PAPER NO. 83-4019

COGENERATION FOR HEATING GREENHOUSES

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For presentation at the 1983 Summer Meeting
AMERICAN SOCIETY OF AGRICULTURAL ENGINEERS

Montana State University
Bozeman, Montana
June 26-29, 1983

SUMMARY: A nominal 30 kW natural gas-fueled engine has been used to provide heat and electrical power for a greenhouse system. The generator is attached to the utility grid. The waste heat from the engine's cooling jacket and exhaust heat exchanger is delivered to the greenhouse via a flooded floor heat exchanger storage system. This paper reviews the performance of the cogenerator as matched to the greenhouse's energy requirements.

American Society of Agricultural Engineers

St. Joseph, Michigan 49085

COGENERATION FOR HEATING GREENHOUSES

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INTRODUCTION

Cogeneration units consisting of electric generators driven by a fossil fuel engine with heat recovery can be used to provide energy for greenhouse operations. This concept is particularly attractive when a low grade fuel is readily available. Cogenerators have the ability to transform this low grade fuel into more valuable energy i.e. electricity, as well as provide a good source of hot water for space heating. Greenhouses inherently use large amounts of heat and a substantial amount of electricity. For this investigation a cogenerator has been used in place of a boiler, providing heat when necessary to a research greenhouse and producing electricity. Most of the time the unit was run on heat demand producing electricity as a by-product. Although pipeline natural gas was used in these tests, it would be anticipated that digester gas could be used.

Normally a greenhouse could not utilize all the electricity produced by the cogenerator if it was sized to provide heat, however, since the enactment of the Public Utilities Regulatory Policies Act, electrical utilities are required to purchase power from small producers under certain criteria (---- Meter Beater pamphlet, undated). Therefore any electricity not being used in the greenhouse complex could be sold to the utility. Electricity in New Jersey is sold retail from 8 to 11 cents per KWHR but when selling electricity back to the utility it is only worth 2.5 to 6 cents per KWHR so it is to the producer's advantage to utilize as much of the electricity produced as practical.

Cogenerators are not new to industry and have been used for decades. Despite obvious fuel savings associated with cogeneration the number of industries using cogeneration has been decreasing due to low rates offered for purchased electric power by utilities to industry. With the prices of electricity rising, industries have again begun to look at cogeneration as a way to relieve high energy bills (Gerlaugh et al.,1980). It has only been very recently that cogeneration is being used by smaller businesses. In general, small businesses which have begun to look at cogeneration most seriously have access to some form of cheap fuel such as bio-gas or landfill gas. Methane produced from manure, sewage treatment plants or landfills can be used to produce electricity and heat. This process is being demonstrated at a Cornell research facility at which a dairy farm attempts to be self sufficient for

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New Jersey Agricultural Experiment Station, Publication No. P03130-10-83 supported by State Funds, U.S. Hatch Act and Public Service Electric and Gas Company funds.

energy (Koelsch et al.,1982).

In this study conducted at Rutgers, Public Service Electric and Gas Company has provided a cogenerator and the gas supply used in a 558 m^2 greenhouse facility. The nominal 30 kw unit has at this writing operated for 2500 hours and can produce a considerable amount of the greenhouse's heat and electrical needs. It is the objective of this paper to review the performance of the cogenerator as matched to the greenhouse's energy requirements. The data on the performance of the unit has been analyzed to enable studies to be conducted on alternate control strategies for the unit. The optimum matching of the unit's capacity and operating time to the heat and electrical demands of a greenhouse facility are necessary if the most economical possible use is to be made of the unit.

DESCRIPTION OF THE TEST FACILITY

The greenhouse used to test the cogenerator at Rutgers Horticultural Farm #3 in New Brunswick, New Jersey, is a double polyethylene glazed, gutter connected house. Its floor space is 18.3 x 30.5 m. and is directly connected to an existing 11 x 30.5 m. greenhouse. The heating system for the greenhouse is identical to flooded floor storage solar systems which have been described in detail in many publications (Manning et al.,1980 and Mears et al., 1981). Figure 1 illustrates the construction of the floor storage system which is comprised of a vinyl liner, gravel and water, and a porous concrete cap. Water is circulated through the gravel past the heat exchanger via sump pumps in opposite corners of the greenhouse, providing equal heat distribution of the warmed floor water. See figure 3 for water circulation patterns. This figure also shows the floor divided into two sections, which was done to allow for independent temperature control for each section.

A major deviation from the solar plan is the fact that the pipe loop heat exchanger is now the main greenhouse heat source. The exchanger has a 3.2 cm header with 4 aperatures. Into each openning a 1.9 cm steel pipe is placed and is then looped back twice so the total number of 1.9 cm pipes comprising the heat exchanger is 16 (figure 2). The headers are linked to the engine's cooling jacket heat exchanger.

The design parameters for the heat exchanger were obtained from Perennial Energy and from the previously known operating conditions and size of the greenhouse. The engine can emit 45 kW of heat energy at a water temperature of approximately 60°C, and this must be removed in order to prevent the generator from dumping heat through the radiator. The heat exchanger must also work in a floor temperature region of approximately 27°C and it must fit into an area of 0.3 x 0.7 x 7.3 m that will have 227 l/min of water passing through it.

The capabilities of the heat exchanger unit were determined by standard heat transfer principles. The equation:

$$Nu=h_oD/k=.638Re^{.466}Pr^{.333}$$

where: Nu=Nusselt's Number
h_o= outside pipe heat transfer coefficient
D= pipe diameter

k= Thermal conductivity
Re= Reynolds Number
Pr= Prandtl Number

is for heat transfer for laminar flow over cylinders (Ozisik, 1977). The equation:

$h_i D^{.2} = 150(1+.011T)(V)^{.8}$

where: h_i = inside pipe heat transfer coefficient
T = water temperature
V = water velocity

is for heat transfer for turbulent flow within cylinders (--- ASHRAE Handbook, 1981).The heat transfer coefficients were determined to be 278.1 and 4615 W/m^2K respectively to produce a total heat transfer coefficient of 262 W/m^2K where:

$1/h_t = 1/h_o + 1/h_i$

Operating water temperatures were expected to be 60 °C from the engine with an 11 degree drop across the pipe loop heat exchanger and floor storage water temperature was expected to be 26.7°C with a 2.7 K rise. Therefore using the log mean temperature difference (LMTD) for parallel flow with the equation:

Surface area of heat exchanger= heat output/ h_t * LMTD

The theoretical surface area of the heat exchanger is calculated to be 6.5 m^2. This equates to 13 lengths of pipe, and to accommodate this many pipes a configuration of 16 pipes was chosen as shown in figure 2. This design also allows for a safety factor as the actual heat exchanger has 8.5 m^2 of surface area.

From the calculations above it is necessary to pump 56.7 l/min through the system at a head of 4.1 meters. This head includes the heat exchanger and delivery system friction heads for piping and fittings.

Heat recovery from the engine is diagrammed in figure 4. Starting from the engine, the engine's cooling jacket water is passed through the exhaust heat exchanger further warming the water by incorporating some of the exhaust heat which ordinarily would be lost. The now 88 °C cooling water passes through the radiator which usually does not operate, so all the collected heat is transferred through a small cross flow heat exchanger. The second part of the heating system is warmed by the cooling jacket heat exchanger and is sent to the floor loop heat exchanger shown in figure 2. It is also this part of the system where boiler water can enter in cases when the cogenerator is not running. The final stage of greenhouse heating is when the floor storage water passes by the floor pipe loop as diagrammed in figure 3.

Electricity is generated by the single phase, 30 kW induction generator. The generator is powered by the Waukesha vrg 220 engine, and it produces voltage and frequency to the exact specifications of the public utility and the power produced varies with the speed of the unit. Figure 4 shows the location of the generator. Power produced from the generator is shown going to the grid and the load of the research greenhouse complex. The power meter or Energy Wattcher* is attached to the generator output to monitor the unit's electrical performance.

The natural gas powered cogenerator unit includes an engine, generator, exhaust heat exchanger, radiator, cooling jacket heat exchanger, safety switches and controls which were packaged by Perennial Energy, Inc. (PEI), Dora, MO.

The cogeration unit is housed in a small wooden shed which is connected to the greenhouse. The 5 x 5 m shed is divided into two rooms, one room houses the cogenerator and the boiler, which provides touch-up air heating and serves as the main back up heater for the greenhouse. It is capable of maintaining a 33 K ΔT between outside and inside ambient air temperatures. The second room stores the data acquisition system and computer as well as providing work space.

EXPERIMENTAL DESIGN

A primary objective of this investigation is the determination of the cogenerator efficiencies i.e. thermal and electric. Thermal energy output was determined through the use of a Corad multi-jet magnetic water meter, which measured the flow of water passing through the floor heat exchanger, and platinum resistance thermometers (RTD's) measuring the change in temperature across the heat exchanger. Whenever the circulator is on, the water flow rate through the floor heat exchanger is a constant 58.7 liters/ minute.

The electrical power output was measured directly from an instrumentation board which is placed within the microcomputer's S-100 bus. Utilizing clamp-on inductive amp probes and a volatage transformer, the Energy Wattcher, measures the power output from the cogenerator in watts. The Energy Wattcher was purchased from Scitronics, Bethlehem, Pa.

Since the cogenerator is fueled by natural gas (methane) from the utility the energy input to the cogenerator was measured by a standard gas meter. A time clock recorded the hours the generator ran.

Secondary objectives such as determination of heat transfer rates from the floor and greenhouse, and verification of various system heat transfer coefficients required numerous thermocouples, RTD's, event recorders and solarimeters. Data acquired from these devices were compiled by a Doric Digitrend 235A. The temperature data, averaged and recorded hourly, and the event recorders, recorded 12 times per hour, were recorded on paper tape. The 235A also can transmit data via cable to peripherals such as magnetic tape, paper punch tape or computers. In this case, a Northstar Horizon 64K computer was used so data was stored on floppy disks. Floppy disks were removed periodically from the greenhouse computer to be analyzed by similar computers at other locations.

Greenhouse temperatures were being recorded from various locations in the

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*Reference to commercial products is made with the understanding that no discrimination and/or endorsement is intended or implied.

air and floor. Thermocouples and RTD's also measured outside temperatures and ΔT's across the overhead heat exchanger as well as the floor heat exchanger. Event recorders were used to keep track of run time of four water pumps, the CO_2 generators and the backup boiler.

PERFORMANCE OF THE COGENERATION UNIT

During this research program the cogeneration unit was operated in several different ways in order to determine potential performance under varying operating conditions. The unit was operated manually at full load for a number of extended operating periods and manually for fixed periods at various throttle settings. During a major portion of the season the unit was operating as a boiler on heat demand as determined by a thermostat in the floor.

A number of test runs were made at variable throttle settings during which fuel consumption, electric power output and heat recovery were measured. The electrical efficiency which was calculated by dividing total electric energy output by total energy content of fuel consumed in each test was plotted against production capacity as a function of rated capacity at the bottom of figure 5. These results indicate that electrical generation efficiency increases linearly with load with the best linear fit of the data from 60 to 90% loading being:

Electrical Efficiency = 3.2 % + 0.22 production capacity

At 80 to 90% loading the mean electrical efficiency is 21.3%.

For these same tests the thermal efficiency was determined by dividing the total thermal energy recovered and delivered to the greenhouse by the total energy content of the fuel consumed. These results which are presented at the middle of figure 5 show no significant dependence of thermal efficiency onloading and the average thermal efficiency is 42.9 %. When combining the thermal and electrical efficiencies to get a total energy efficiency, the significance of the increase in electrical efficiency with load disappears. The combined data showing total efficiency is presented at the top of figure 5 and the mean total efficiency from 80 to 90% loading is 63.5%.

It should be noted that the transducer measuring electrical power output which is being integrated measures true power output in watts. The power factor was measured independently by taking a number of hand readings of volts and amps independently and it was found that the average power factor measured at the generator output was 83% $\pm$ 1%. This is reasonable in this application as almost all the electrical load in the greenhouse complex consists of induction motors and HID lamps with rated power factors of 80 to 85%. It could be reasonably expected that if the generator could be operated at unity power factor about a 5% increase in electrical generation efficiency could be expected.

In assessing the thermal performance of the particular unit being tested, it is clear that there are significant possibilities for enhancing heat recovery from the unit. There is no attempt on this unit to recover the resistance electrical losses in the generator or the radiant and convective heat losses from the engine block and exhaust manifold. Also, measurements of exhaust gas temperatures after the heat exchanger indicate that there are significant recovery possibilities for a condensing exhaust gas heat exchanger. Significant energy could be recovered during engine operation by circulating greenhouse air through the building housing the cogenerator taking care not to introduce exhaust gases or crankcase ventilation exhaust into the growing area. For heat storage an air to water heat exchanger could be utilized.

There were a number of design problems on the prototype unit and a great deal of effort was expended in making the unit operational. There were several significant logical design problems in the control unit that had to be redesigned out before the unit could operate in the automatic mode. Additional improvements in this area are still needed. A major redesign of the mounting of the exhaust heat exchanger was required to isolate the mass of the exhaust heat exchanger from the engine. In the original design the close mechanical coupling of the heat exchanger caused severe vibrational stress in the exhaust manifold leading to frequent failures of the exhaust gasket. The unanticipated mechanical and electrical control problems caused delay and expense to correct, but are relatively straightforward design problems to solve.

THERMAL PERFORMANCE OF THE GREENHOUSE

In order to determine the thermal performance of the greenhouse and its heating system, several controlled experiments were conducted and energy flows were monitored during normal operation throughout a cropping season. In a series of tests conducted during a period when there was no crop in the greenhouse and air temperatures could be significantly varied, measurements were made to determine the heat transfer characteristics of the structure and the components of the heating system.

The heat transfer coefficient based on glazed area for a gutter connected, double polyethylene covered greenhouse has been found to average 4.59 w/m^2K (Roberts et al, 1981). It has been found that instantaneous measurements of the heat transfer coefficient for such greenhouses vary widely, but that mean values determined by totalling energy flow and temperature differences over long time periods produce very consistent measurements. This is because heat loss which is the total of radiation, infiltration and conduction/convection heat loss is highly dependent on a number of weather and internal environmental conditions. These include inside and outside temperatures and relative humidities, wind speed, and cloud cover and condensation on the inner glazing.

In a series of tests conducted in January, 1983 the heat transfer coefficient of the test greenhouse with a thermal curtain in place was determined. The curtain material tested was an aluminized polyethylene installed to move from truss to truss across the greenhouse ceiling. An extension of the ceiling curtain moved from post to post to close off the north wall. The south side of the test section was closed off by a vertical curtain from another greenhouse kept at the same temperature as the test section. The east and west walls are glazed with a rigid double layer of polycarbonate, but were uncurtained. As supplemental lighting was to be used in horticultural experiments, the curtain was installed with the aluminized side down. The curtain was perforated to allow for drainage of condensate. The measured heat transfer coefficient for the insulated roof and north wall which total 892 m^2

glazed area is 2.55 ± 0.43 W/m^{2}K. The double glazed end walls total 112 m^2 and have a heat transfer coefficient of 3.69 W/m^{2}K (--- GE Sheet, undated).

Determination of the heat transfer characteristics of a flooded floor is difficult due to the large thermal mass. An energy balance over time involves accounting for heat added to the system, heat transferred up to the greenhouse air, heat loss down to the ground and the change in sensible heat stored in the floor. As the thermal mass is large, small changes in average floor temperature relate to significant amounts of energy. In this case the energy input to the floor can be measured most accurately as temperature difference across the heat exchanger is measured with RTD's accurate to 0.05°C and the flow rate through the exchanger is measured with a flow sensor accurate to 0.4 l/min. Heat loss down has been determined to average 3% of the heat flux upward from such floors and can, therefore, be neglected in most cases (Mears et al, 1981).

From a series of energy balances taken during the heating season, the thermal mass of the floor and the heat transfer coefficient were determined. The total thermal mass of the floor is 456 MJ/K which is 0.82 MJ/m^{2}K of floor area. This greenhouse was operated with a slightly lower water level than the solar demonstration greenhouse reported on by Mears et al (1981) which had a thermal mass of 0.9 MJ/m^{2}K. The heat transfer coefficient of the floor determined by the difference in the average temperatures of the water in the floor and the greenhouse air was measured to be 6.2 ± 1.4 W/m^{2}K which is consistent with previous measurements in solar floor systems (Mears et al, 1981) and waste heat utilization systems (Manning and Mears, 1980).

The steel pipe floor heat exchanger's (figure 2) actual performance compared quite closely to the design calculations. Some of the statistics, discussed in the design section, are listed here:

Flow thru heat exchanger	- 58.7 l/min.
Flow over heat exchanger	- 227.1 l/min.
ΔT thru heat exchanger	- 10.0 K
ΔT over heat exchanger	- 2.6 K
LMTD (parallel)	- 13.3 K
Heat transfer coefficient	- 361 W/m^{2}K

Although the ΔT's and flows are similar to the design values the LMTD and the heat transfer rates were incorrect by a factor of approximately 0.5 and 1.4 respectively. One inaccuracy in the design was due to lower water operating temperatures from the engine. However, because of a higher heat transfer coefficient and extra surface area the amount of heat transferred was more than adequate.

In this application the cogenerator unit is plumbed to deliver heat to the floor storage which then warms the greenhouse. The backup boiler is normally connected to the supplemental overhead heating system and can provide touch up heat as needed under control of a thermostat sensing air temperature. With the thermal mass of the floor and a nominal heat output from the cogenerator of 45 KW it would take 8.4 hr to satisfy a 3K hysterisis on the floor temperature sensing thermostat controlling the cogenerator if there were no heat loss from the floor during that time. On the other hand, if the water in the floor were 13K warmer than the greenhouse air, the cogenerator would add heat to the floor continuously with no increase in storage temperature. In actual practice there is substantial heat transfer from the floor so duty cycles on the cogenerator are significantly longer than 8.4 hours.

The duty cycle of the cogenerator operating on floor thermostat control is shown in the solid line at the base of the graph (figure 6). Also shown on the same graph are the average floor water temperature, greenhouse air temperature and outside air temperature. During this period the insulating curtains were closed at 5 PM and opened again at 7 AM. The energy totals during this period are:

2373 — Gross KWHR electric power produced by the cogenerator
3547 — Gross KWHR electric power required by Horticulture farm complex
221 -- Net KWHR electric power exported to the grid

16.3 -- Gross GJ thermal energy delivered to the greenhouse floor by cogenerator
9.8 -- Gross GJ thermal energy delivered to the greenhouse air by the touch-up boiler
20.6 -- Total GJ thermal energy demand of the greenhouse required to maintain minimum setpoints

The numbers above represent the results of the dual outputs of energy from the cogenerator. Determination of the value of the energy produced depends on time of day and usage. Two ratcheted meters, between the grid power and the farm's wiring, measure the incoming or outgoing electricity. These meters measure KWHR as well as time of day. During the time period indicated on figure 6, the Horticulture farm complex required 3547 KWHRs of electricty. The generator has produced 2373 KWHRs of electricity. Therefore the farm must have purchased 1395 KWHRs of retail electricity. This represents the potential savings of $237.30 at retail rates assuming all this energy was utilized before it went to the grid. The value of this electric power could represent as little as $59.33 if all the electricity had been sold back at off peak hours. As shown above the amount of electricity actually going out to the grid was only 221 KWHR. Most of the energy was utilized, but even retail pricing or savings depends on time of day and therefore an exact amount of money saved must be determined by totalling the hour by hour dollar savings over an entire season.

The second set of three numbers shown above represent heat or thermal energy used by the greenhouse. The energy required to warm the greenhouse during the time period indicated in figure 6 was 20.6 GJ. By adding the remaining numbers it is evident that the greenhouse used 26.1 GJ representing 5.5 GJ of energy that went into the greenhouse and maintained air temperatures above the set point.

Setting the floor thermostat higher increases the occurrence of providing more than the minimum heat, but reduces the touch-up heat needed. Most of the 9.8 MJ of touch-up heat was utilized to bring the greenhouse up to the daytime set point early in the morning. Determination of optimum floor operating temperatures needs further work. It is important to realize that the floor thermostat can be set higher than normal because although the greenhouse may use some heat inefficiently the cogenerator will still be producing electricity. Also, the cogenerator may be using a cheap source of fuel while the boiler may be using higher priced fuel. This is important in economics as electric generation goes with over supply from the cogenerator, but not from the boiler. Even with the use of high cost natural gas, the cogenerator will produce enough electricity and heat to make a profit (from an energy standpoint)as compared to standard greenhouse heating and electrical uses. However, because of high capitol investment, maintenance, and cost of fuel, a cogenerator of this size can have a profit margin ranging between 1 and 6 $/hour for different scenarios. The lower end of the scale represents an infeasible technology while the other end of the scale enables cogeneration to be an economic way to power a greenhouse complex.

During other periods the cogenerator unit was continuously on manual control. In this mode of operation, the floor water temperature will seek an average value more than 13K warmer than the average greenhouse air temperature. This is because there is some direct solar heat gain on the floor surface during the daytime which reduces heat transfer upwards from the floor surface.

SUMMARY AND CONCLUSIONS

The basic performance characteristics of a cogeneration unit used to heat a greenhouse have been determined along with the electrical and thermal efficiencies of a particular unit. The utilization of the electrical and thermal outputs of the unit vary with the method of control of the unit's operation. Electrical output is maximized by continuous running, but thermal output is not fully utilized. By coupling the thermal output to a heat sink in the greenhouse and operating the cogenerator as a unit to maintain thermal storage above a set point temperature, the utilization percentage of the thermal output increases.

When the thermal storage unit is massive relative to the heat load, as was the case in this test, a realistic hysterisis on the thermostat will insure reasonably long duty cycles on the cogeneration unit. Increasing the set point temperature of the floor storage increases duty cycle time, electrical energy production, the contribution of the cogeneration unit to the thermal demand of the greenhouse and the excess heating of the greenhouse. The data gathered in this study can be used as the input for a simulation study to determine the optimum management procedure for utilization of a cogeneration unit.

It should be noted here that any study to determine the economics of using cogenerators in greenhouses must carefully consider a number of important parameters. The capitol costs of the unit and the greenhouse heating system need to be amortized against the electrical and thermal outputs. Therefore utilization of both heat and electricity are key factors particularly if greenhouse lighting for crop production is contemplated. Analysis of the value of electricity produced is complicated by the fact that there are different values for electricity utilized by the greenhouse and exported to the grid. Furthermore, both of these rates vary with time of day and month of the year. Thus simulation studies which consider the timeliness of energy generation under various management strategies are needed. Simple energy balances provide important insight, but do not answer all questions. More work along these lines is needed before practical systems can be designed.

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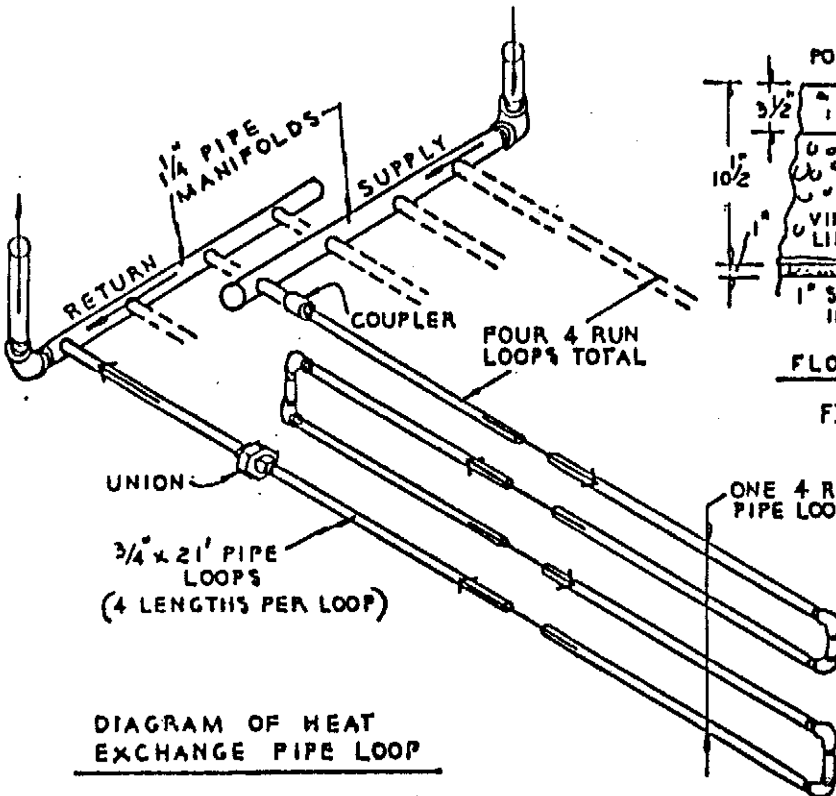
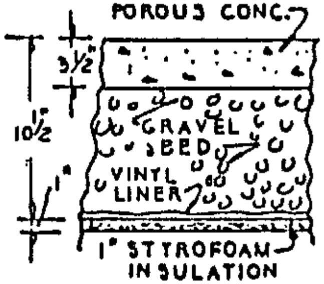


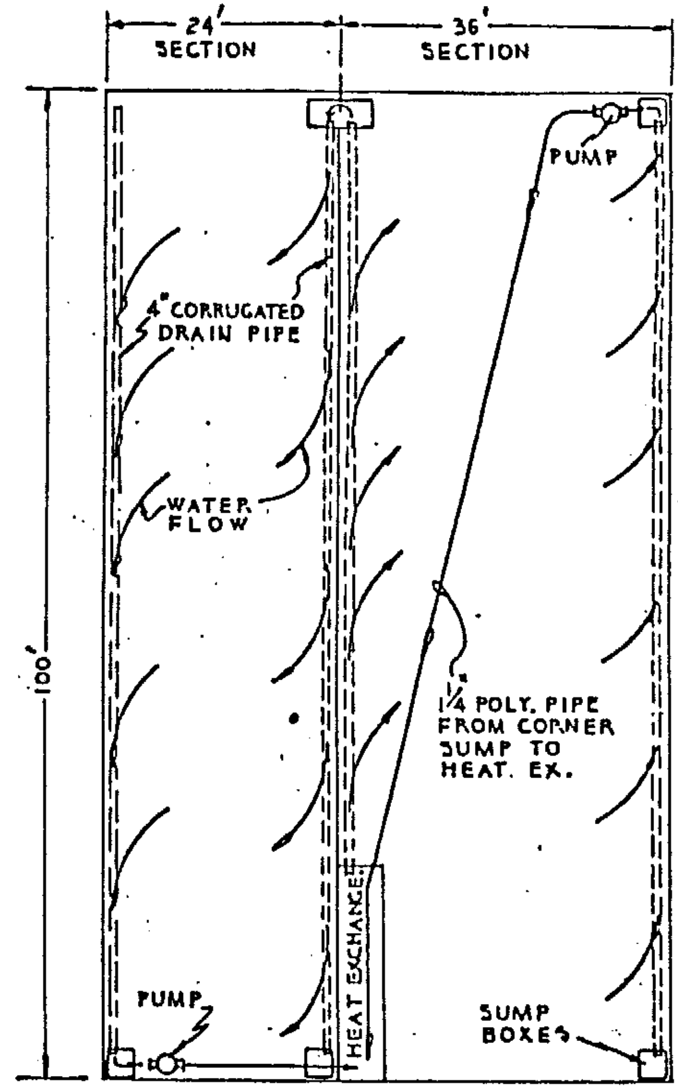
DIAGRAM OF HEAT EXCHANGE PIPE LOOP

FIGURE 2



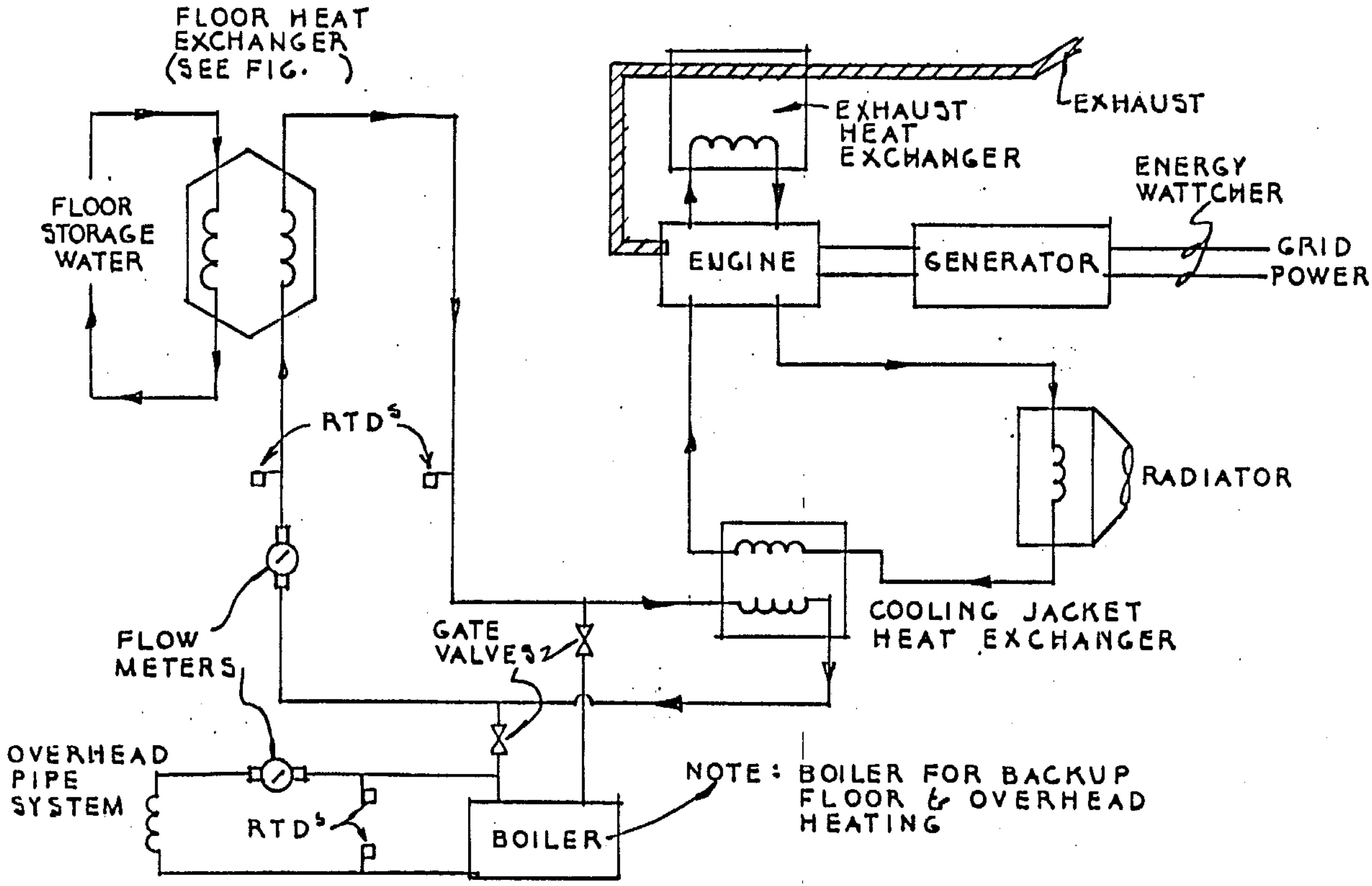
FLOOR SECTION

FIGURE 1



PLAN—HORT. FARM #3 RESEARCH FACILITY

FIGURE 3



SIMPLIFIED SCHEMATIC OF CO-GENERATOR
HEAT RECOVERY SYSTEM

FIGURE 4

EFFICIENCIES VS. PRODUCTION CAPACITY

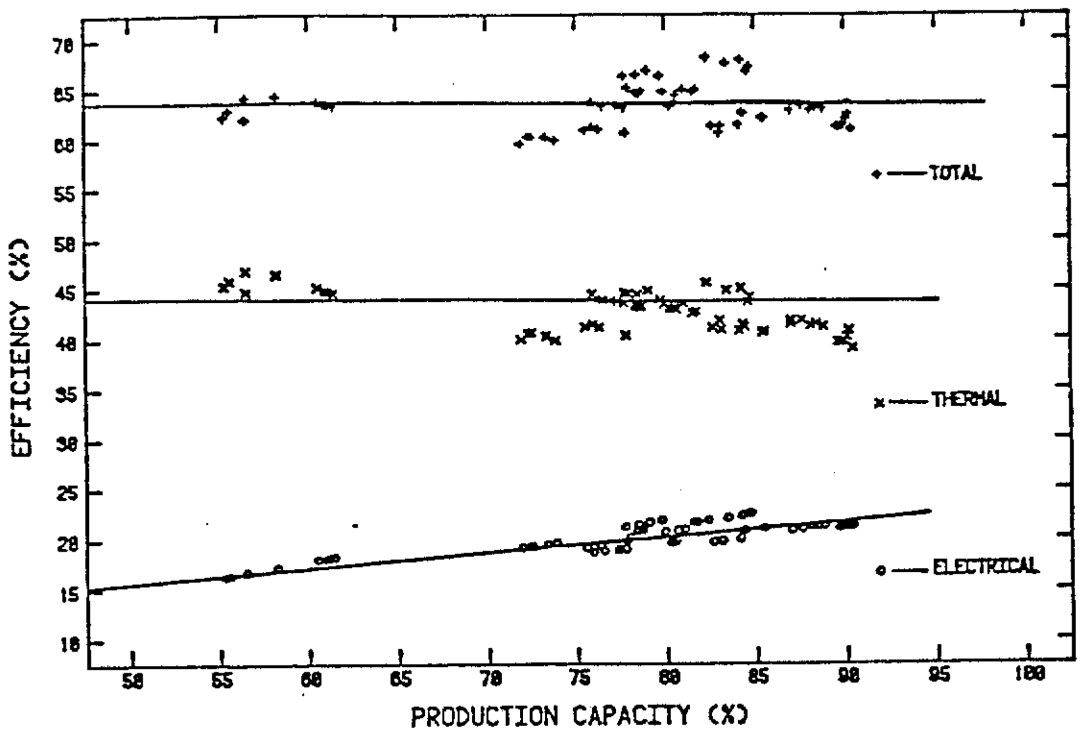


FIGURE 5

GREENHOUSE TEMPERATURES

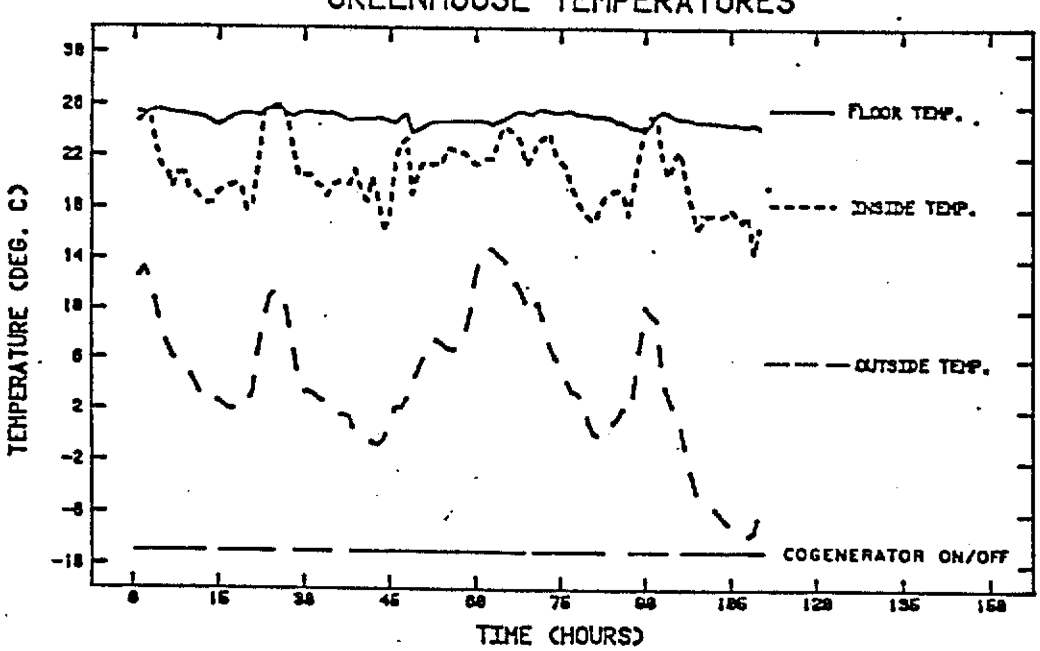


FIGURE 6

TABLE 1
PARTIAL DATA FOR AVERAGE U VALUE OF GREENHOUSE FLOOR
HOURS BETWEEN 12 AND 6 A.M.

DATE	AVG FLOOR TEMP.(°C)	AVG. OUTSIDE TEMP.(°C)	AVG. INSIDE TEMP.(°C)	AVG. FLOOR U VALUE(W/M^2K)
12/25	23.40	4.14	15.19	9.41
12/26	23.87	15.60	18.56	4.82
12/27	23.42	0.01	12.26	7.83
12/28	24.39	6.73	15.44	7.04
12/30	22.77	5.15	10.20	5.89
12/31	23.78	5.75	13.89	5.93
1/1	24.80	0.58	11.75	6.37
1/2	24.90	-1.71	9.41	4.96
1/3	24.52	1.23	11.22	5.37
1/4	23.90	7.07	10.70	4.59
AVG.				6.22

TABLE 2
PARTIAL DATA FOR DETERMINING U VALUE FOR GREENHOUSE
WITH INSULATING CURTAIN

DATE	AVG. FLOOR TEMP.(°C)	AVG. INSIDE TEMP(°C)	AVG.OUTSIDE TEMP.(°C)	AVG. GHOUSE U VALUE(W/M^2K)
1/12	25.2	19.0	-0.1	2.84
1/13	24.6	17.5	-6.5	2.09
1/14	24.5	19.4	-8.5	2.26
1/15	25.4	17.8	1.1	2.03
1/18	26.0	18.5	-9.1	2.33
1/20	25.7	17.3	-11.6	3.00
AVG.				2.55