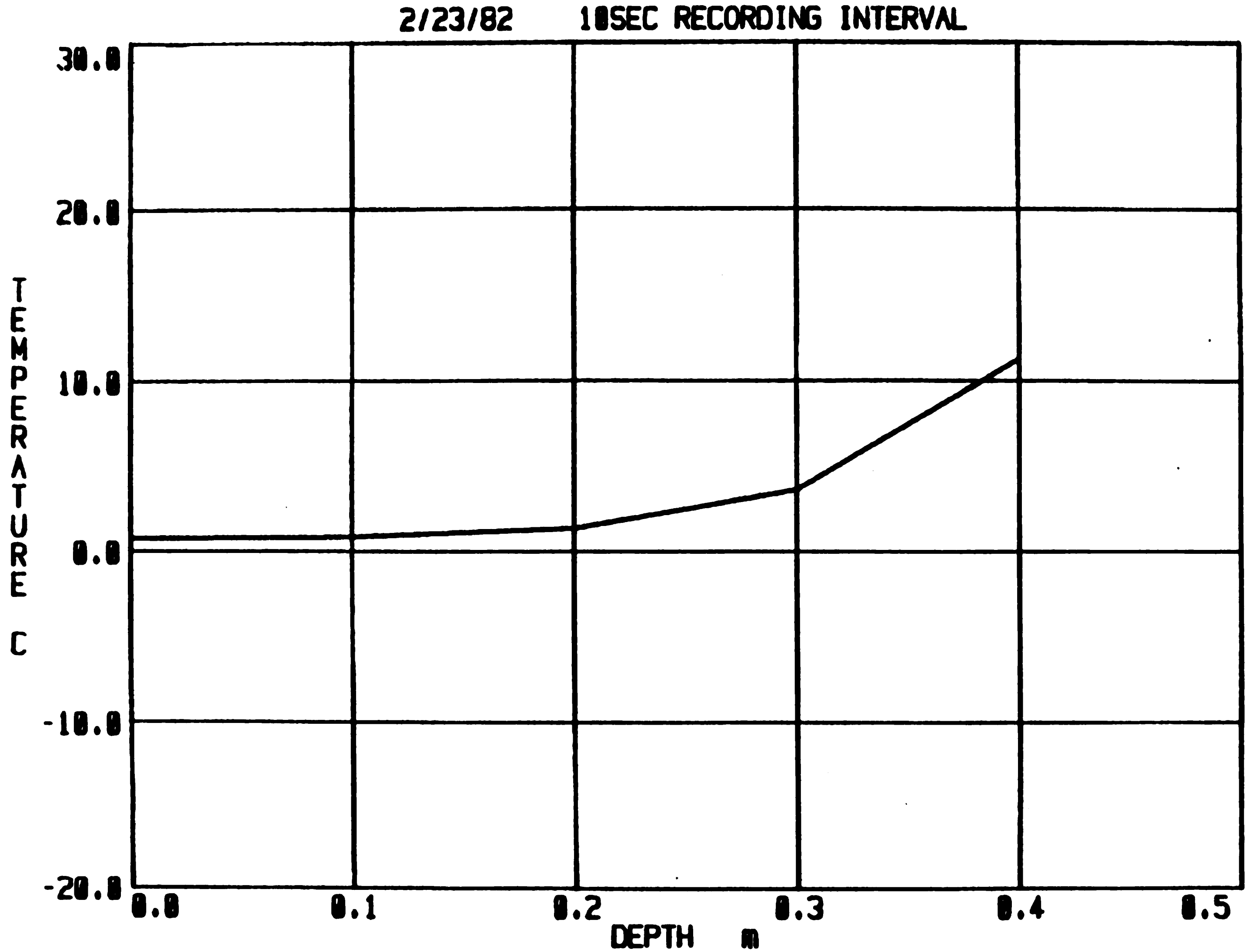


FIGURE 5.5 AVERAGE TEMPERATURE FROM ROCKBED BOTTOM UPWARDS (CONTINUED)

(c) 8min CYCLE, 0.41m DEPTH, .110m^3/s EXHAUST AND .172m^3/s MAKE-UP FLOW RATES

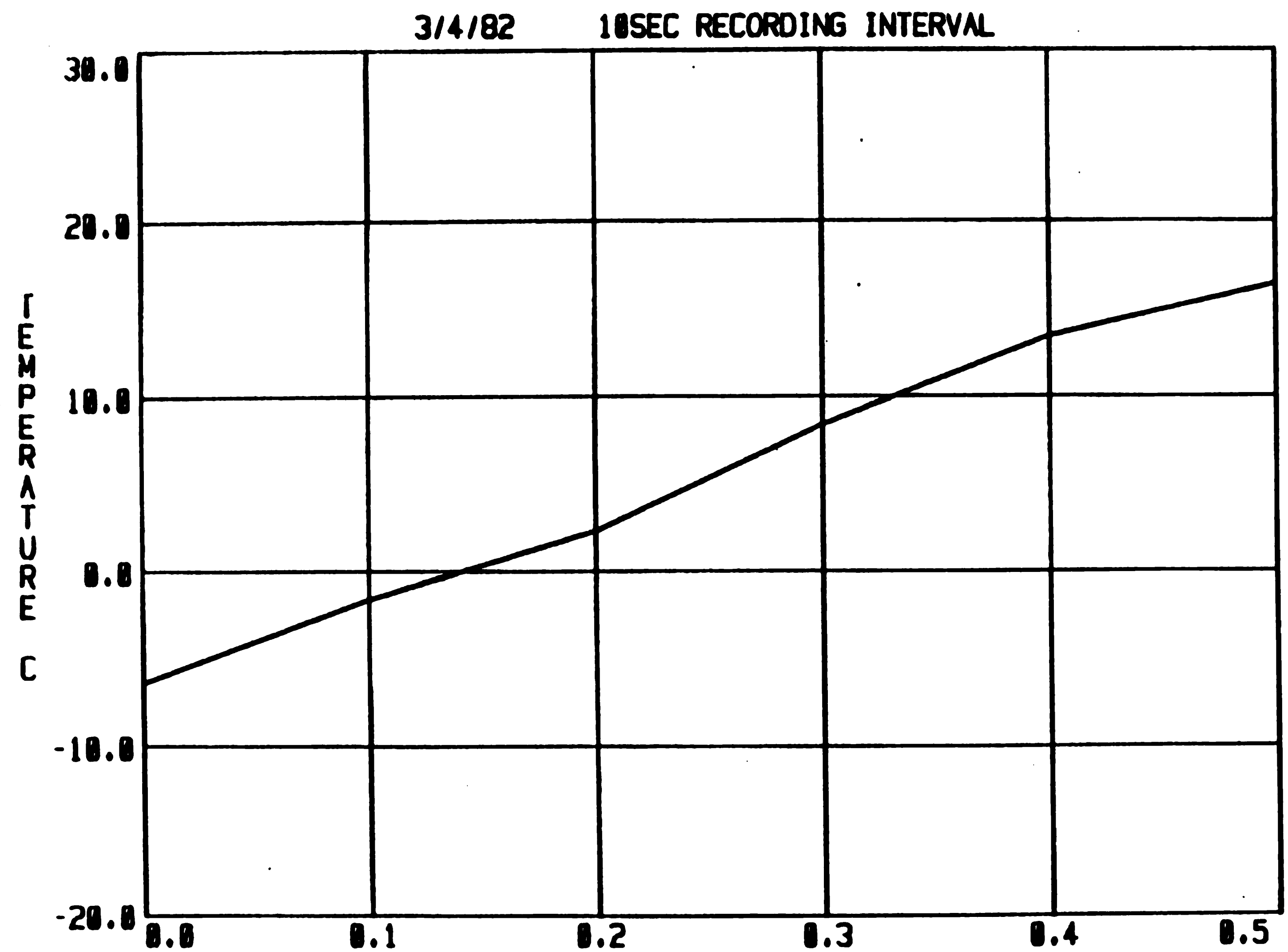


FIGURE 5.5 AVERAGE TEMPERATURE FROM ROCKBED BOTTOM UPWARDS (CONTINUED)
(d) 12min CYCLE, 0.51m DEPTH, .159m^3/s EXHAUST AND .165m^3/s MAKE-UP FLOW RATES

no temperature change. Fig. 5.5 (c) presents an example for the 1140 rpm exhaust fan speed and the 1725 rpm make-up fan speed. In the lower 0.1 m thickness the average temperature was nearly a constant. Then, the temperature gradient increased at the high depth levels. Fig. 5.5 (d) illustrated an uniform temperature gradient throughout the rockbed depth for the 12 min cycle time and the 1725 rpm speed of both fans.

5.7.3 Temperature Profiles in the Ventilation System

Fig. 5.6 shows temperature profiles measured at the entrance and the exit of the sample rockbed, at the room inlet and outlet, and at the rockbed bottom and the outside. The upward segments on the temperature curve for the make-up air correspond to the discharge half cycles in the sample chamber. The downward segments on the curve represent the discharge half cycles in the other chamber. An interesting phenomenon was observed that the make-up air temperature from the sample chamber was higher than that from the other. These two chambers had the same size and insulation, and were packed with the same rocks in the same way. A noticeable difference between them was with or without the expanded metal sheets. These metal sheets divided the sample rockbed into several blocks. They disturbed air flow paths within the sample chamber and air was redistributed while the air passed across each sheet. This might improve the heat transfer process within the rockbed. Moreover, the

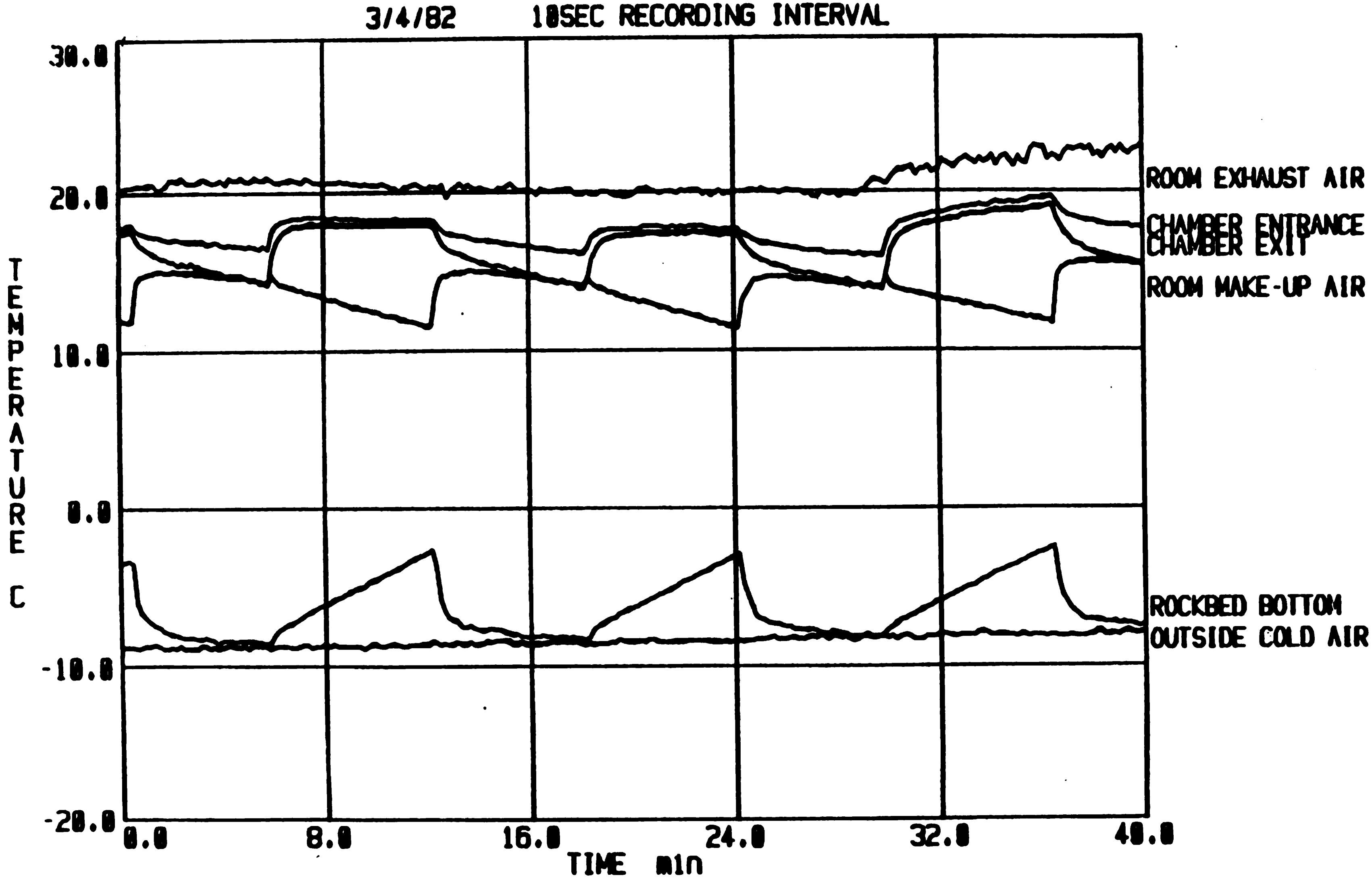


FIGURE 5.6 TEMPERATURE PROFILES IN THE VENTILATION SYSTEM

(a) 12min CYCLE, 0.51m DEPTH, .159m^3/s EXHAUST AND .165m^3/s MAKE-UP FLOW RATES

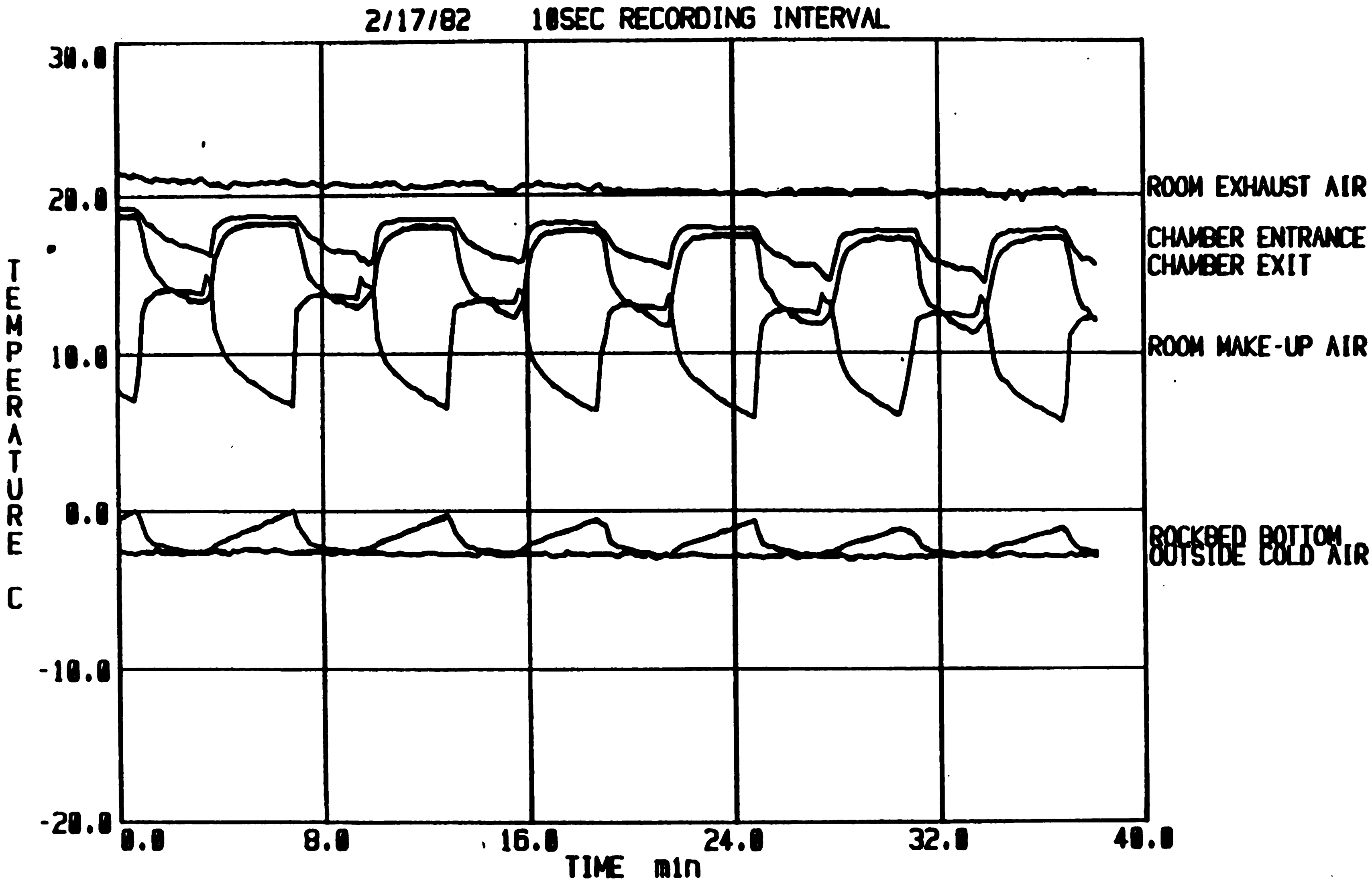


FIGURE 5.6 TEMPERATURE PROFILES IN THE VENTILATION SYSTEM (CONTINUED)

(b) 6min CYCLE, 0.30m DEPTH, .111m^3/s EXHAUST AND .184m^3/s MAKE-UP FLOW RATES

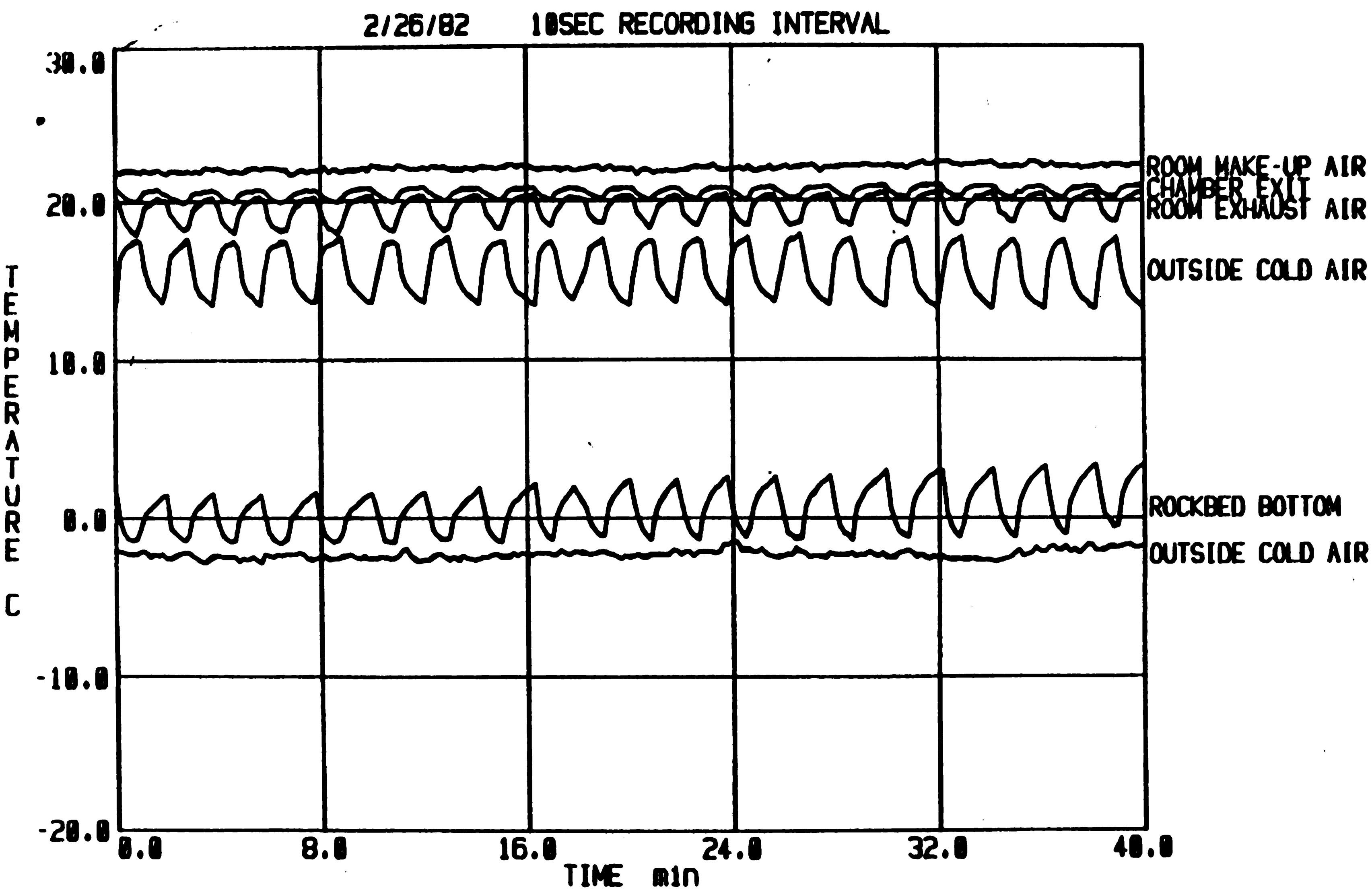


FIGURE 5.6 TEMPERATURE PROFILES IN THE VENTILATION SYSTEM (CONTINUED)

(c) 2min CYCLE, 0.41m DEPTH, .110m^3/s EXHAUST AND .118m^3/s MAKE-UP FLOW RATES

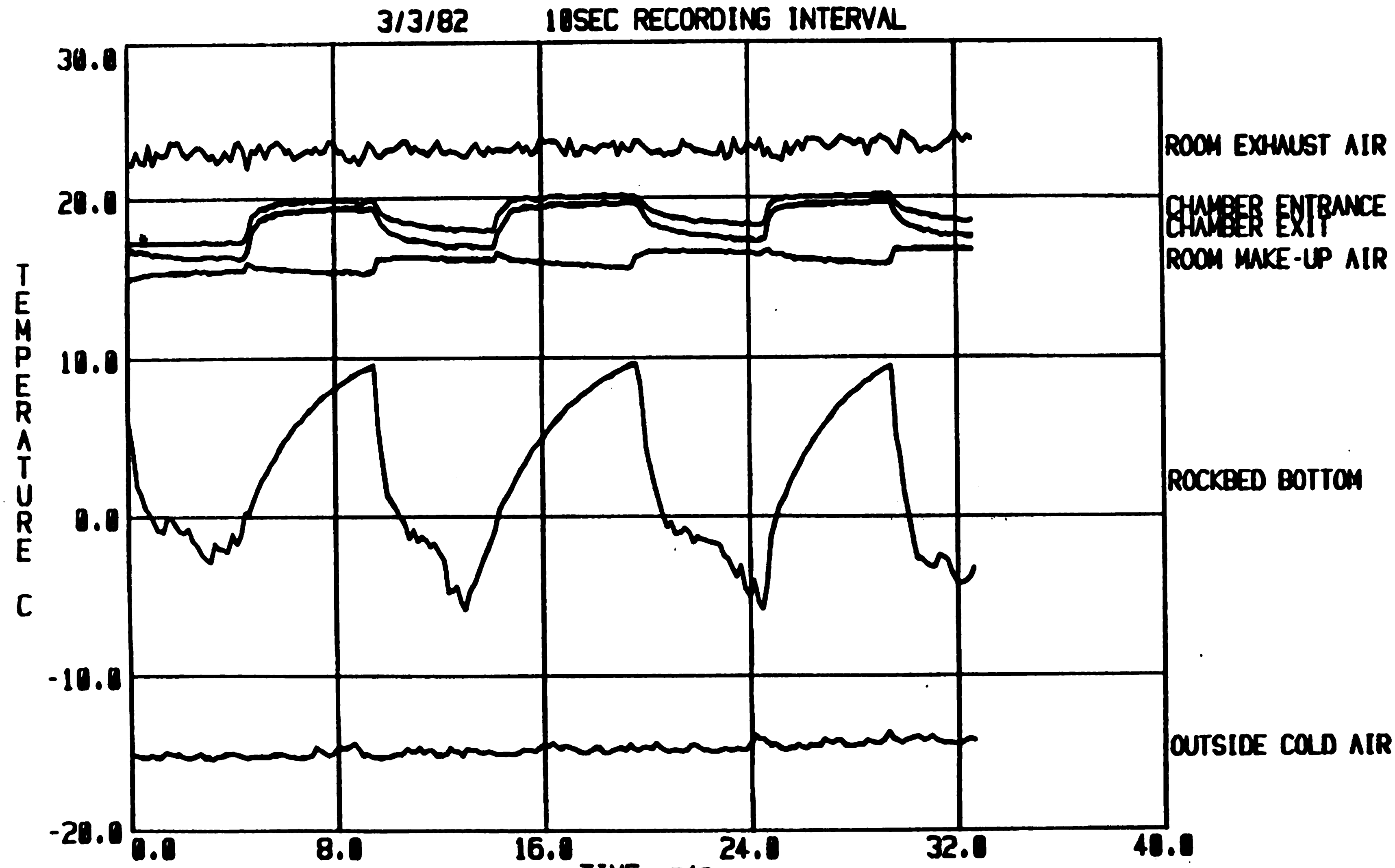


FIGURE 5.6 TEMPERATURE PROFILES IN THE VENTILATION SYSTEM (CONTINUED)

(d) 10min CYCLE, 0.51m DEPTH, .159m^3/s EXHAUST AND .113m^3/s MAKE-UP FLOW RATES

sheets were good conductors, which made temperature more even on the plains themselves. The phenomenon of different temperature gains in make-up air flow became severe when the exhaust fan was at 1140 rpm and the make-up fan - 1725 rpm, see Fig. 5.6 (b). This was due to the high make-up flow could easilier recover most heat in the sample rockbed with screen sheets. When the cycle time was short (Fig. 5.6 (c)), the heat was not transferred thoroughly due to the thermal inertia of the rockbed. In this case, the influence of the metal sheets became significant, which also led to high temperature gains in the make-up air from the sample rockbed. On the other hand, when the exhaust fan was at 1725 rpm and the make-up fan - 1140 rpm, associated with a long cycle time, this phenomenon became slightly (Fig. 5.6 (d)). This was because the high exhaust flow rate provided much heat to the rockbed during long cycles.

Some small peaks on the temperature curves of the make-up air occurred at each turning point of the discharge half cycle to the charge cycle. This was due to a time delay between closing and opening dampers. As claimed by the manufacturer, it took 40 to 56 sec or 64 to 80 sec for the control timer to open or close a circuit, respectively. As long as the entrance damper was opening, the warm air flowed into the chamber immediately. However, the exit damper was still closing at the same time. Thus, an amount of warm air flowed through the exit directly and mixed with the make-up air. This resulted in a peak temperature of the make-up air until the exit damper closed tightly.

The temperature drop between the chamber exit and the room inlet indicated some heat loss in the duct. So did the temperature drop between the room outlet and the chamber entrance. In Fig. 5.6 (b) it was observed that the make-up air temperature was, even, higher than that at the chamber exit during the discharge half cycle. This was caused by the warm room air, which influenced on the incoming air at the room inlet. When the exhaust fan was at 1140 rpm and the make-up fan - 1725 rpm, the air temperature at the chamber entrance dropped very much during the cooling half cycle, so this phenomenon was recognized.

5.7.4 Overnight Performance

Fig. 5.7 shows the overnight performance of the rockbed regenerator system during 3 test runs. The shape of these proflies depend on the combination of the cycle time and the recording interval used. Therefore, they do not reflect the actual performance in cycles.

It took about 2 hours for the system to approach a steady state after adjusting some parameters, such as cycle time, fan speed and so on, see Fig. 5.7 (a). The temperature fluctuation of the exhaust air indicated that the room gas heater was turned on once about each hour, then, off (Fig. 5.7 (a) and (b)). It seems that the thermostat was not sensible enough to maintain a stable room temperature. Fig. 5.7 (b) illustrats an overnight

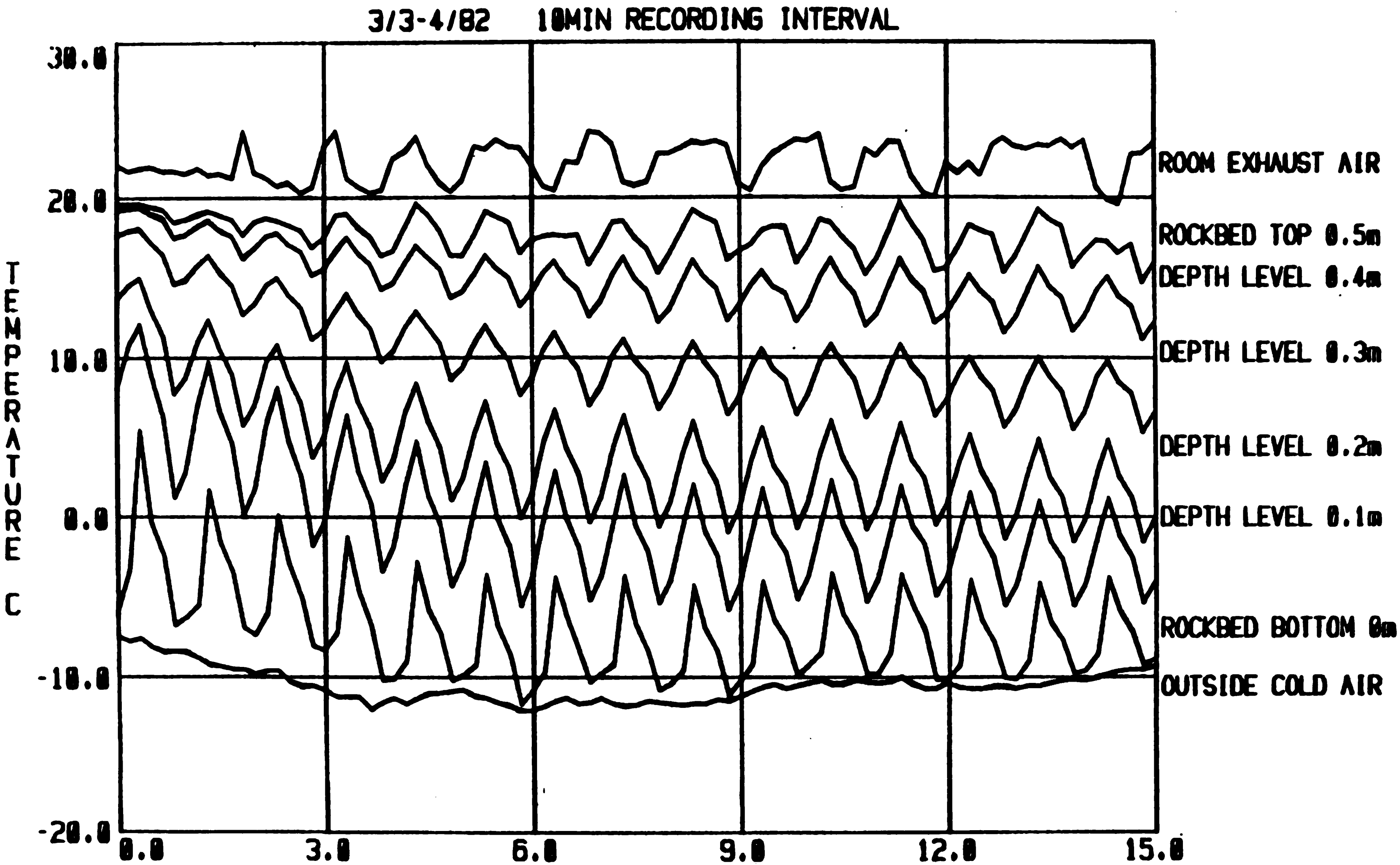


FIGURE 5.7 OVERNIGHT PERFORMANCE OF THE ROCKBED SYSTEM

(a) 12min CYCLE, 0.51m DEPTH, .159m^3/s EXHAUST AND .165m^3/s MAKE-UP FLOW RATES

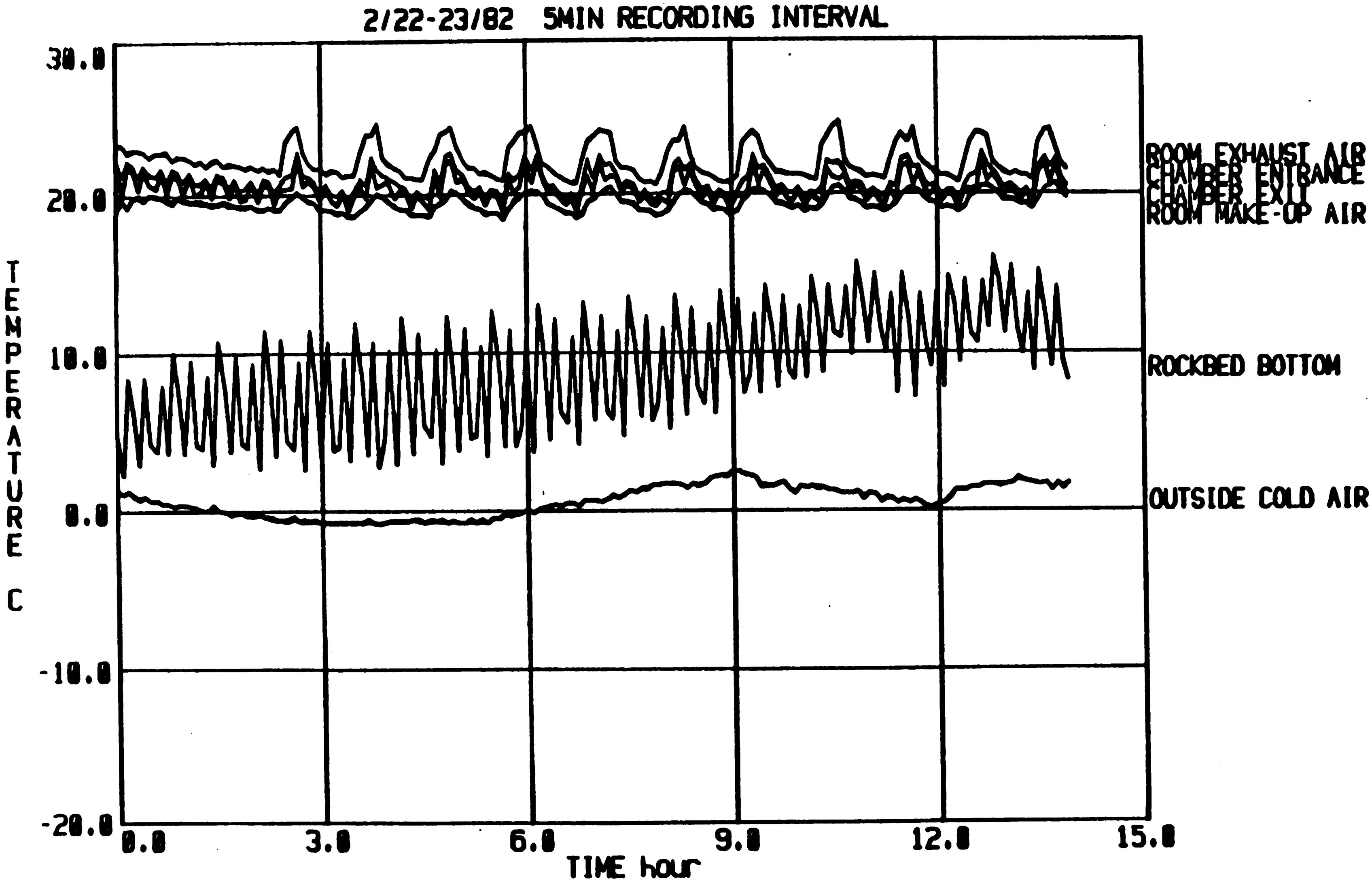


FIGURE 5.7 OVERNIGHT PERFORMANCE OF THE ROCKBED SYSTEM (CONTINUED)

(b) 8min CYCLE, 0.41m DEPTH, .162m^3/s EXHAUST AND .118m^3/s MAKE-UP FLOW RATES

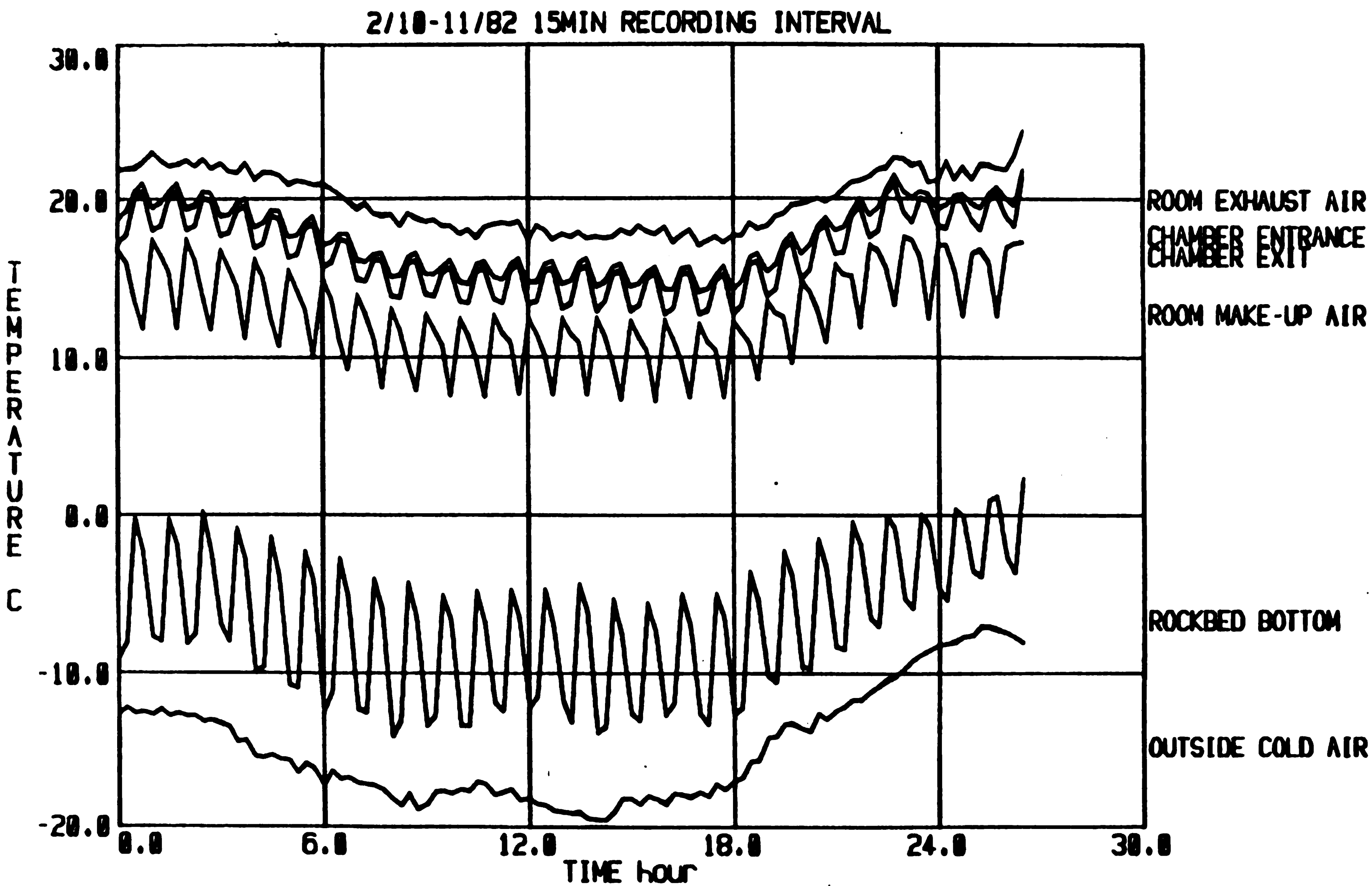


FIGURE 5.7 OVERNIGHT PERFORMANCE OF THE ROCKBED SYSTEM (CONTINUED)
(c) 4min CYCLE, 0.30m DEPTH, .167m^3/s EXHAUST AND .184m^3/s MAKE-UP FLOW RATES

performance of the ventilation system. This also shows the poor performance of the thermostat. Fig. 5.7 (c) shows the ventilation system performance over the coldest night during the test period. It was seen that the room temperature dropped when the outside was the lowest as the gas heater did not work. The rockbed, however, maintained the room temperature at a reasonable level between 17 to 23°C.

5.8 Other Results

5.8.1 Frost

During the whole experiment period, frosting was observed only once. Several small frost spots were found on the bottom frame of the rockbed in the coldest morning of February 11th, when outside temperature was -17°C, see Fig. 5.8. The frost disappeared soon as outside temperature increased in the daytime. Relatively low humidity in the farrowing room associated with moderate climate temperature eliminated most frosting and freezing.

5.8.2 Dust Accumulation

Dust accumulation was observed on the surfaces of the rockbeds, dampers and ducts in the ventilation system. The dust accumulated heavier in the exhaust duct (Fig. 5.9), than in the make-up duct (Fig. 3.6), since much dust settled on various surfaces while the exhaust air flowed through bending corners and dampers in the exhaust duct.

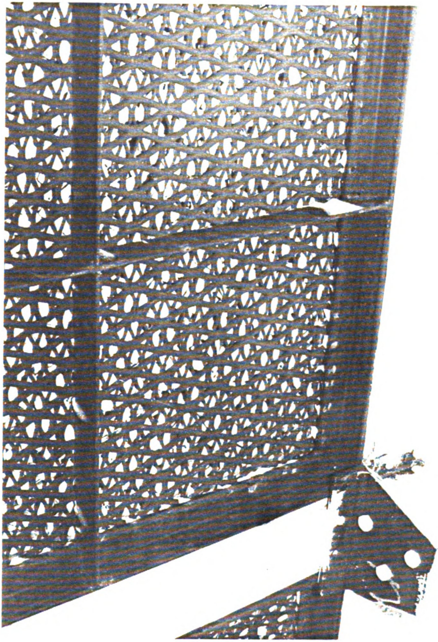


FIGURE 5.8 FROSTING ON THE FRAME

FIGURE 5.9 DUST ACCUMULATION ON SURFACES

Also, dust components were filtered out of the air when it moved through the rockbed as the air velocity was very low in the rockbed chambers. The dust settled on the rockbed top and the make-up air could not lift the dust. The system was not cleaned for the whole test period. No significant effect of the dust problem on the static pressure drop or on other system performance was observed.

Low room air humidity along with dry feed supply caused undesirable dust. Since the size of dust, mostly feed powder, was so small, using fine screen was not a practical way to filter it out.

6 BALANCE EQUATIONS AND DESIGN CRITERIA

Regenerative ventilation systems, in which a part of heat and, perhaps, moisture is returned back to animal buildings, differ from conventional ventilation systems. Mass, moisture and energy balances in the building subsystem should, thus, be studied. New balance equations and design criteria for ventilation systems with regenerators are proposed.

6.1 Air Mass Balance in Animal Housing

An air mass balance must be maintained in the building subsystem. The incoming air mass should equal the outgoing air mass per unit of time. When these two mass rates are not the same, some air, i.e. the difference between the two mass rates, either gets into or escapes out through some leakage or gaps of the room. In the former case, i.e. the room exhaust mass flow rate M_e is greater than the room make-up mass flow rate M_i, cold outside air leaks into the room due to a negative air pressure in the room. More supplemental heat is needed to warm the cold air. In the latter case, i.e. $M_e < M_i$, some indoor air leaks out of the room caused by a positive indoor air pressure. Some heat is lost since it does not pass through the rockbed. The mass balance between the outgoing and the incoming air in an animal building is written as:

$$M_i = M_e + M_l \qquad (6.1)$$

where M_l is the mean leakage mass rate, kg dry air/s. A positive M_l value means leaking out whereas a negative M_l indicates leaking in. When air is leaking out, it is the same as the exhaust air. If some air leaks in, it is the same as the outside air.

In conventional ventilation systems, there is no make-up fan. So $M_i = 0$ and $M_l = -M_e$, which means that the "leaking-in" air mass through inlets or gaps is automatically balanced with the exhaust air mass. Hence, the expression (6.1) does not lose its generality.

6.2 Moisture Balance in Animal Housing

A primary principle of ventilation is to maintain a desirable humidity level for the animal environment (MWPS-1, 1980). The moisture added to an enclosure by animals and the incoming air must equal the moisture removed. Then, the indoor humidity remains stable at a certain level of h_e. Otherwise, h_e and others would change until reaching a new moisture balance. The moisture balance in an aniamal building is of the form:

$$M_e h_e + M_l h_l = M_i h_i + W_a \qquad (6.2)$$

where h denotes the air humidity ratio at the given state and W_a is the total vapor production rate in the room due to animals. Rewrite (6.2)

$$W_a = M_e h_e + M_l h_l - M_i h_i \qquad (6.3)$$

The right side is the net moisture removal rate from the building. It equals the maximum removable moisture rate minus the real moisture return rate into the building, so

$$W_a = M_{max}(h_e - h_o) - M_i(h_i - h_o) \qquad (6.4)$$

It can be written with the moisture removal factor β

$$W_a = M_{max}(h_e - h_o)\beta \qquad (6.5)$$

Therefore, β can be computed as

$$\beta = W_a/[M_{max}(h_e - h_o)] \qquad (6.6)$$

Since β is never larger than 1, M_{max} should be larger than $W_a/(h_e - h_o)$.

In conventional ventilation systems with no heat exchanger, β becomes unity and M_{max} equals M_e. Therefore,

$$W_a = M_e(h_e - h_o) \qquad (6.7)$$

In the case with the heat exchanger, the make-up air state and, consequently, β, are dependent not only on the indoor and outdoor conditions, the animals and the ventilation rates, Q_e and Q_i, but also on the heat exchanger. Consider a special case that M_e is assumed to be the same as M_i. Thus

$$\left\{ \begin{array}{l} \beta = (h_e - h_i)/(h_e - h_o) \\ W_a = M_e(h_e - h_i) \end{array} \right. \qquad (6.8)$$

6.3 Energy Balance in Animal Housing

Another principle function of ventilation is to control the room temperature within a suitable range. The heat supplied to an animal environment must equal the heat removed from the room. Otherwise, the room temperature and/or others would adjust themselves until a new energy balance is reached. ASAE D270.4 (1979) states that a portion of sensible heat produced by animals evaporates some water from various wetted surfaces in the room. Therefore, enthalpy should be used in the energy balance equation instead of sensible heat. The indoor temperature is implicitly represented in the exhaust air enthalpy. The energy balance in animal housing with heat recovery is written as

$$q_a + q_i + q_{sp} = q_e + q_l + q_b \qquad (6.9)$$

where q_a - total animal heat production rate, kW,

q_i - energy rate in the incomimg air, kW,

q_{sp} - supplemental and mechanical heat rate, kW,

q_e - energy rate in the exhaust air, kW,

q_l - energy rate in the leaking air, kW,

q_b - building conductive heat loss rate, kW.

Rearrange (6.9)

$$(q_a - q_b) + q_{sp} = q_e + q_l - q_i \qquad (6.10)$$

The right side of (6.10) is the net energy loss due to ventilation and leakage. Similar to the moisture balance case, it can also be expressed as the remainder of the maximum recoverable energy rate for the heat exchanger minus the actual recovered portion:

$$q_e + q_l - q_i = M_{max}(E_e - E_o)(1 - \eta) \qquad (6.11)$$

So

$$(q_a - q_b) + q_{sp} = M_{max}(E_e - E_o)(1 - \eta) \qquad (6.12)$$

It can be shown that starting with the mass balance (6.1), one can derive the same results as (6.6) and (6.12). In other words, both (6.6) and (6.12) satisfy (6.1).

If no heat exchanger is used, i.e. $\eta = 0$ and $M_{max} = M_e$ in conventional ventilation, the energy balance becomes

$$(q_a - q_b) + q_{sp} = M_e(E_e - E_o) \qquad (6.13)$$

On the other hand, when the heat exchanger is applied, if the exhaust and the incoming flow mass rates, M_e and M_i, are assumed equal, it leads to

$$\begin{cases} \eta = (E_i - E_o)/(E_e - E_o) \\ (q_a - q_b) + q_{sp} = M_e(E_e - E_i) \end{cases} \qquad (6.14)$$

So far, it is seen that (6.2) or (6.5), and (6.9) or (6.12) are the general expressions of the moisture and energy balances in animal housing, respectively.

6.4 Design Criteria of Ventilation Systems

ASAE D270.4 (1979) states that ventilation systems should maintain temperature and relative humidity in the animal shelter within desired limits and provide a desired amount of fresh air, without draft, to all parts of the shelter. In confined animal housing the minimum winter ventilation rate Q_w is to remove toxic gases, odors and dust, and to bring in fresh air (MWPS-1, 1980). Maintaining desirable temperature and humidity means satisfactory management of the energy and moisture balances discussed in previous sections. The general criteria for ventilation design and evaluation could be mathematically described as follows:

$$\begin{cases} Q = \max(Q_w, Q_t, Q_m) \\ q_{sp} = \max\{ q_b - q_a + W_a(E_e - E_o)(1-\eta)/[\beta(h_e - h_o)], 0 \} \end{cases} \qquad (6.15)$$

where Q - Q_e or Q_i correspnoding to M_e or M_i whichever is larger, m^3/s,

Q_w - minimum winter ventilation rate, m^3/s,

Q_t - temperature control flow rate, m^3/s,

Q_m - moisture control flow rate, m^3/s.

The recommended Q_w values for various species and size of animals can be found in MWPS-1 (1980). The temperature and moisture control rates, Q_t and Q_m, can be computed.

Consider the conventional case without the heat exchanger. From (6.7) and (6.13), it yields

$$\begin{cases} Q_t' = V_e(q_a - q_b)/(E_e - E_o) \\ Q_m' = V_e * W_a/(h_e - h_o) \end{cases} \qquad (6.16)$$

Q_t' and Q_m' are readily found if the design conditions of the indoor and outdoor air, the animals and the building are known. Then, the design ventilation rate Q_e can be selected based on (6.15).

Once the actual ventilation rate Q_e is greater than

Q_m', the indoor humidity h_e decreases until a new moisture balance is reached, consequently, a new Q_t' can be calculated, with which Q_e may be recompared. The real humidity ratio in the building is

$$h_e = h_o + V_e * W_a/Q_e \qquad (6.17)$$

ASAE D271.2 (1979) recommends the following correlations:

$$\begin{cases} h = 0.6219P_v/(P_o - P_v) \\ V = 287T/(P_o - P_v) \end{cases} \qquad (6.18)$$

where P_o - atmospheric pressure, Pa,

P_v - vapor pressure in Pa, ($P_v < P_o$),

T - dry bulb temperature in Kelvin,

($255.38 < T < 533.16$)

Applying (6.17), the following is derived

$$\begin{cases} h_e = (P_oQ_eh_o + 287T_eW_a)/(P_oQ_e - 461T_eW_a) \\ V_e = (287 + 461h_o)T_eQ_e/(P_oQ_e - 461T_eW_a) \end{cases} \qquad (6.19)$$

It is observed that any increases in Q_e will causes decreases in both h_e and V_e.

When Q_e is above Q_t', more sensible heat is removed, consequently, it is necessary to supply additional heat in the building.

$$q_{sp} = q_b - q_a + Q_e(E_e - E_o)/V_e \qquad (6.20)$$

Notice that the actual E_e and V_e from (6.19) should be used in (6.20). In this case, i.e. conventional ventilation, the criteria (6.15) becomes

$$\begin{cases} Q_e = \max(Q_w , Q_t', Q_m') \\ q_{sp} = \max[q_b - q_a + Q_e(E_e - E_o)/V_e, 0] \end{cases} \quad (6.21)$$

Turn to the case of the application of the heat exchanger. When M_e and M_i are assumed equal, and q_{sp} is zero, it yields from (6.8) and (6.14)

$$\begin{cases} Q_t'' = V_e(q_a - q_b)/(E_e - E_i) \\ Q_m'' = V_e * W_a/(h_e - h_i) \end{cases} \quad (6.22)$$

Compared with (6.16), it is evident that both Q_t'' and Q_m'' are greater than Q_t' and Q_m', respectively, because a part of moisture and energy returns back into the building.

If the correlations of E_e, E_i and E_o, and of h_e, h_i and h_o are known for a given type of regenerator, E_i and h_i could be estimated. For example, for the rockbed-type like that in this study the extrapolations of the regressing equations (5.9) in Section 5.5 and (5.11) in Section 5.6 may be used. Therfore, Q_t'' and Q_m'' can be estimated. The design flow rate Q_e can be determined with the first criterion in (6.15). Then Q_i can also be found since $M_e = M_i$.

If the actual ventilation rate Q_e exceeds the moisture control rate Q_m'', the following equation could be derived

$$(h_e - h_i)/(.6219 + h_e) = 461W_aT_e/P_o*Q_e \quad (6.23)$$

Applying the correlations of both enthalpy and moisture, e.g. (5.9) and (5.11), the actual indoor humidity h_e could be determined.

The supplemental heat q_{sp}, when $Q_e > Q_t''$, is

$$q_{sp} = q_b - q_a + Q_e(E_e - E_i)/V_e \qquad (6.24)$$

Also, E_i, E_e and V_e should be their actual values based on (6.23). In this special situation, i.e. $M_e = M_i$ with the exchanger, the criteria in (6.15) can be replaced by

$$\begin{cases} Q_e = \max(Q_w , Q_t'', Q_m'') \\ \\ q_{sp} = \max[q_b - q_a + Q_e(E_e - E_i)/V_e, 0] \end{cases} \qquad (6.25)$$

If M_i is not equal to M_e, but q_{sp} is zero, it leads to

$$\begin{cases} Q_t = V(q_a - q_b)/[(E_e - E_o)(1 - \eta)] \\ \\ Q_m = V * W_a/[\beta(h_e - h_o)] \end{cases} \qquad (6.26)$$

where the air specific volume V and the expected flow rate Q are either V_e and Q_e or V_i and Q_i corresponding to the mass flow rate M_e or M_i whichever is larger. In this case, a moisture removal factor β and a heat recovery effectiveness η need to be estimated first. The estimation of β may be made from correlation (5.6) if η is estimated. Values of η may be chosen within a reasonable range if sufficient experimental results are available for various types of heat regenerators under various real conditions. If expected Q used is larger than either Q_m or Q_t, the actual indoor humidity h_e and supplemental heat q_{sp} can be estimated based on (6.5) and (6.12) in a way similar to those discussed above.

In ventilation systems with exchangers, some estimation needs to be made for design purposes. Therefore, more experimental and analytical investigations are required to obtain comprehensive knowledge about the heat exchanger performance.

Finally, as to the design information, the local historical climatological data should be used to determine the outside design conditions. ASAE D270.4 (1979) and MWPS-1 (1980) recommend the use of:

1. the local low temperature exceeded more than 97.5 % of the winter time to determine the minimum continuous air change rate,

2. the local average daily temperature for January to estimate the maximum winter ventilation for preventing any net accumulation of vaporized moisture in the building,

3. the local hot weather data to calculate the maximum design ventilation capacity for keeping the indoor temperature the same or slightly warmer by 1 to 2 °C than the outdoor temperature.

It is noticed that the atmospheric humidity varies as the climate temperature changes in accordance with seasons. Therefore, both the outdoor temperature data and the corresponding data of the outside relative humidity should be selected and used in order to obtain more precise design parameters and predict more realistic indoor environment conditions. ASAE D309 (1976) provides the information dealing with the monthly maps of the mean wet-bulb temperature and the mean wet-bulb depressions over the United States.

7 CONCLUSIONS AND SUGGESTIONS

7.1 Conclusions

The following conclusions were reached in this study:

1. The regenerative system was applicable. The heat recovery effectiveness varied from 0.36 to 0.81, and the moisture removal factor was of 0.24 to 0.67 during these test days.

2. Cycle time affected the heat recovery effectiveness and the moisture removal factor significantly. Increasing the cycle time from 2 to 12 min resulted in a 13 % increase of the heat recovery effectiveness, but a 9 % decrease of the moisture removal factor.

3. The rockbed depth was not important in the tested range for the heat recovery effectiveness and the moisture removal factor. For the rockbed depth of 0.4 m or more, the heat recovery effectiveness and the moisture removal factor were appoximately constant.

4. Flow rates of the room exhaust and make-up air had strong interaction on the system performance. Essentially equal mass flow rates for both the exhaust and the make-up air led to high heat recovery effectiveness.

5. Flattened expanded metal sheets divided the sample rockbed into 0.1 m thick blocks, which caused a high

temperature gain up to 7°C in the make-up air, compared with that in the other rockbed.

6. The correlation of the heat recovery effectiveness and the moisture removal factor was found as:

$$\beta = -0.51 * \eta^3 + 0.61 * \eta^2 - 1.1 * \eta + 1$$

where β - moisture removal factor, decimal,

η - heat effectivenes, decimal.

Increasing the heat recovery effectiveness reduced the moisture removal factor significantly. In the design of a rockbed regenerative ventilation system, both the heat effectiveness and the moisture removal factor should be considered. A heat recovery effectiveness about 0.5 would be desirable.

7. Generalized balance equations of moisture and energy in animal housing are the following :

$$\begin{cases} W_a = M_{max}(h_e - h_o)\beta \\ (q_a - q_b) + q_{sp} = M_{max}(E_e - E_o)(1 - \eta) \end{cases}$$

where W_a - vapor production rate in housing, kg water/s,

M_{max} - either exhaust or make-up flow mass rate whichever is larger, kg dry air/s,

q_a - total animal heat production rate, kW,

q_b - housing conductive heat loss, kW,

q_{sp} - supplemental heat rate, kW.

8. Design criteria for animal housing ventilation could be mathematically expressed :

$$\begin{cases} Q = \max(Q_w, Q_t, Q_m) \\ q_{sp} = \max\{ q_b-q_a+W_a(E_e-E_o)(1-\eta)/[\beta(h_e-h_o)],0 \} \end{cases}$$

where Q - either Q_e or Q_i corresponding to M_e or M_i whichever is larger, m^3/s

Q_w - minimum winter control rate, m^3/s,

Q_t - temperature control rate, m^3/s,

Q_m - moisture control rate, m^3/s.

7.2 Suggestions

The following suggestions were made for further study:

1. Further investigation on the effects of the cycle time, air flow rates and their interactions on the heat recovery effectiveness and the moisture removal factor under a constant rockbed depth.

2. Increasing both exhaust and make-up air flow rates and managing mass rates of exhaust and make-up air as similar as possible in further study.

3. Further study on the effect of mesh screen materials on the rockbed performance.

4. Blocking up two rock bed from their bottom to avoid short circuiting.

5. Development of effective dust filter devices for the room exhaust air path.

6. Analysis of energy efficiency of the whole building-ventilation system, including supplemental heaters, solar collector, rockbed regenerator and so on, as well as economic analysis of the system operation.

7. Study on new regenerators which recover heat but do not recirculate moisture from the exhaust air to the room.

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