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Environmental Control in Poultry Laying Houses

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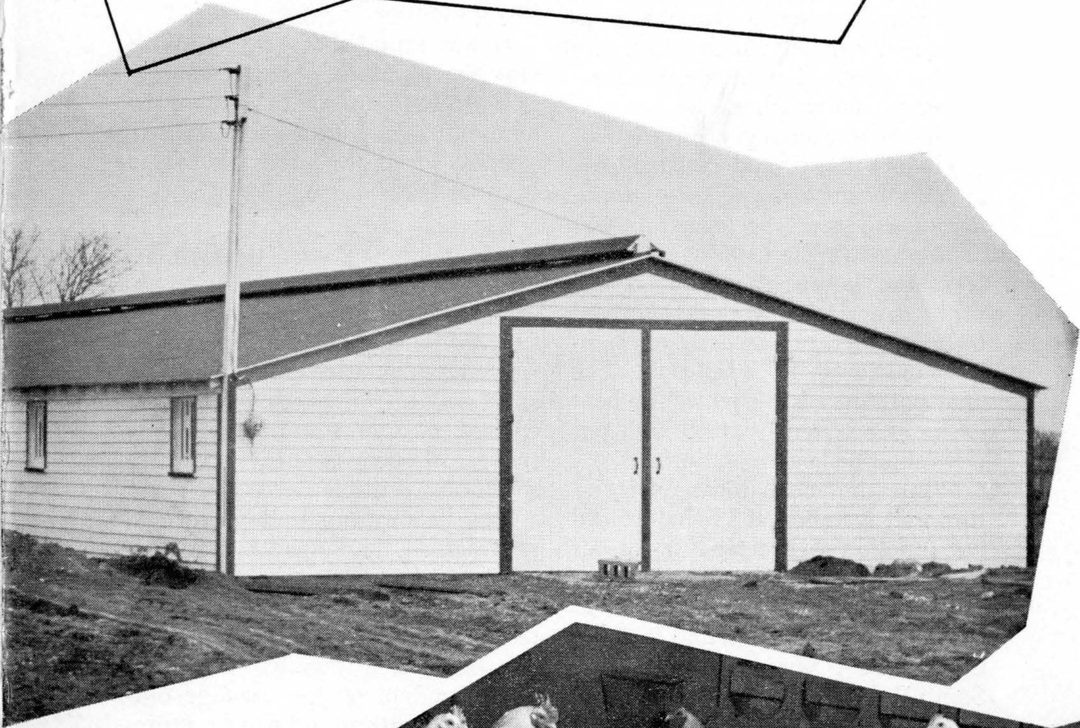

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SUMMARY

The ventilation studies conducted at Highmore did not prove satisfactory because of the low bird density that existed in the laying house. In addition, the house was not adequately insulated which made any rate of ventilation impossible during very cold weather.

The heat exchanger installed at Highmore performed satisfactorily and under normal operating conditions recovered approximately 50% of the heat in the exhaust air. Dust from the house proved a serious problem and the heat exchanger had to be cleaned periodically to maintain efficiency of operation. The energy required to operate the heat exchanger system was 18 kwh per day under normal operations.

While the mechanical refrigeration unit installed at Highmore did not function satisfactorily as an air to air heat pump, information gains from the study proved valuable in the subsequent design and installation of heat pump equipment at South Dakota State College in 1958.

The heat pump system used in the 32 x 50 foot cage layer house at South Dakota State College kept the poultry house inside temperature and relative humidity at satisfactory levels for overall conditions during the test period. During extremely warm outside weather, the temperature inside the house did rise somewhat above the design condition of 70°F. during the middle of the afternoon but this was not considered serious because of the short duration of this higher temperature. The relative humidity in the house was quite uniform during the entire test period averaging between 60 and 70%. A statistical analysis of outside temperature data and inside relative humidity revealed there was no significant relationship between the two quantities.

It was found that up to 2 cubic feet per minute per bird of ventilation air was required to prevent the ammonia odor from exceeding comfortable levels for humans working in the house.

Cleaning the air from the poultry house before it passed through the unit coils proved to be the main obstacle to trouble-free operation. Conventional permanent filters became clogged with dust in a very short time and an experimental centrifugal cleaner was only partially successful because of the extremely small dust particle size.

The heat pump unit performed satisfactorily having an average coefficient of performance of 1.7 for cooling and 2.8 for heating. For the test period the power consumption rate for the entire system proved to be dependent on the average outside temperature and can be represented by the equation: $Y = 5.423 - 0.20639X + 0.003266X^2$ where Y is the power consumption rate in kilowatts and X is the average outside temperature in degrees Fahrenheit. The minimum power consumption occurred at an average outside temperature of approximately 30°F.

The calculated yearly cost of owning and operating the 6 horsepower heat pump system, assuming a 10 year equipment life, amounted to $786. It is probable that this could be reduced by improved dust removal or filtering methods.

CONTENTS

Tables

Figures

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Enviromental Control in Poultry Laying Houses

Donald D. Hamann, Harvey G. Young, Dennis L. Moe[1]

A well designed and constructed poultry laying house should provide substantially more than merely a place for poultry to be sheltered and a place to roost. Farmers in South Dakota have realized for years the importance of good, sound poultry house construction, both from the standpoint of serviceability and from evaluation, as an asset to the farm. Only in relatively recent years, however, have many poultrymen become aware of the importance of poultry flock environment in production costs.

In constructing a new poultry house or remodeling an existing building for poultry, careful planning and design should be the first step. The poultryman should be ever conscious of the eventual use the structure is to provide. A wise choice of materials, serviceability, labor saving devices, and environmental equipment should be made from a functional and economical aspect. Some items to consider in any poultry house for the laying flock are: (1) What is the desired winter temperature? (2) What is the desired summer temperature? (3) What extreme temperature changes may take place? (4) What excessive moisture conditions may develop? (5) How can labor time and cost be reduced? (6) What equipment is needed to aid in the performance of the above listed items and what will it cost for installation and operation? High production at a reasonable cost is the aim and desire of poultry laying flock owners.

Environmental control in a poultry laying house in South Dakota during the winter months consists of control of moisture accumulated from droppings, waterers, and respiration; removal of ammonia fumes and dust; and maintenance of desired temperature. Assumptions are normally made that bird density, insulation, and ventilation are at a point that will prevent temperatures below freezing under normal operating conditions. During the summer the demand for dust removal increases in addition to the problem of sufficient air movement at a temperature difference to produce a temperature radiation for a cooling effect. In this process the incoming air must naturally be cooler than

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the exhausted air. In any system concerned with moisture removal, cooling, or heating, always remember that insulation, ventilation, and environmental control all must work together.

POULTRY HOUSE CONSTRUCTION

A 24 by 34 foot poultry house designed for 250 birds was constructed at the Central Substation, Highmore, South Dakota, in 1952. The house was constructed from Extension Service Plan 313 and was used for the major portion of the ventilation studies. The walls were insulated with a 1 inch balsam wool blanket located so there was an air space on each side. The ceiling was insulated with 3 inches of vermiculite and the windows were provided with storm sash for winter operation.

Ventilation was supplied by a two-fan system. The low volume fan had a capacity of 600 cubic feet per minute (cfm) and was controlled to operate continuously at above freezing temperatures in the house. The large volume fan had a capacity of 1,800 cfm and operated at house temperature above 41°F. Fresh air was admitted to the structure through one-half inch slots in the ceiling adjacent to the side walls. These intakes drew air from the attic since this provides a warmer source than drawing fresh air directly from outside.

The structure was divided into four pens, each housing an experimental flock of 55 birds. The total of 220 birds was somewhat less than the unit was designed to accommodate.

The two-fan ventilation system was operated from 1952 until January 22, 1954, and proved unsatisfactory during cold weather since inside temperatures were recorded as low as 4°F. during extremes in weather conditions. A large part of the time heat lamps were operated over the water pipes and water to prevent freezing. The average electrical consumption for ventilation and heat lamp operation was 17.1 kilowatt hours (kwh) per day.

VENTILATION STUDIES INVOLVING THE USE OF A HEAT EXCHANGER

A heat exchanger unit designed for the poultry house at Highmore was constructed during the summer of 1953. It consisted of 20 two-inch diameter sheet metal tubes 28 feet long, arranged parallel in a duct. A centrifugal blower driven by a one-third horsepower motor forced air from the attic into the duct space surrounding the tubes. This air passed through the duct, and was discharged into the house through a distribution duct in the ceiling. A second centrifugal fan drew air from the ceiling at one end of the building and forced it through the tubes. It was then discharged outside. Figure 1 shows the heat exchanger system and the air flow patterns.

The unit was designed for an air flow of 520 cfm on each path, and it was expected that, with a 25 degree temperature difference (outside temperature 10°F., inside temperature 35°F.) that 9,400 Btu of heat would be recovered by the exchanger per hour. This represents

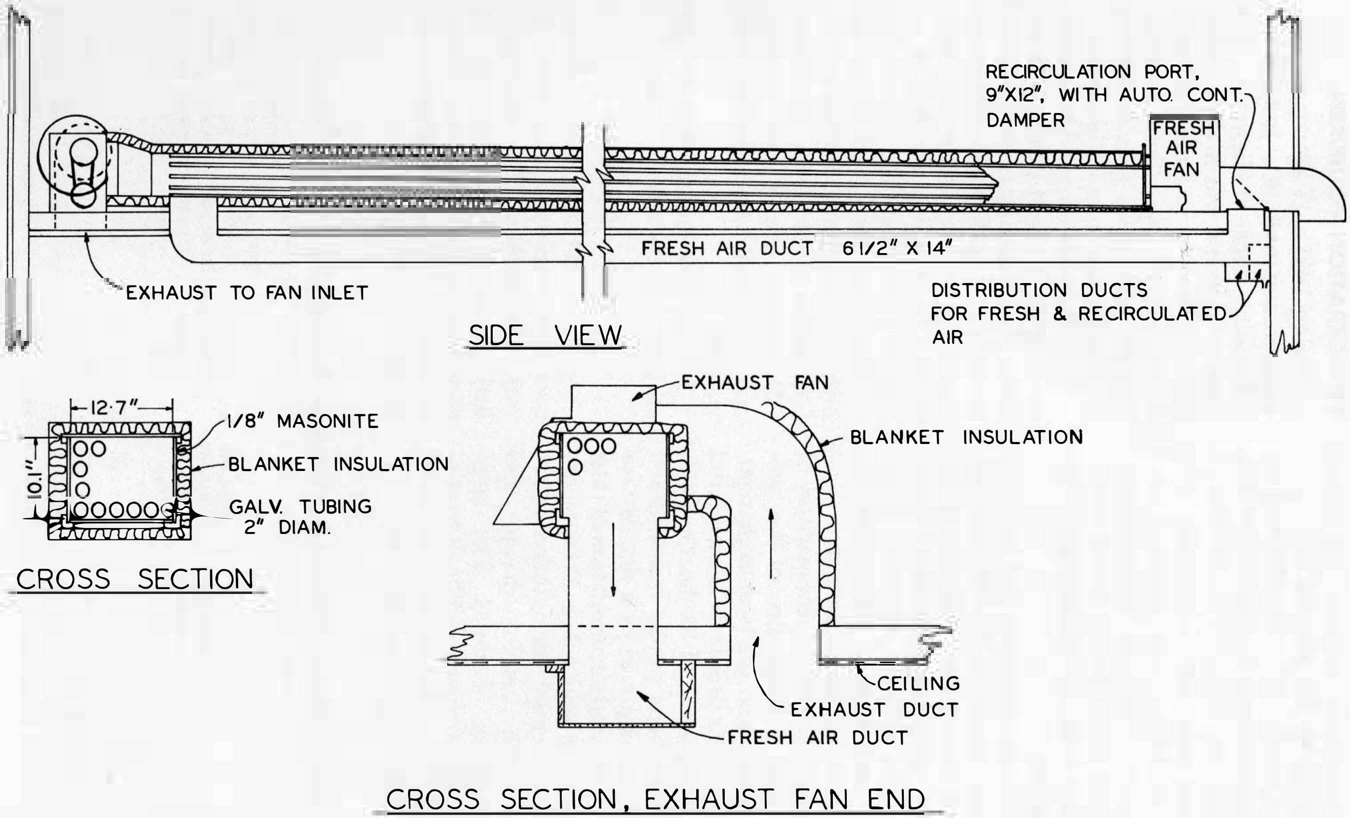


Figure 1. Plans for the heat exchanger installed in the poultry laying house at the Central Substation, Highmore.

approximately one-half the heat available in the exhaust air.

On January 22, 1954, the heat exchanger was put into operation. Fan speeds were adjusted to 600 cfm on exhaust and 500 cfm intake. The heat exchanger was operated until March 15, 1954, but the inadequate delivery of air during warm weather forced the discontinuance of the operation.

The unit was again put in operation on December 17, 1954, and operated during the remainder of the winter. The temperatures of the intake and exhaust air were measured continuously throughout the operation of the exchanger.

Table 1 shows a comparison of the effect of ventilation of the two-fan system and the heat exchanger.

The major problems encountered in the operation of the heat exchanger have been power consumption (approximately 18 kwh per day for 220 birds) and accumulation of dust in the fans and tubes. Power consumption could be reduced somewhat with careful design and streamlining of ducts. The large quantities of dust are difficult to control as filters clog rapidly and need daily cleaning or replacement.

REFRIGERATION EQUIPMENT

During 1955 a 1½ horsepower air conditioning unit was installed in the poultry house at Highmore. The primary purpose was to dehumidify and add heat to air from the house during winter operation. In addition the unit would provide limited cooling during summer operation.

The unit was designed to operate as an air to air heat pump with the heating and cooling cycles obtained by directing air flow over either the condenser or evaporator coils rather than by reversing the flow of refrigerant. Basically the system was designed for three methods of operation: (1) dehumidifying and recirculation of air from the house; (2) cooling during hot weather; and (3) ventilation during mild weather.

The air flow patterns for the unit are shown in figures 2, 3, and 4.

Winter Operation

Air from the house was drawn over the evaporator coil where it was dehumidified. The dehumidified air was then passed over the condenser coil where it was reheated and was then recirculated in the poultry house (figure 2).

Table 1. Effect of Ventilating System on Temperature in Poultry House

Type of ventilating system	Number of readings averaged*	Average minimum outside temp.	Average minimum inside temp.	Difference between inside and outside temp.	Number of days temp. inside less than 32° F.	% of days temp. inside less than 32° F.
2-fan system	16	0° F.	52° F.	25	14	87
Heat exchanger system	45	0.5° F.	31° F.	30	20	44

*Only days on which outside temperature was between —15 and +10° F. are included.

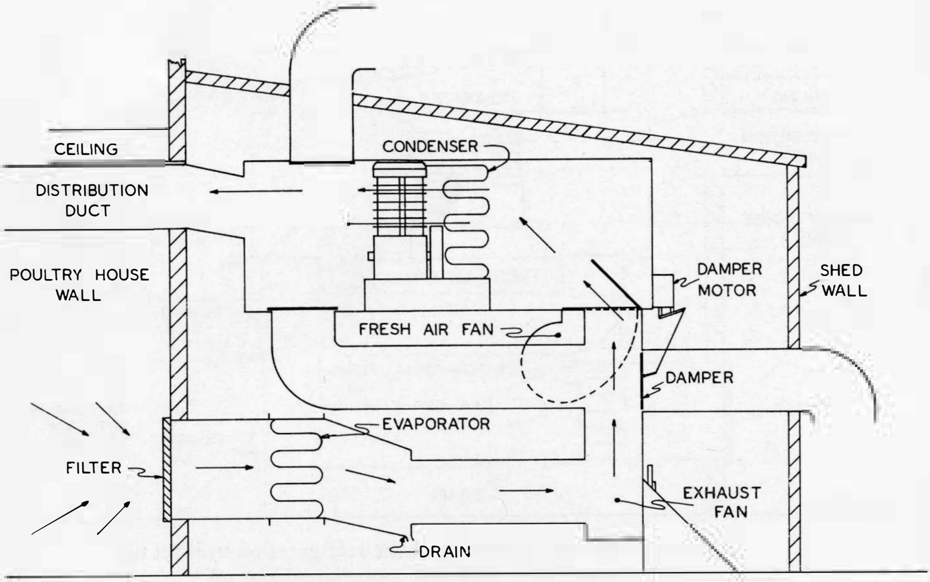


Figure 2. Schematic drawing of the refrigeration unit showing the air flow pattern for winter operation.

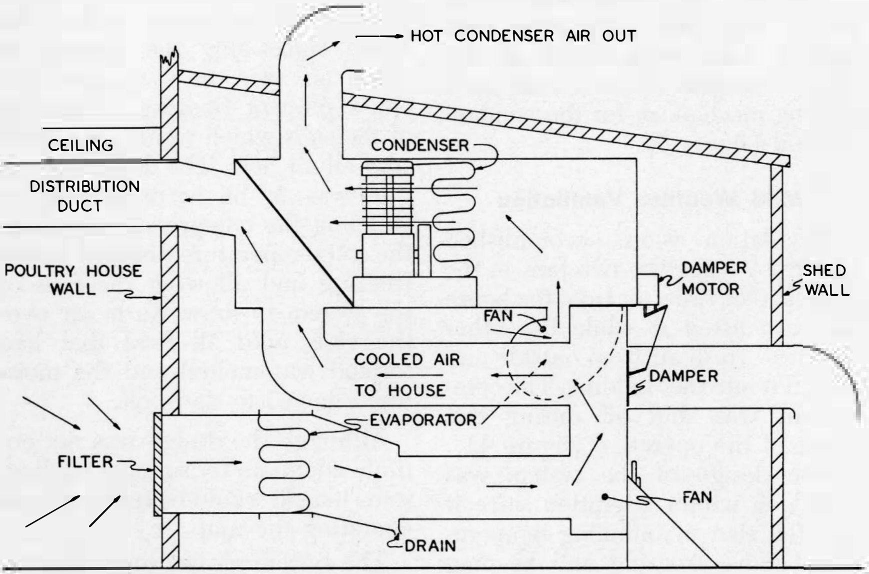


Figure 3. Schematic drawing of the refrigeration unit showing the air flow pattern for summer operation.

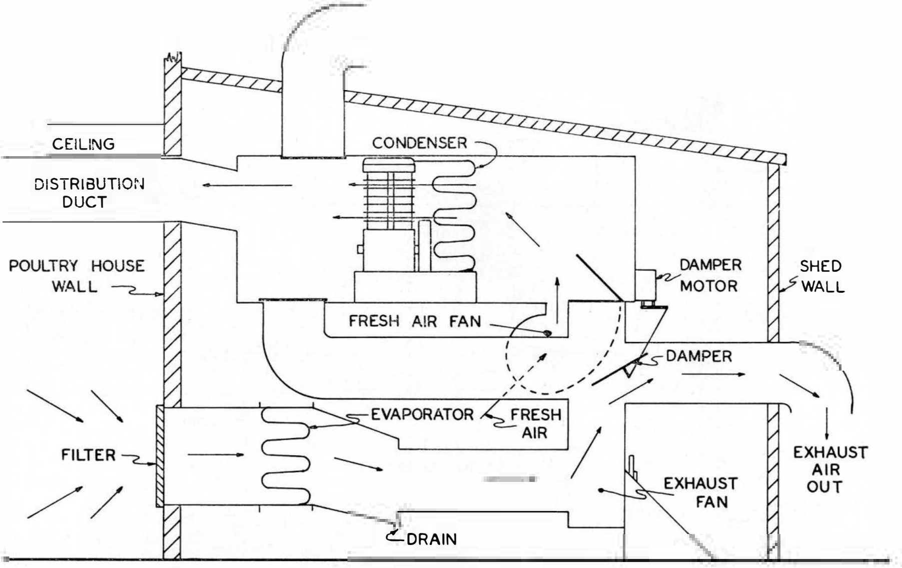


Figure 4. Schematic drawing of the refrigeration unit set up for ventilation with the compressor shut off.

Summer Cooling

Air from the poultry house was drawn over the evaporator coil, cooled, and recirculated in the house. Outside air was used as a cooling mechanism for the condenser coil (figure 3).

Mild Weather Ventilation

Ventilation was accomplished simply by using the two fans in the system. One drew air from the house and exhausted it while the other fan drew fresh air from outside and forced it into the building. The compressor was shut off during this phase of the operation (figure 4).

The design of the system was based on winter operation since it was felt that maintaining temperatures above freezing and keeping the relative humidity within comfortable limits were of greatest importance. Satisfactory operation depended upon maintaining above freezing temperatures within the house since only the passage of warm air over the evaporator coil could prevent freezing or frosting of the coils which would ultimately stop all air flow. The defrosting action was to be accomplished by stopping the compressor whenever the coil temperature dropped below freezing and allowing the fans of the system to draw warm air over the coils until all frost that had formed was melted and the moisture allowed to drain off.

Although the design was not entirely adequate for summer cooling, some benefit would be realized from operating the unit.

The system proved to be unsatisfactory for winter operation due to difficulty in preventing frost from accumulating on the evaporator coils. The main shortcoming of the unit was in design of the system since it was impossible to maintain coil surface temperatures above freezing and still have the unit operating sufficiently long periods of time to function as a dehumidifier. Data were not recorded during operation of the unit since it was impossible to stabilize conditions for an appreciable length of time.

In addition to the problems encountered during winter operation, use during summer also proved to be inadequate. The unit did not maintain satisfactorily low house temperatures to warrant its operation. Almost constant adjustment and maintenance of the unit were required to keep it in operation which did not prove practical. Dust accumulation still proved to be a problem although filtered intake areas were increased four times in an attempt to obtain adequately clean air.

Although the installation proved unsatisfactory for obtaining data, the information gained here proved invaluable in the design of an air to air heat pump at the Agriculture Experiment Station at Brookings.

HEAT PUMP SYSTEM

This phase of the study was carried on at the State College campus in Brookings. The building used was a windowless caged layer house, housing approximately 1,000 birds. This house was of frame construction 32 feet wide by 52 feet long with the long axis of the building running north and south. It was well insulated and had previously contained a thermostatically controlled fan ventilation system which had worked very well in cold weather but was not adequate during hot weather because of the high bird density in the house. This was in contrast to the condition at Highmore where the bird density had been very low and winter operation was not satisfactory.

Design of the Installation

A poultry house heat pump installation must be designed to use the heat produced by the birds efficiently while maintaining a healthy comfortable environment in the house. The installation should also be fairly easy to make with readily available equipment and materials. In this study, it was decided to use a standard air to air unit with two coils although systems using three coils have been proposed by various investigators including Cloud (4). Since laying hens produce most efficiently between temperatures of about 55 and 70°F., these temperatures were the minimum and maximum inside design temperatures. The humidity inside the building should be low enough to prevent damp conditions but high enough to prevent excessive dust in the building. The commonly accepted value of 80% relative humidity recommended by Cloud (4) was used as the inside design relative humidity in this study. Outside design conditions for this area are taken from the *"Heating, Ventilating, and Air Conditioning Guide"* (2). They are as follows:

Summer 95°F. and 40% rel. hum.
Winter -20°F. and 90% rel. hum.

It was decided that the system be designed to house 700 birds in the summer and 1,000 birds in the winter. During previous summers it had been necessary to reduce the number of birds housed to under 500 to prevent death from heat prostration.

The amount of ventilation air necessary in the design was somewhat uncertain since its main purposes are to remove moisture and objectionable odors. Some investigators, including Cloud (4), have used only infiltration air in their calculations but this has proven impractical in most cases because of the build-up of ammonia and other odors. Therefore, for the design in this study low ventilation rates of about one-half cubic foot of air per bird per minute were used. This added considerably to the cooling and heating loads. The ventilation rate could be increased during the test period if it became necessary to do so. Since the cooling and heating would be done primarily on recirculated air, this air flow rate would be quite high. Therefore in the design, an assumed flow rate of 2,100 cfm (slightly over a cfm per bird) was used for the total flow rate through the indoor coil.

The calculated cooling load was 55,336 British thermal units per hour (Btu/hr) or 4.6 tons of refrigeration. Of this amount 35,216 Btu/hr was sensible heat and 20,120 Btu/hr was latent heat indicating that the unit must do considerable dehumidifying as well as cooling. This is quite practical since the calculated temperature drop is only about 16 degrees in the evaporator with an assumed air flow of 2,100 cubic feet per minute. The calculated winter heat load was 49,380 Btu's per hour. This total would have been smaller except that it was necessary to raise the amount of ventilation air to 600 cubic feet per minute to keep a moisture balance between moisture produced by the birds and moisture removed by ventilation. The temperature rise of the air through the condenser would be from 55 to 77°F. or 22 degrees which is a practical value. If the ventilation air is passed through the condenser before going into the house, conditions on this side of the heat pump would be still more favorable. Since the heat obtainable from the minus 20 degree outside air is a small quantity, the bulk of the heat must come from the inside air leaving the building. This air can be drawn through the evaporator to the outside thereby giving up most of its heat. If the temperature of this air would drop from 55 to 10°F., adequate heat could be obtained. It was decided to include an electric heating element in the design for extreme conditions however, since it was doubtful that a 45 degree temperature drop could be obtained in the evaporator. It also became evident that the heat pump selected would have to have an automatic defrost mechanism since considerable frost would form on the evaporator.

After considering various makes and models of heat pump units, it was decided to use two units—a 3 horsepower heat pump with supplemental electric heat, and an ordinary 3 horsepower cooling unit.

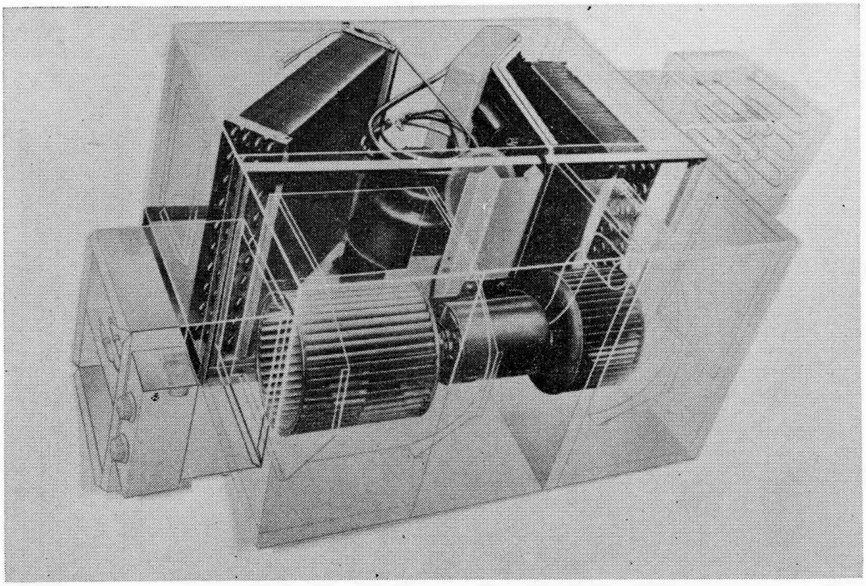


Figure 5. Phantom view of the heat pump.

The overall cost using these units was considerable less than for one large central unit. A phantom view of the heat pump chosen is shown in figure 5. The cooling unit was identical except it did not have refrigerant reversing valves and a defrost mechanism.

The method of using the two units was to operate both of them when cooling was required but to use only the heat pump when heating was required. At times when the heating load was more than the heat pump could handle, the electric resistance heat would cut in.

An overall plan of the installation is shown in figure 6. The bulk of the installation was in the attic with only the distribution ducts, filters, and thermostats in the house. During cooling conditions both units obtain inside air from their respective filter areas and pass this air through their evaporator coils into the evaporator discharge ducts. From these ducts the air enters the cross ducts and moves toward the sides of the building and down into the distribution ducts running the length of the building where it is metered into the building through a one-half inch horizontal slot in each distribution duct.

Figure 7 shows the filter bank and how the filters are removed to be cleaned. There are three 20 by 25 inch filters in each bank giving a total filter area of 1,500 square inches which is nearly five times the manufacturer's recommended area for residential and commercial applications. This large filter area was considered necessary because of

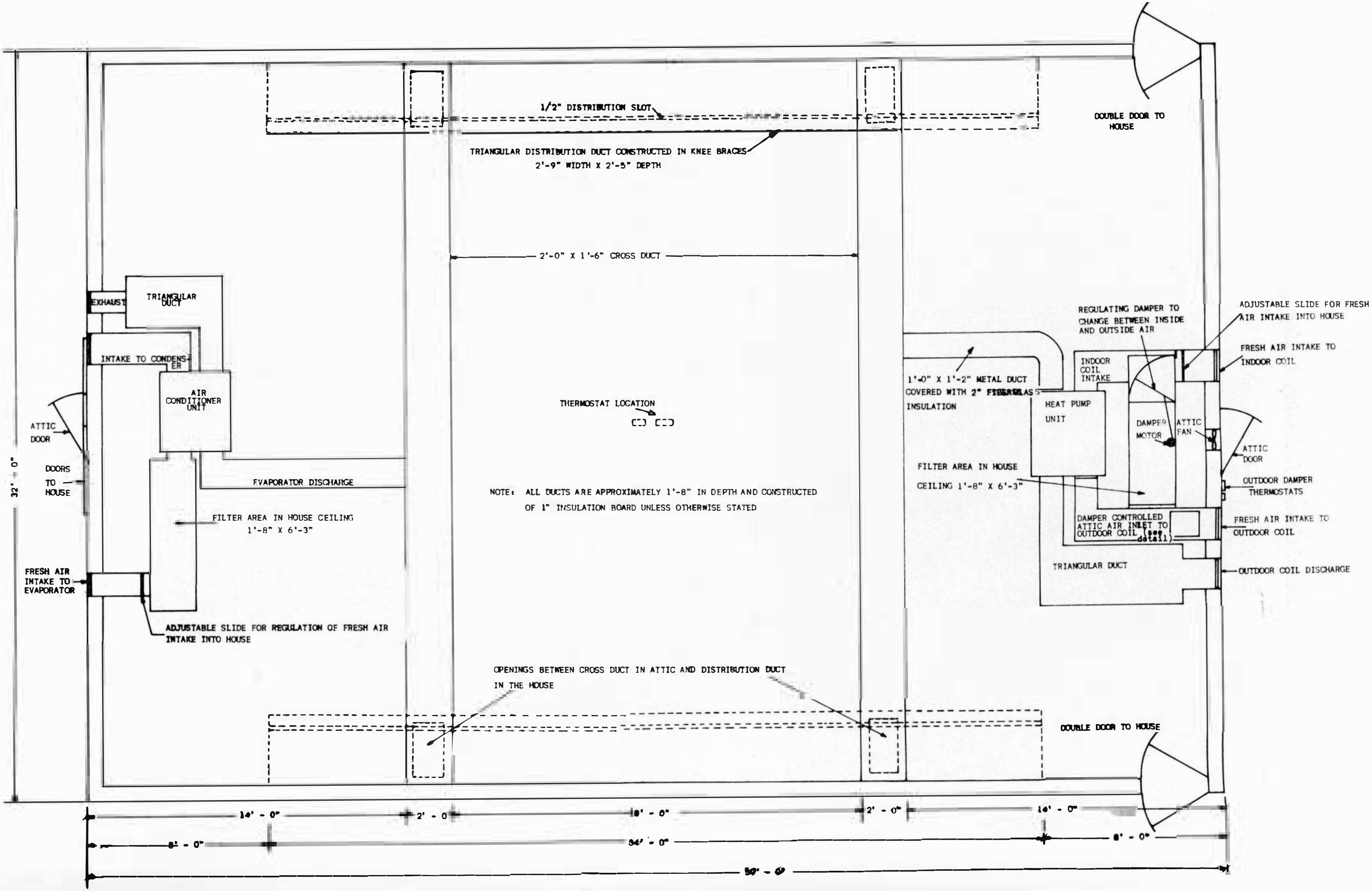


Figure 6. Layout of the heat pump system in the poultry house.

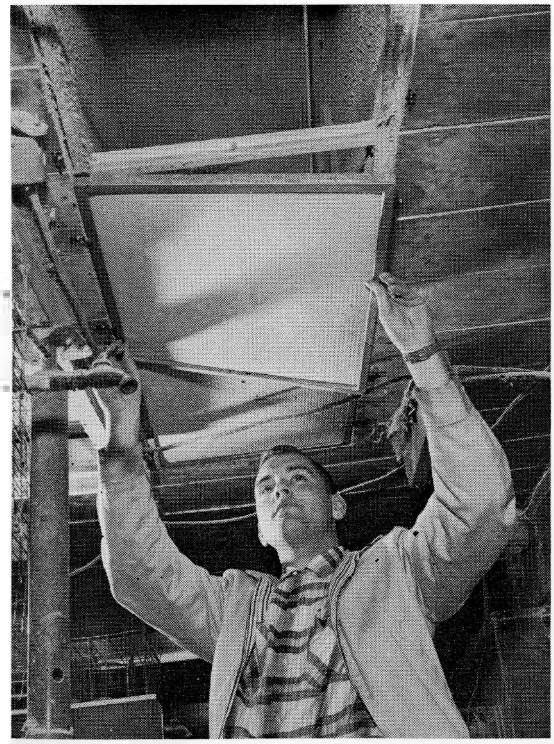


Figure 7. Removal of filter from filter bank for cleaning.

Figure 8. Filter bank with all filters removed.

the extremely dusty condition in poultry houses. Figure 8 is somewhat the same view as figure 7 except all the filters are removed. The damper system for changing between fresh air and inside air going to the indoor coil is clearly shown. With the damper in the position shown, predominately fresh air is drawn into the house. With the damper in the alternate position, most of the air going into the house is recirculated air. The damper is auotmatically controlled by outside thermostats so that during hot or cold temperatures the damper moves to the recirculating position while during mild weather the damper is in the position shown in figure 8.

Typical duct construction can be seen in figure 9. All ducts were constructed using 1-inch insulation board except for the duct from the heat pump indoor coil to the cross-duct which was constructed of metal and covered with fiberglass insulation. This was necessary because of the high temperatures when operating the auxiliary resistance heating elements. Existing braces in the attic were used as a main frame for the insulation board ducts where possible. These ducts were oversized from one and one-half to two times the manufacturer's recommendations to allow for the relatively rough sides.

Figure 10 shows one of the distribution ducts in the house. This

Figure 9. Typical duct construction.

duct was built using the existing knee braces as a frame giving a duct cross-sectional area of 3.4 square feet. With this large area, the duct acts as a plenum with a static pressure maintained inside and the air is forced out through the slot. Since these ducts are on opposite sides of the building a uniform distribution of incoming air should be maintained throughout the building. Another advantage of putting these ducts in the location shown was that they took no usable space out of the building.

The heat pump was mounted in the end of the building shown in figure 11. The louver at the far left is the outdoor coil exhaust, the louver next to it is the outdoor coil intake, and the louver to the right is the fresh air intake to the indoor coil. This end also contains the attic exhaust fan which is shown at the top of the picture. The fan was moved from the position covered by the lower door in the picture where it had served as the ventilation fan for the house. This fan is controlled by a thermostat in the attic so that it turns on only when the temperature is above the level set. The two thermostats shown in figure 11 control the damper previously shown in figure 8.

The main thermostats were located in the center of the house as shown in figure 6. It was possible with these thermostats to have the fans in constant operation regardless of heating or cooling demands giving good air movement and constant ventilation rates.

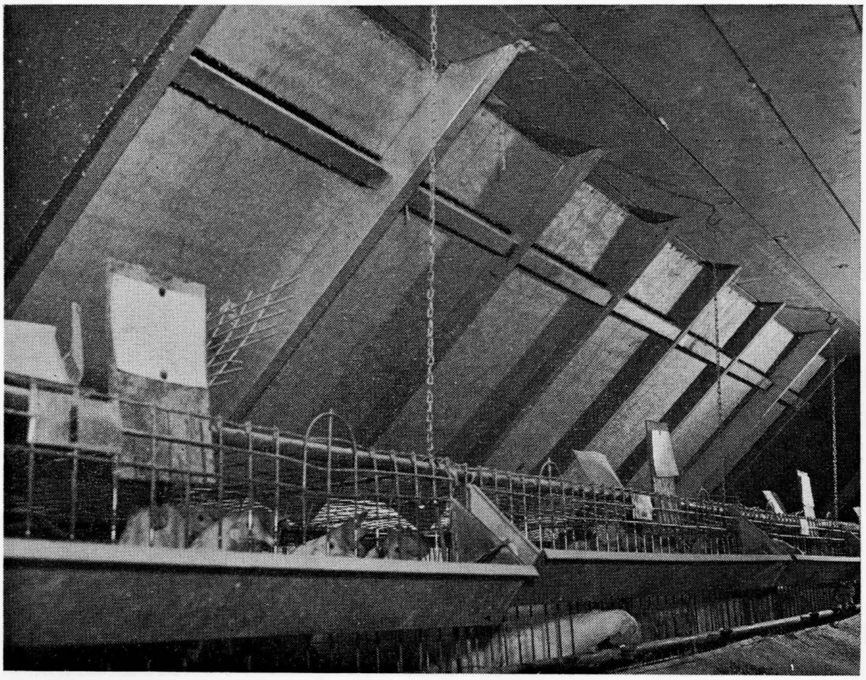
As the system was put into operation, minor changes were made in the design in an attempt to gain simpler and more efficient operation. Most of these changes came as a direct result of some faulty operation and therefore will be mentioned later.

Methods of Operating the System

Several methods of operating the system were used during the test period. The differences were mainly in the form of various ventilation rates and different sources of air for the respective coils.

During the first part of the test period which was from July 14, 1958, to September 1, 1958, there were 620 laying hens in the house and a constant ventilation rate of 350 cfm of outside air was used. This air entered the heat pump indoor coil (evaporator) and mixed with the recirculated indoor air. Excess air in the building left by passing from above the filter area through an adjustable opening to the outdoor coil (condenser) and finally outside. The location of most of these parts can be seen in figure 6. During the latter part of this period the old ventilation system fan

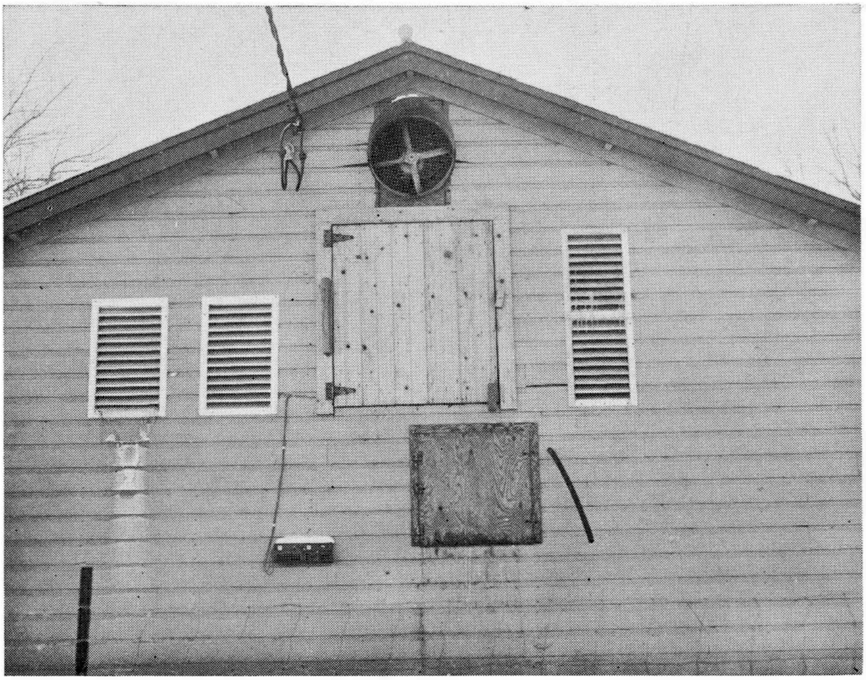
Figure 10. Distribution of ducts in the poultry house.



was used as an attic fan and set to exhaust air from the attic when the attic temperature was 85°F. or above.

From September 1 to October 1, the number of birds in the house was increased to 980 with the ventilation rate remaining the same. During October, the number of birds was decreased to 740 and the heat pump indoor coil damper system set to admit 1,000 cfm fresh air when the outside temperature was between 80 and 25°F. At other outside temperatures there was no positive ventilation. On November 1, the number of hens was increased to 980 birds. This number was held throughout the winter until April 1. Starting November 1 and throughout the rest of the winter, the outdoor coil air and indoor coil ventilation air was taken from the attic. The quantity of attic air to the outdoor coil (evaporator) was somewhat reduced so that excess inside air could leave the building through the outdoor coil and thus give much of its heat to the incoming ventilation air. During this time the heat pump was serving primarily as a heat exchanger. At various times during the winter minor changes were made in the operation of the system such as removing filters and using a clock-timed defrost cycle. These changes were made to overcome operating difficulties and will

Figure 11. Heat pump mounted in end of poultry house.



be discussed in the section covering these problems.

In the spring the ventilation air and the air to the outdoor coil were again taken from the outside rather than from the attic. The exact dates were March 11 for ventilation air and April 11 for the outdoor coil air. The number of birds was increased to 1,180 on April 1 and reduced to 1,100 on May 1 for the remainder of the test. The ventilation rate was increased to approximately 2,000 cfm on April 24 by using the air conditioner evaporator fan to bring in fresh air along with the heat pump indoor coil fan. This high ventilation rate made changing filters unnecessary since neither unit was recirculating any large amount of air. The 2,000 cfm ventilation rate was maintained to the end of the test period on May 31.

Problems in Equipment Operation

The overall operation of the system was satisfactory but several serious difficulties that needed correction did become evident.

The most serious deficiency of the system was the inability to obtain clean air from inside the house. Although the filter area for each unit was extremely large, it still became saturated with the dust rapidly and restricted the air flow into the indoor coil section of the units. It was necessary to clean these filters about every 4 or 5 hours under extreme conditions where the units were in constant operation. Although dust was considered to be a problem under conventional housing conditions, these extreme conditions were not expected because the house contained only confined cage layers which could not stir up litter and feed with their feet. To further complicate matters, when the filters became clogged, there was not sufficient air flow through the coils to keep them from freezing. The heat pump indoor coil (evaporator) froze several times during cooling and the outdoor coil froze several times during heating due to clogged filters.

The freezing of the outdoor coil was not as serious as the indoor coil freezing since the unit had an automatic defrost cycle for this coil. However, during extremely cold outside temperatures, the defrost of the outdoor coil (evaporator) failed because its temperature would not rise high enough to satisfy a defrost thermostat attached to the coil which would again turn on the unit. In these cases the unit remained off and the temperature in the house rose because almost no ventilation air entered the house and the heat produced by the birds was greater than the heat loss from the building. With below zero readings outside, the inside temperature would rise to as high as 80° F. during coil freeze-ups.

The faulty defrost was partly due to wind forcing some outside air to pass through the unit even though the fans shut off during defrost. This prevented adequate warm-up of the outdoor coil. Adequate warm-up was also prevented because the air at the indoor coil was too cold to be an efficient heat source. It was exposed to attic air through the ventilation air intake openings.

Several methods were used to improve defrost action. One of these was to use auxiliary heat to warm up the defrost thermostat on the outdoor coil. This did not work well because the amount of heat required changed with outside wind and temperature. Another method used was to put a time clock in the compressor circuit so that the compressor would shut off for short intervals even though the fans kept running. Since the air passing through the outdoor coil was always near 60°F. it supplied the heat for the defrost rather than reversing the cycle of the heat pump unit. Some of the intervals used were: off 10 minutes each hour; off 10 minutes every 2 hours; and off 20 minutes every 2 hours. The last setting worked quite successfully in cold weather but the other two failed to give adequate defrost action. It must be said, however, that when operating the heat pump as a heat exchanger, no defrost action at all would be necessary if a satisfactory air cleaning method was devised. This was shown during a short period when no filters were used.

From January 23, 1959, to February 5, 1959, the filters for the heat pump unit were removed and dirty air was allowed to pass through the unit. Although outside temperatures were cold during these periods, no defrost difficulties were encountered because sufficient warm air from inside passed through the outdoor coil to keep it from freezing. The air temperature entering the outdoor coil averaged a little under 60°F. since the unit took most of its air from inside the house and acted as a heat exchanger. At the end of the short period without filters, however, the outdoor coil was quite dirty and required a complete cleaning indicating that some type of air cleaning was necessary. As an attempt to eliminate some of the load on the filters, an experimental centrifugal type precleaner was constructed and added to the system. A large glass jar was to collect the heavier dirt particles as they dropped from a collector located at the circumference of a spiral passage. This centrifugal cleaner did take out the larger particles but was not very successful because the bulk of the particles were too small to be taken out. During the last portion of the test period, the filter system was again used because there was not time for further research into other air cleaning methods. It was very evident, however, that before a heat pump system could be recommended for farm use, an efficient, econical, and trouble-free filtering system should be developed.

Another minor problem that was encountered was that the damper system for the indoor coil tended to freeze in extremely cold weather. This was caused by warm moisture-laden air coming through the filters and depositing some of its moisture in the form of frost on the cold damper door causing it to freeze tight. To eliminate this problem, the door was changed to admit a constant ventilation rate of 500 cfm on January 13, 1959.

Methods of Obtaining Operation Data

During the entire test period data were obtained on the environmental

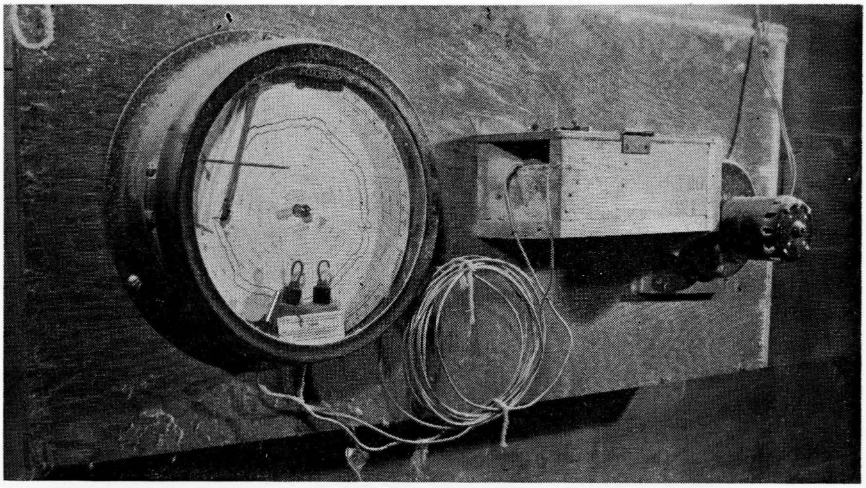


Figure 12. Inside conditions were recorded on psychrometer.

conditions inside the house, power consumption for each unit, power consumption for the supplemental resistance heat, air flow quantities, and outdoor conditions. During certain stages of the test period other records such as duct temperatures were obtained. These were used to check the efficiency of the units and to see if they were cycling properly.

At all times inside conditions were constantly recorded on a recording-psychrometer. This particular instrument was made using a two-bulb mercury-filled recording thermometer, blower, filter, and precleaner. Figure 12 shows the finished psychrometer which worked very well during the 10½ month test period. The air is drawn through a precleaner at the top (where large dirt particles are removed) and then down into the filter box. A 20 by 20 inch permanent type filter in the box removed most of the remaining dirt particles before the air entered the blower. The blower then forced the air over the wet and dry bulbs located in the small box attached to the front of the filter box. The recording portion of the thermometer can be seen mounted at the left side in figure 12.

The recording psychrometer differs from most other instruments of this type in that the air passes through the blower before passing over the bulbs. Most other units have the bulbs on the suction side to eliminate the effect of a temperature rise as the air passes through the blower. However, in this case, the small amount of heat picked up in the blower had a negligible effect on the recorded temperature, as was shown by calibration against a mercury-in-glass thermometer. Units with the bulbs on the suction side had been used previously and had performed very poorly because the wet wick became extremely dirty in a short time. Spot checks during the

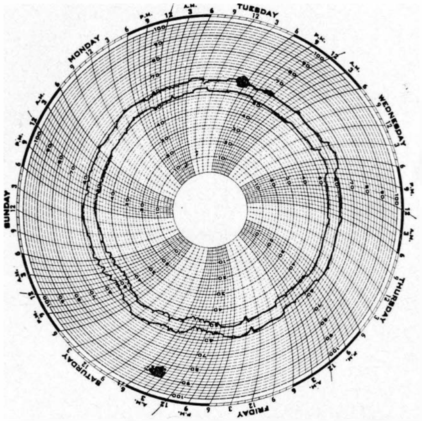


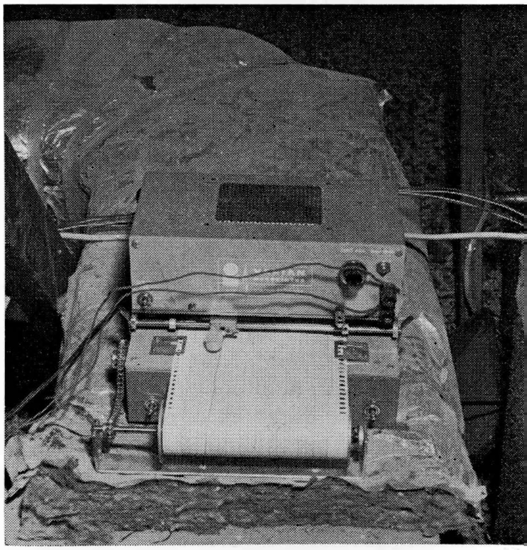
Figure 13. Typical chart from recording thermometer.

test period showed that no recalibrating was necessary. A typical chart from the recording thermometer is shown in figure 13. This chart was made during January when the dry bulb temperature averaged around 60°F. The inside line is the wet bulb temperature and averaged around 8 or 9 degrees less than the dry bulb temperature. Each chart of this type was for 1 week and by changing charts regularly a continuous record was obtained. Separate power consumption readings for each unit and the electric resistance heat were obtained by metering them separately. Meter readings were taken at approximately 1-week intervals so that the average rate of power consumption for various dates and outside temperatures could be obtained.

Outside weather data were obtained from the Weather Engineering Department South Dakota State College. From these charts it was possible to tabulate average, maximum, and minimum temperatures for any desired interval.

Duct conditions were obtained by several methods. Figure 14 shows an arrangement used for recording temperatures in the duct leading from the heat pump into the poultry house. By observing this chart it was possible to tell how often the unit was cycling, whether it was cycling properly, and what temperatures were being obtained during various types of operation. One thermopile, consisting of two thermocouples in series, was placed in the duct and a reference thermopile was placed about 3 feet into the soil at the center of the poultry house floor. A difference in temperature between these points produced a voltage which was proportional

Figure 14. Arrangement used for recording temperatures in duct leading from heat pump.

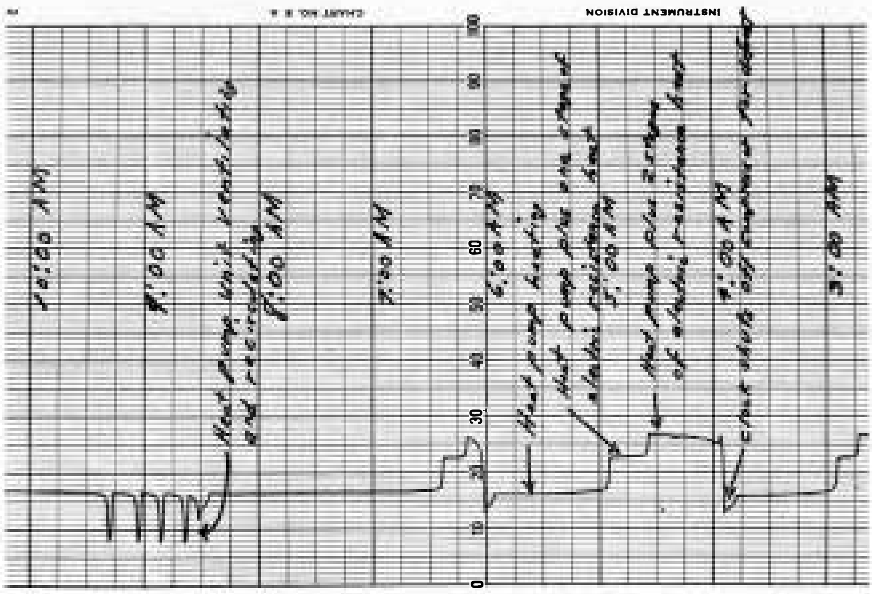


to the temperature differential and this was recorded on the recording instrument. Since the soil temperature in the poultry house floor remained constant, the instrument was calibrated to read in degrees Fahrenheit. A typical section of chart which has been labeled to show the various operations is shown in figure 15. Wet and dry bulb temperatures in the ducts at various times were obtained by inserting wet and dry bulbs into them. Air movement in the duct was rapid enough to obtain wet bulb readings. The air velocities and resulting volumes at these times were obtained by inserting a hot wire probe into the duct at various spots and averaging the readings.

Conditions Inside the Poultry House

The data have been put into approximately weekly averages in table 2. Here the power consumption has been changed to an average rate for each component and an average rate for the entire system is also given. In this table, it is quite easy to compare variables such as the power consumption rate with conditions such as average inside temperature, outside temperature, relative humidity, or any minimum or maximum value. This data has also been segregated and plotted in figures 16 through 21. Figure 16 indicates how the average inside and outside temperatures varied during the test period. It is important to note that during the coldest winter months of December, January, and February the outside temperature did drop excessively but the average inside temperature dropped only a few degrees from its overall average. This was shown even more clearly in figure 17 when average mini-

Figure 15. Typical section of chart showing various operations.



NOTE: The actual temperature equals 2.2 x the reading on the chart.

Table 2. Poultry House Data Summary (Approximate Weekly Intervals) for the Test Period July 15, 1958-May 31, 1959

Interval	Av. temp. inside, ° F.	Av. rel. hu. inside, %	Av. max. inside temp., ° F.	Av. min. inside temp., ° F.	Av. temp. outside, ° F.	Av. max. outside temp., ° F.	Av. min. outside temp., ° F.	Power consumption rate, kilowatts: Air cond. unit	Heat pump unit	Elec. res. heat	Total
Jul. 19-26 (167 hrs.)	71	66	75	68	70	83	59	2.72	4.20	0.00	6.92
July. 26-Aug. 2 (194 hrs.)	71	70	75	69	70	84	59	2.52	4.24	0.00	6.76
Aug. 2-9 (167 hrs.)	74	68	80	69	78	95	62	3.85	3.78	0.00	7.63
Aug. 9-16 (167 hrs.)	74	64	80	70	76	89	63	3.60	4.59	0.00	8.19
Aug. 16-23 (169 hrs.)	73	63	75	68	69	83	57	2.89	4.30	0.00	7.19
Aug. 23-30 (166 hrs.)	74	64	80	70	67	83	53	3.36	3.38	0.00	6.74
Aug. 30-Sept. 6 (168 hrs.)	72	64	76	68	64	76	52	2.92	3.28	0.00	6.25
Sept. 6-13 (168 hrs.)	70	68	72	68	65	79	51	2.49	3.58	0.00	6.07
Sept. 13-20 (169 hrs.)	72	71	76	69	60	75	51	3.06	2.32	0.00	5.38
Sept. 20-27 (166 hrs.)	71	63	74	65	70	75	49	2.83	3.04	0.00	5.87
Sept. 27-Oct. 4 (169 hrs.)	66	59	71	64	53	65	40	1.61	1.69	0.00	3.30
Oct. 4-13 (215 hrs.)	64	60	67	61	53	67	38	1.35	1.63	0.00	2.98
Oct. 13-18 (120 hrs.)	66	67	70	63	61	79	46	2.12	1.88	0.00	4.00
Oct. 18-25 (169 hrs.)	65	57	67	62	52	61	37	1.03	1.46	0.00	2.49
Oct. 25-Nov. 1 (166 hrs.)	63	57	66	61	43	55	30	0.37	1.54	0.00	1.91
Nov. 1-8 (169 hrs.)	65	53	67	60	44	57	27	0.26	1.47	0.00	1.73

Table 2. Poultry House Data Summary (Approximate Weekly Intervals) for the Test Period July 15, 1958-May 31, 1959 (continued)

Interval	Av. temp. in-side, ° F.	Av. rel. hu. in-side, %	Av. max. inside temp., ° F.	Av. min. inside temp., ° F.	Av. temp. out-side, ° F.	Av. max. outside temp., ° F.	Av. min. outside temp., ° F.	Power consumption rate, kilowatts: Air cond. unit	Heat pump unit	Elec. res. heat	Total
Nov. 8-15 (168 hrs.)	65	52	68	62	41	54	30	1.00	1.46	0.00	2.46
Nov. 15-22 (168 hrs.)	64	62	68	61	36	43	30	0.42	1.30	0.00	1.72
Nov. 22-29 (168 hrs.)	61	64	67	54	19	31	09	0.10	0.78	0.00	0.88
Nov. 29-Dec. 6 (168 hrs.)	60	58	64	57	19	26	08	0.00	0.98	2.79	3.77
Dec. 6-13 (171 hrs.)	64	66	72	58	01	18	–07	0.00	0.85	4.70	5.55
Dec. 13-20 (168 hrs.)	64	64	71	58	15	25	04	0.00	2.06	1.73	3.79
Dec. 20-29 (216 hrs.)	65	65	68	62	30	35	17	0.00	0.79	1.73	2.52
Dec. 29-Jan. 6 (192 hrs.)	63	65	68	56	07	15	–02	0.00	1.30	1.80	3.10
Jan. 6-13 (168 hrs.)	60	68	67	56	19	27	13	0.00	1.88	0.00	1.88
Jan. 13-19 (168 hrs.)	62	69	68	59	14	22	02	0.00	1.72	0.41	2.13
Jan. 19-27 (192 hrs.)	59	57	63	57	67	13	–05	0.00	2.26	1.50	3.76
Jan. 27-Feb. 3 (144 hrs.)	58	57	62	56	10	21	–01	0.00	2.30	0.68	2.98
Feb. 3-11 (191 hrs.)	59	58	62	57	07	16	–10	0.00	1.95	1.66	3.61
Feb. 11-17 (143 hrs.)	60	70	62	59	15	26	02	0.00	2.15	2.08	4.23
Feb. 17-24 (168 hrs.)	61	68	64	59	10	21	–12	0.00	1.93	3.11	5.04
Feb. 24-Mar. 4 (191 hrs.)	63	57	67	61	27	37	17	0.00	1.57	0.89	2.46

Table 2. Poultry House Data Summary (Approximate Weekly Intervals) for the Test Period July 15, 1958-May 31, 1959 (concluded)

Interval	Av. temp. inside, ° F.	Av. rel.hu. inside, %	Av. max. inside temp., ° F.	Av. min. inside temp., ° F.	Av. temp. outside, ° F.	Av. max. outside temp., ° F.	Av. min. outside temp., ° F.	Power consumption rate, kilowatts			
								Air cond. unit	Heat pump unit	Elec. res. heat	Total
Mar. 4-11 (166 hrs.)	65	60	69	63	30	37	23	0.00	1.52	0.71	2.23
Mar. 11-18 (170 hrs.)	63	60	67	62	31	38	23	0.00	1.62	0.12	1.74
Mar. 18-26 (192 hrs.)	66	63	71	63	37	52	26	0.00	2.22	0.03	2.25
Mar. 26-Apr. 1 (144 hrs.)	66	61	70	64	39	48	28	0.12	2.27	0.00	2.39
Apr. 1-8 (170 hrs.)	66	59	70	62	48	64	35	1.35	1.85	0.00	3.20
Apr. 8-15 (166 hrs.)	70	66	74	67	36	48	25	0.81	2.22	0.00	3.03
Apr. 15-22 (168 hrs.)	74	62	77	69	41	51	34	1.41	1.88	0.00	3.29
Apr. 22-29 (173 hrs.)	67	63	72	64	48	60	36	1.67	2.05	0.00	3.72
Apr. 29-May 7 (194 hrs.)	73	72	78	69	63	76	51	2.82	3.52	0.00	6.34
May 7-14 (168 hrs.)	69	72	71	65	50	60	42	1.50	2.63	0.00	4.13
May 14-21 (168 hrs.)	71	63	76	66	57	69	43	2.20	3.16	0.00	5.36
May 21-28 (168 hrs.)	72	67	76	68	56	67	46	2.62	3.40	0.00	6.02

mum inside and outside temperatures are plotted against dates during the year. Here the distance between the two plotted lines is greater because the outside temperature during a day varied much more than the temperature inside the poultry house. Although the average minimum outside temperature dropped as low as -12°F., the average minimum inside temperature dropped only to 54°F. This temperature also occurred early in the winter when the electric resistance heat was not operating properly. The lowest average minimum outside temperature occurred later in the winter and the average minimum inside temperature only dropped to 63°F. on this occasion. Since the desired tempera-

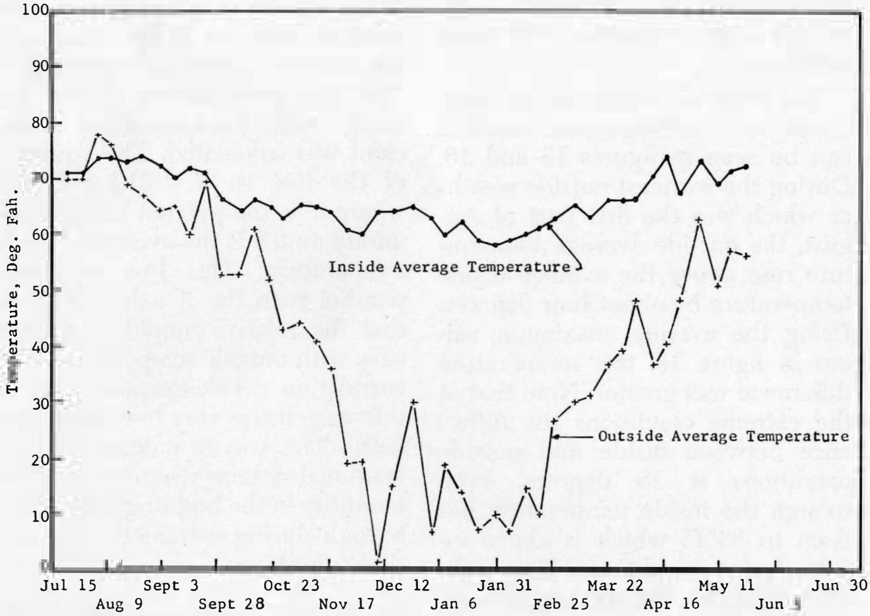


Figure 16. Comparison of weekly averages of inside and outside average temperatures, July 15, 1958-June 30, 1959.

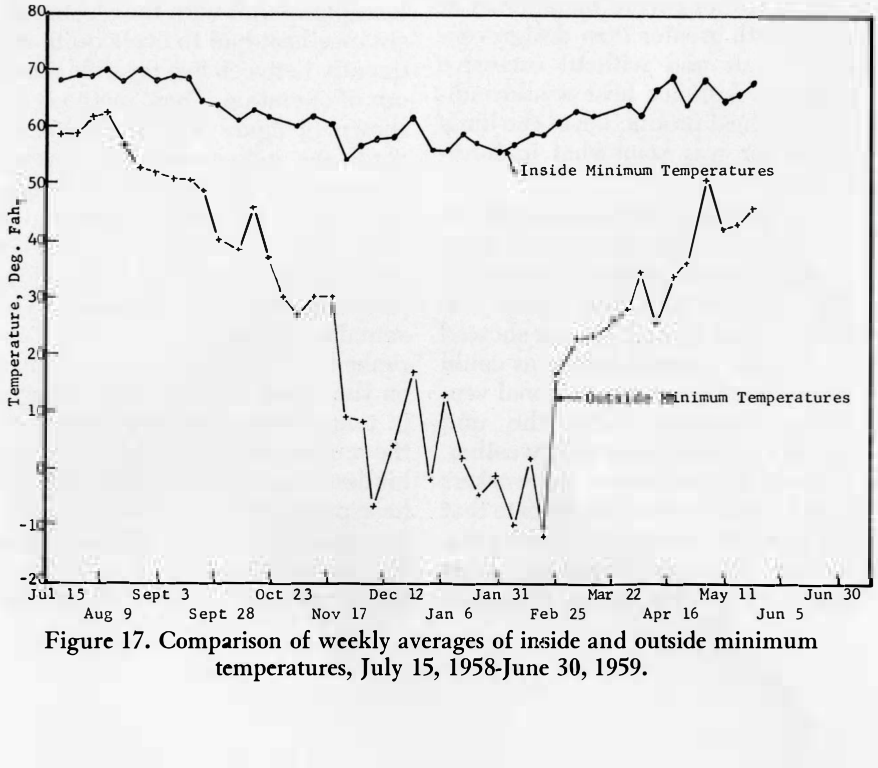


Figure 17. Comparison of weekly averages of inside and outside minimum temperatures, July 15, 1958-June 30, 1959.

ture range was from 55 to 70°F., the overall winter operation was quite satisfactory.

The effect of the system on inside temperature during warm weather can be seen in figures 16 and 18. During the warmest outside weather which was the first part of August, the outside average temperature rose above the average inside temperature by about four degrees. Using the average maximum values in figure 18, this temperature difference was greater. Note that at the extreme conditions the difference between inside and outside conditions is 15 degrees, even though the inside temperature has risen to 80°F. which is above inside design conditions. This indicates that the cooling capacity required was somewhat greater than that calculated. This result along with the fact that winter inside design conditions were maintained at times with greater than design ventilation air and without excessive electric resistance heat would indicate the heat production of the birds and litter was somewhat underestimated.

The average relative humidity in the house was plotted against dates during the test period in figure 19. The relative humidity varied between about 52 and 72% but showed no definite seasonal trends as could be expected from conventional ventilation systems where the unit would shut off during cold weather. General observations of the workers in the house seemed to indicate that the humidity range was quite satisfactory although somewhat drier than most poultry houses. Figure 20 shows the relationship between the relative humidity in the house and the average outside temperature. The regression formula of the line along with the correlation coefficient was calculated. The equation of the line is: $Y = 61.3 + 0.464X$ where Y is the percent relative humidity and X is the average outside temperature. This line is nearly parallel with the X axis indicating that the relative humidity did not vary with outside temperature. The correlation coefficient also showed this since it was very low being only 0.210. This was in contrast to conventional systems where the relative humidity in the building is extremely high during extremely cold outside conditions.

Operation of the Heat Pump Unit

To produce the desired conditions inside the poultry house, the heat pump unit with the electric resistance heat had to cycle quite frequently between five possible methods of operation. These methods are shown in figure 15. This chart records the indoor coil side exit air temperature for a portion of February 1, 1959. The times are tabulated along with the explanation of what was taking place when the various temperatures were recorded. The actual temperature in degrees Fahrenheit is 2.2 times the value shown on the chart. Definite step changes in temperature are shown between the various stages of operation, the highest temperature being with the heat pump and three states of electric resistance heat in operation and the lowest temperature was when the compressor was shut off and the

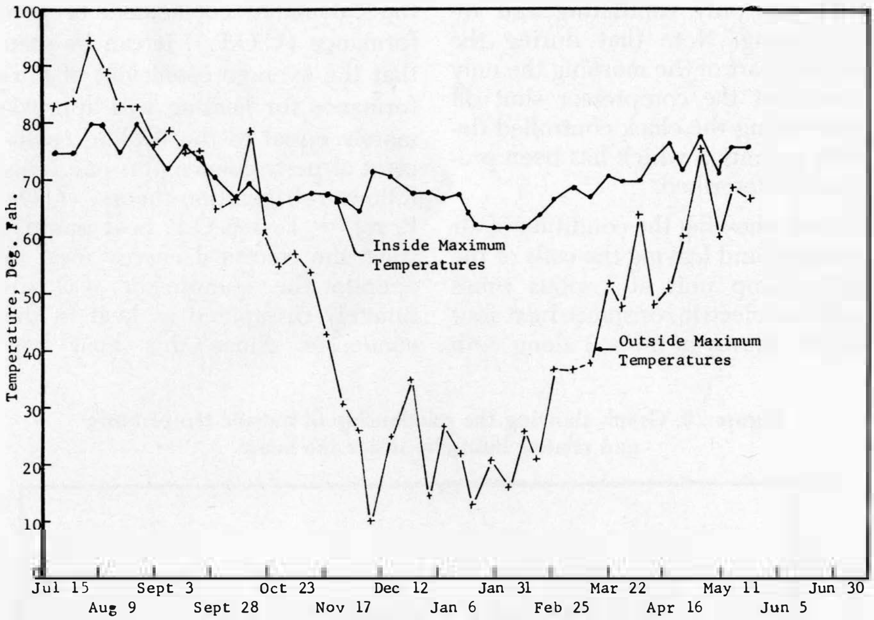


Figure 18. Comparison of weekly averages of inside and outside maximum temperatures, July 15, 1958-June 30, 1959.

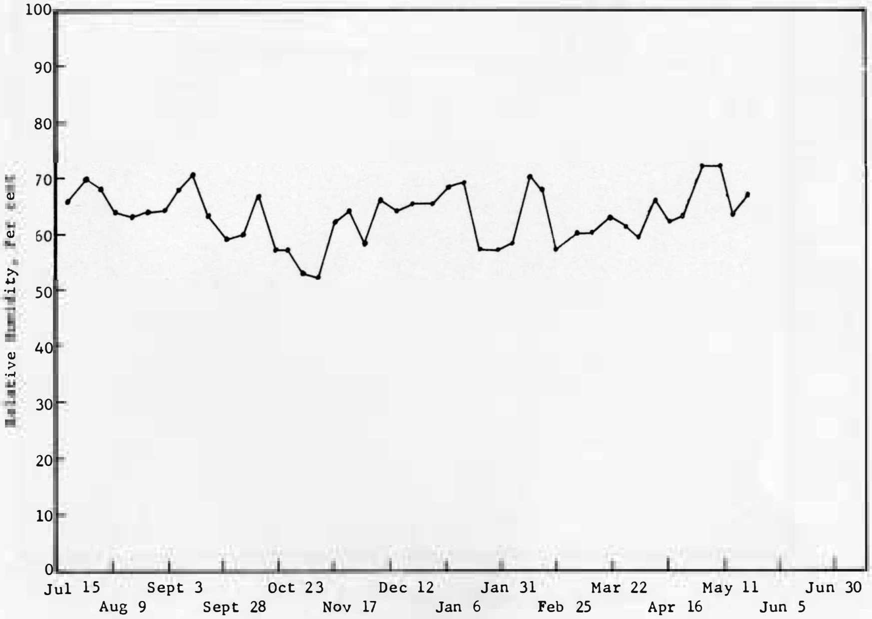
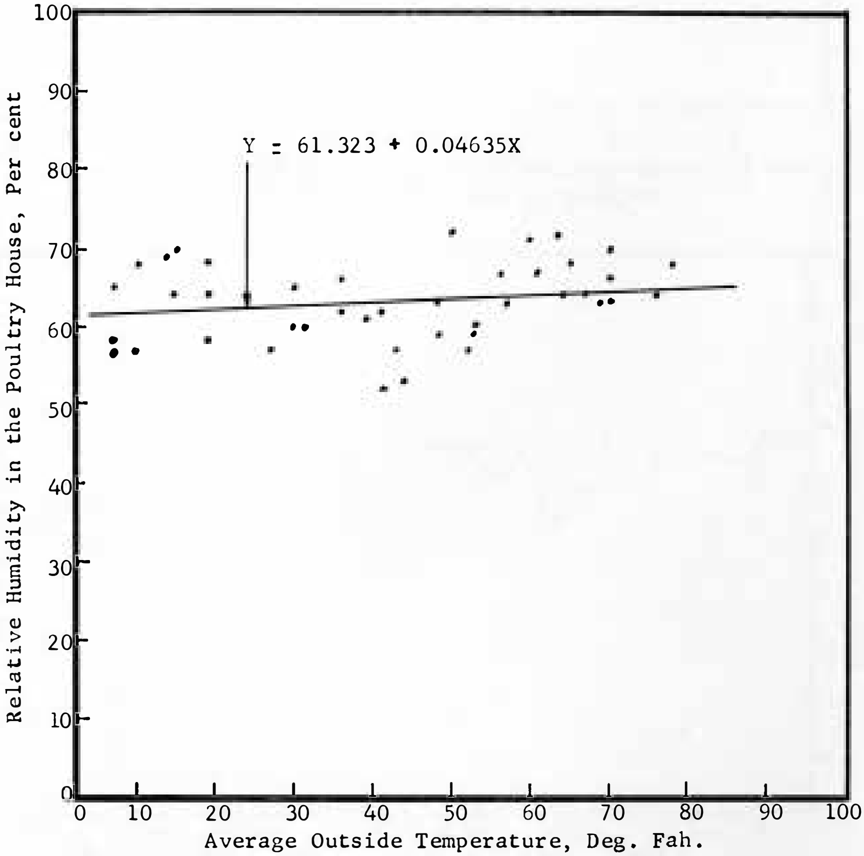


Figure 19. Average relative humidity in the poultry house during the test period, July 15, 1958-June 30, 1959.

unit was only ventilating and recirculating. Note that during the coldest part of the morning the only time that the compressor shut off was during the clock controlled defrost operation which has been previously described.

Data showing the condition of air entering and leaving the coils of the heat pump unit at various times with the electric resistance heat shut off are shown in table 3 along with the calculated coefficients of performance (C.O.P.). It can be seen that the average coefficient of performance for heating was approximately equal to the cooling coefficient of performance plus one. This follows refrigeration theory, (C.O.P. ref. + 1 = C.O.P. heat pump), since the electrical energy used to operate the compressor was ultimately dissapated as heat in the condenser. Since this heat was

Figure 20. Graph showing the relationship of outside temperature and relative humidity inside the house.



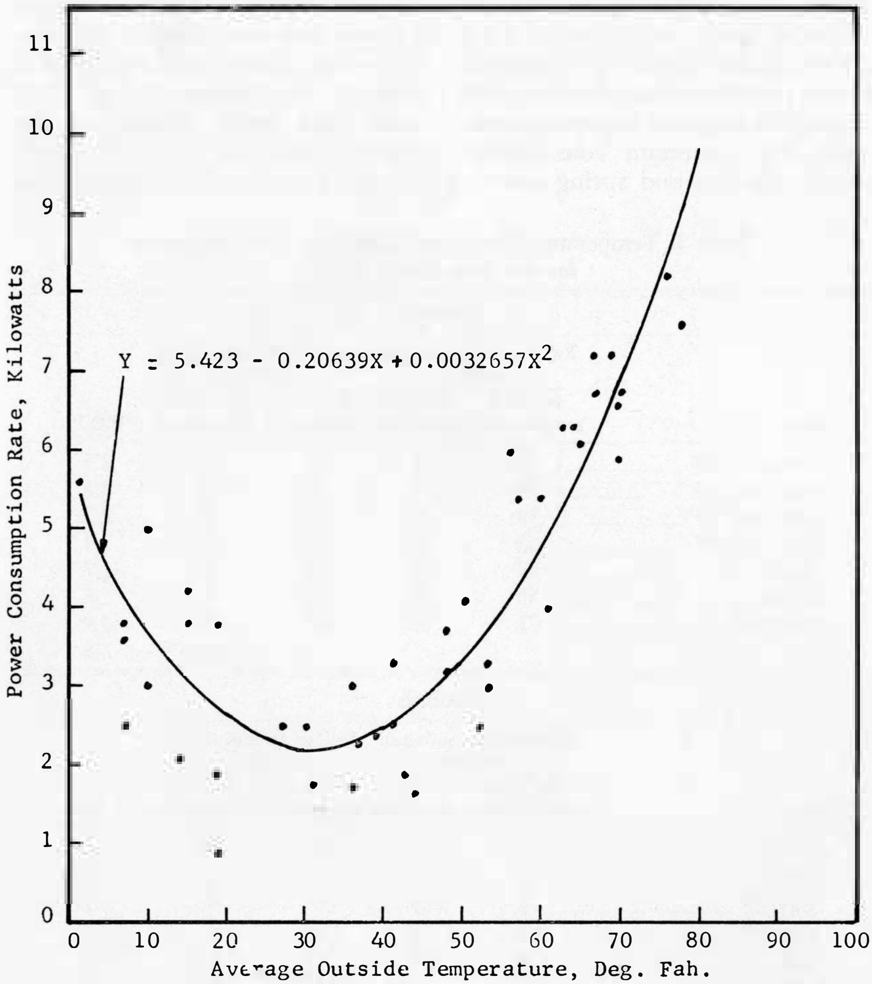


Figure 21. Power consumption of the poultry house heat pump system versus average outside temperature for maintaining inside temperatures at 60-75° F.

passed outside in summer and wasted, the coefficient of performance was lower.

The coefficients of performance in table 3 are not extremely high but do equal the values stated by the manufacturer. The air flow rates during these tests were approximately the same. However it was possible that in some cases such as on May 29 and June 1, the air flow was somewhat reduced by clogged filters or intake louvers.

Cost of Operation

The cost of owning and operating the system depends on several things, including original cost, installation cost, maintenance, and power consumption. The maximum rate of power consumption occurred during the warmest summer months while the minimum consumption was in the fall and spring months when no excessive heating or cooling was required. During the winter months power consumption was between the maximum and minimum rates partly because of the electric resistance heat. The consumption in winter would have been

Table 3. Temperature Data and Coefficient of Performance for the Heat Pump Unit*

Heating

Date	Evaporator incoming air conditions		Condenser incoming air conditions		C.O.P.
	Dry bulb temperature	Wet bulb temperature	Dry bulb temperature	Wet bulb temperature	
February 20	58	54	33	30	2.5
February 24	59	51	44	40	3.6
February 27	59	51	48	43	2.7
February 27	65	56	50	45	3.4
March 1	52	40	52	46	1.7
March 1	50	39	52	41	2.8
May 29	71	59	66	58	2.9
				Average	2.8

Cooling

Date	Condenser incoming air conditions		Evaporator incoming air conditions		C.O.P.
	Dry bulb temperature	Wet bulb temperature	Dry bulb temperature	Wet bulb temperature	
March 1	55	43	49	39	1.6
March 1	55	42	50	40	1.6
May 29	72	59	65	56	1.1
May 30	78	68	70	26	2.1
June 1	76	60	70	56	1.1
June 3	92	70	85	67	1.5
June 4	91	71	82	68	1.6
June 5	85	66	78	65	1.8
June 6	97	72	88	70	1.3
June 7	85	68	78	66	2.5
June 8	88	69	81	66	2.2
June 9	95	73	88	71	1.4
June 10	85	71	80	69	1.6
June 11	84	68	75	65	1.8
June 12	74	60	68	57	1.7
				Average	

*These values were obtained with air flows at 1,250 cfm through the condenser and 1,400 cfm through the evaporator.

relatively low without the electric resistance heat. Part of this power consumption by the resistance heat was due to faulty operation of the heat pump unit at which time the resistance heat carried most of the load. This was especially true early in the winter during December.

Figure 18 compares total power consumption with the average outside temperature. It was very evident in this case that there was a very strong relationship here with the power consumption rising at either temperature extreme. The equation of the curve was calculated along with the correlation coefficient. The equation is: $Y = 5.423 - 0.20639X + 0.003266X^2$ where Y is the power consumption in kilowatts and X is the average outside temperature in degrees Fahrenheit. Notice that this is a second degree curve with the second term being negative which gives the upward direction to the curve at the lower temperatures. The coefficient of correlation was 0.928 which indicates the strong relationship between power consumption and outside temperature. This formula should therefore be useful for predicting power consumption of a similar system in this area or for predicting power consumption for this system in future applications.

The total energy consumed during the test period of 10½ months was 31,842 kilowatt hours which would cost $477.63 at 1½ cents per kilowatt hour. For a full year this would be about $546. This seems like an extremely high cost but probably could be reduced somewhat if the air filtering problem was solved. The total installed cost of the system amounted to approximately $1,600 of which $1,100 was the cost of the two units in the spring of 1958 and the other $500 was paid for materials and labor for the installation during June of 1958. Using $1,600 as the original installed cost and a life of 10 years, the annual depreciation amounts to $160. Interest at 6% amounts to an average value of $48 per year. Assuming that service and repairs amounts to 2% of the original cost per year, a value of $32 per year is obtained. Adding these values gives an annual cost of owning and operating of $786 per year. This value must then be compared with the benefits to properly evaluate the system.

Benefits of the system are difficult to accurately determine and probably vary from year to year depending on the length and temperatures of the winter and summer seasons. Feed consumption, egg production, and mortality records for birds in the house were kept, but since there was no control house of comparable construction, there can be no accurate comparison.

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