Refinements in heat exchangers for the ventilation of animal shelters

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REFINEMENTS IN HEAT EXCHANGERS FOR THE
VENTILATION OF ANIMAL SHELTERS

by

Eric B. Moysey

A Thesis Submitted to the Graduate Faculty in Partial Fulfillment of The Requirements for the Degree of

MASTER OF SCIENCE

Major Subject: Agricultural Engineering (Farm Structures)

Signatures have been redacted for privacy

Iowa State College
1950
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I. INTRODUCTION

The problem of ventilating dairy barns and other animal shelters in the colder climates of this country dates back to the time when improved building construction resulted in barns which were more or less air tight. In early times barns were merely shelters from winds and storms, but as time passed the animal husbandry-man tried more and more to control the temperature of the surrounding air. This was accomplished by insulating the barn and by making it tight against the weather. A reasonable degree of temperature control was attained but only at the expense of stagnant and humid air in the building. The humid air resulted in condensation on the walls, windows and ceiling of the barn, usually in the form of frost which melted later during a period of mild weather and dripped on the animals and floors. Comfort conditions for the animals and operator were thus often worse than the barns of poorer construction. In addition, deterioration of the structure was definitely enhanced.

The aim of any ventilation is to provide conditions of air temperature, air quality, humidity and air circulation conducive to maximum performance, health and comfort of the individual and, where production is carried on, to provide conditions which result in a satisfactory product. According to Hutcheon (10,p.1):
The most desirable conditions must first be determined by careful scientific investigation and must then be provided by the engineer in-so-far as it is economically and practically feasible to do so. As in all practical problems, there is finally some inevitable sacrifice of the ideal in effecting a satisfactory compromise between various conflicting factors. The one factor which most often comes into conflict with others is cost; the final result must justify the means so that people are willing to contribute the human effort necessary to achieve the final result.

The environmental conditions most desirable for the dairy cow have been quite well established. Recent work in the psychro-energetic laboratory at Columbia, Missouri had, among other objectives, the provision of basic engineering data for adapting ventilating and air conditioning systems to dairy barns.

The heat given off by the animal is customarily the only source used to balance the radiation and conduction heat loss through the building walls and the heat loss to the air of ventilation and infiltration. An analytical solution of the problem based on this information indicates that it is impossible to maintain a comfortable temperature in a barn of average construction when outdoor temperatures drop to near the zero mark and still ventilate at a rate which is sufficient to remove the water vapor given off. Experience has proven this to be true. At least three solutions have been suggested which would aid in solving the problem. One is the addition of insulation to the walls and ceiling of the structure, thereby reducing the building heat loss to conduction and radiation. This can provide at least a partial solution.
Secondly, heat may be added to the structure from an outside source, a satisfactory but expensive answer which has not gained wide acceptance. A third and more common procedure is to restrict the ventilation rate during periods of low outdoor temperature and so reduce the heat loss to ventilation. If the cold wave is of short duration, only minor amounts of condensation may occur, but in many areas of the United States and Canada the temperature may remain below zero for a period of days or even weeks, resulting in condensation of serious proportions.

Since none of these methods provides a completely satisfactory answer, research was undertaken at Iowa State College four years ago on the possibility of using a sample multtube heat exchanger to conserve the ventilation heat loss by using the warm exhaust air to heat up the cold incoming air. In this way, a higher ventilation rate would be permissible during periods of cold weather. Much of the basic research has been done in evaluating heat transfer co-efficients, condensation rates, power requirements and such.

It is, therefore, the purpose of this investigation to determine how a heat exchanger can best be made practical and economical for the ventilation of animal shelters.
II. REVIEW OF LITERATURE

A. Heat and Water Vapor
Production in Cattle

Respiration in cattle involves the consumption of oxygen and the liberation of carbon dioxide, water vapor and methane. The production of methane and carbon dioxide in the alimentary canal provides a substantial difference between respiration in herbivores and respiration in man. The variation in rate and depth of respiration with environmental temperature is also considerably different, for the cow depends to a very great extent on respiration for the regulation of body temperature. Cattle of European origin do not have functional sweat glands except in certain limited areas. It is this inability to sweat which forces them to respire at a sharply increasing rate with increasing environmental temperature so as to liberate latent heat from the lungs. When the surrounding air temperature reaches the level of the body temperature, which is about 101°F in cattle, it is impossible for them to lose heat by radiation and convection from the body surface and vaporization of water from the lungs provides practically the only means of heat elimination. This compensation by evaporative cooling is not very substantial in Holstein and Jersey cows, for a rapid rise in rectal temperature occurs as
the environmental temperature rises above 70°. The respiration rate begins to rise sharply at about the same temperature and when the environmental temperature reaches 100°F the respiration rate is four to five times that at the 50° level. Regan and Richardson (20) found that the respiration rate of cattle doubled for every 18°F rise in temperature above 40°. Gaalaas (7) did not find as rapid an increase above 50° while Kibler (13) corroborated Regan's results above 70° but found a lower rate of increase below this temperature.

The amount of heat lost from the body of a cow by conduction, convection and radiation is about 75 per cent of the total heat eliminated under ordinary conditions. The vasomotor nervous system controls the amount of heat lost in this manner. Vasoconstriction causes a decrease in the blood flow through the surface layers of the skin which decreases the heat loss while vasodilation has the opposite effect. As might be expected, the pulse rate increases with decreasing respiration rate occasioned by decreasing environmental temperature. The load on the heart thus increases when the vasomotor nerves call for more blood to keep the surface of the body warm. Heat loss is further controlled by the piloimeter nerves which regulate the degree of erection of the hair. Erection of the hair entraps a thicker layer of air and the animal keeps itself warm much as a bird keeps itself warm by ruffling up its feathers.

Heat production in cattle varies with environmental conditions but at a particular condition is dependent on the
basal metabolism and the increments of heat due to feeding and movement. Basal metabolism refers to the heat production of an animal at rest in a thermally neutral environment when the stimulating effect of food has disappeared. Armsby, (2) in 1921 stated the heat production for fasting katabolism as:

\[ y = 0.998 \frac{x^{2/3}}{} \]

where \( x \) = live weight in pounds
\( y \) = heat produced in Btu/hour

The total heat produced by the animal was assumed to be the sum of the fasting katabolism and the increments of heat due to feeding. Kriss (16) later established a relationship between the weight of the dry matter in the ration and the heat produced per unit live weight. The relationship he found was:

\[ y = 1.025x + 1366 \]

where \( y \) = calories per day per 100 kg. of live weight
\( x \) = grams of ration per day per 100 kg. live weight

Transposed to English units this is

\[ q = 76.91w + 102.71 \]

where \( q \) = Btu/hr per 100 pounds of live weight
\( w \) = dry matter in ration in pounds per 100 pounds of live weight.

The heat production found from this equation is about 4 percent lower than values obtained by the Armsby method. Forbes and associates (6) made studies of the energy metabolism of
cattle and also got results which were slightly higher than Kriss. The very recent work done by Kibler and associates (13) at Columbia, Missouri confirms the work of the earlier investigators to a great extent.

A correlation of the results of the aforementioned investigators allows an assumption that an average cow weighing 1250 pounds producing about 30 pounds of milk per day on a standard ration produces approximately 3600 Btu of heat per hour when the environmental temperature is around 50°F. A cow weighing 1100 pounds produces about 3100 Btu/hr under similar conditions. The heat production of dry cows is, of course, somewhat less.

The agreement between the results of various investigators into the water vapor production of cattle is rather poor. The major portion of the water vapor produced comes from respiration, small amounts being evaporated from the skin, urine and feces. The water of respiration is usually stated in terms of the total heat production, for a portion of the heat produced is in the latent form due to vaporization in the lungs. Figure 1 shows the results of five investigators with per cent heat in the latent form plotted against environmental temperature. Thompson and associates (24) explains the hump in his curve in the region of 40° as being due to experimental procedure, for the temperature was lowered from 40 to 0 degrees and then raised from 50 to 90 degrees. The curves for total moisture production found by Ibrahim (12) and Downing (4)
Fig. 1 Water vapor production in dairy cattle.
were incidental to their work on heat exchangers. Their curves were not presented as being particularly accurate as estimation of infiltration and total heat production were involved. Both are based on a statistical analysis of numerous points and should give a fair indication of conditions. It is interesting to note how Ibrahim's estimation of the proportion of the total moisture which comes from respiration checks the curve of Thompson and associates (24) which is based on direct measurement of respired air. Forbes and others (5) did their work on steers. They also found that the per cent of heat in the latent form was considerably less for sheared steers than unsheared steers. This is to be expected for dissipation of heat could then occur more readily by radiation and convection.

As previously stated, heat production varies widely with body weight and plane of nutrition. It also differs between individuals of apparently equal physical characteristics and may vary within one individual at different times. Kibler and associates (13) also state that heat production declines as the environmental temperature rises above 80°, which is rather surprising. On the other hand, the per cent of latent heat is assumed to vary greatly with environmental temperature but to be independent of body weight.

B. Desirable Environment

It has been stated (10) that the aim of ventilation is
to provide conditions of air temperature, air quality, humidity and air circulation conducive to maximum performance, health and comfort of the individual and, where production is carried on, to provide conditions which result in a satisfactory product. Applied to dairy barn ventilation, this means provision of an environment conducive to the most economical production and of satisfactory conditions of sanitation. Much has been written in this regard but there is still considerable contention that stables are kept warm more for the comfort of the operator than of the cow. Perhaps this argument could be settled if cows could talk, but judging from the differences of opinion among men, cows probably could not come to agreement either. One point is quite evident from the literature; conditions in the vicinity of 50° and 70 per cent relative humidity are generally considered favorable while temperatures between 70 and 100° have considerably more adverse effects than temperature between 0 and 40°F. This is borne out by the recent work of Ragsdale and others (19). Another statement often made is that rapid fluctuation of temperature reduces the milk yield more than continued exposure at low temperature. Management, then, is a factor, for temperatures may fluctuate more in good barns poorly managed than in fair barns which are well managed.

The plane of nutrition has been shown to influence the heat production in a dairy cow. For any particular plane of nutrition there will be a certain environmental temperature at
which the heat dissipated from the body will exactly equal the heat of production. Below this temperature there will be a tendency for oxidation of tissue to occur in order to keep the animal warm; above this temperature the animal will have to respire more rapidly to keep cool. There is also the effect of appetite which tends to compensate for temperature by decreasing the amount of feed taken in at high temperatures, thus reducing the heat produced. Conversely, the appetite increases with temperatures below the normal critical temperature, thus increasing the food consumed and heat produced. Another way of stating it is to say that feeding lowers the critical temperature. Ragsdale and others (19) found 80° to be the critical high temperature but discovered no sharp critical low temperature. The point of most economical production appeared to be about 50°F.

From the foregoing discussion it would seem that temperatures of 45 to 55°F are optimum with 50 to 60° being allowable so long as the temperature does not fluctuate rapidly. Humidity should be below 80 per cent, exact value depending on condensation problems.

Carbon dioxide content serves as a convenient means for measuring the purity of air and air purity is often stated on the basis of carbon dioxide content. This is probably the reason many people even at the present time believe that the ill effects experienced in poorly ventilated quarters are due to an excess of carbon dioxide. The carbon dioxide content
of outdoor air is about 0.03 per cent and rarely exceeds 0.5 per cent with the poorest of ventilation whereas it is possible for it to be present in inspired air by as much as 2 or 3 per cent without the individual being aware of anything unusual, for carbon dioxide is not poisonous. Similarly, the oxygen content of air in poorly ventilated quarters does not drop to dangerous levels. The feelings of drowsiness and discomfort usually associated with poor ventilation are due more to heat stagnation within the body and the presence of odors than to the excess of carbon dioxide. Stale humid air which is stagnant or lacking in circulation does not remove either heat or odors from a body readily.

Some forty years ago King (14) suggested some standards of ventilation which were widely accepted and used. Purity of air was the basis he used for determining the quantity of ventilation, his recommendation being that a building should never contain more than 3.3 per cent of air once breathed. On this premise, a cow should receive about 3545 cu. ft. of air per hour. However, ventilation in a dairy barn serves as practically the only means of temperature control. The heat production from cattle is reasonably constant for a given temperature but the heat lost through the building walls varies with the temperature difference from outside to inside. If the indoor temperature is to remain constant during periods of mild weather outdoors, as is desirable, the quantity of ventilation must be greatly increased. If the removal of
water vapor is used as a criterion the rate of ventilation varies widely with outdoor temperature also, for the difference in water holding capacity decreases with the difference in outside to inside temperature. On the water vapor removal basis, the quantity of ventilation required is about 2000 cu. ft. per hour at an outside temperature of \(-10^\circ\), which is more than sufficient from the standpoint of health of the animal.

C. Ventilation of Dairy Barns

Only a brief look through the Agricultural Index to periodical literature is required before one realizes that a mass of literature has been written on the subject of dairy barn ventilation. Examination shows that a great part of it consists of reports on the operation of some particular system and most of these close by stating that the system functioned well, fairly well, poorly or a similar type of conclusion. Many of these have been reported as unsatisfactory when, in fact, it was impossible for them to have performed better under the conditions. Most of these reports have the following factors in common: (1) the aim of ventilation is to remove water vapor and to maintain a constant level of temperature in the barn, (2) the animal is the only source of heat. However, few take cognizance of the engineering phases of the problem and therefore do not realize what the optimum results can be within the limitations they have imposed.
Methods of ventilation to the present fall in two general categories, namely natural draft and mechanical. The natural draft system is, of course, the oldest and most generally used. It depends for its motive power on the chimney effect, wind pressure or a combination of the two. The chimney effect in a flue is only obtained when the air inside the flue is warmer, and therefore less dense than the air surrounding. The motive force produced is equal to the difference in weight of two columns of air, equal in height to the height of the flue. A flue 35 feet high in a barn at 50° and 75 per cent relative humidity will produce a motive head of 0.061 inches of water when the outside air is at 0° and 75 per cent relative humidity. Pressures in conventional systems of commercial ventilation are often thirty times this value. When outdoor conditions change to 30° and 75 per cent relative humidity, the head available for flow is only 0.024 inches of water. As has been previously shown, it is during periods of mild weather that greater quantities of ventilation are required to remove water vapor and dissipate heat and it is at this time that the chimney effect is least. Reliance on wind pressure is equally as unsatisfactory, for although wind can cause appreciable flow, the system will be extremely variable in operation. Proper design and location of inlets and outlets can eliminate the effects of wind to a large extent, making a natural draft system less sensitive to
changes in weather. Manual control of dampers, however, will be required to give reasonable ventilation during changes of temperature.

The three types of natural draft systems of ventilation most widely known are the King, the Rugherford and the modified Rutherford. In the King system the out-take flue draws air from near the floor and exhausts it above the ridge of the barn while the fresh air enters at the ceiling. Intakes and outlets are insulated to reduce condensation and prevent cooling of the air in the outlet. The system induces continuous movement and mixing of the air in the barn, while the heat reservoir maintained by the floor out-take prevents sudden changes in temperature. In the Rugherford system, the insulated exhaust flue extends from the ceiling to the ridge and fresh air enters the barn at the floor level. A smaller area of exhaust flue is required but the system is sensitive to changes in outdoor weather. It does provide complete circulation and air movement. The modified Rutherford system differs by having the intakes open into the barn at the ceiling. The circulation of air effected is probably not so complete and the system is quite sensitive to changes in outdoor weather. The modified King system, sometimes known as the Fairbanks-Goodman system of ventilation is a more recent innovation. In it, a single intake flue which draws air from near the floor is used for barns up to 200 feet in length. Several inlets are used to deliver air straight up along the
side walls and so promote complete circulation. It is recommended that the flue be well insulated from top to bottom and that two inlets, each 60 sq. in. in area, be provided for every seven cows.

Any natural draft system of ventilation has the inherent disadvantage that the motive head is very low when the weather is mild, although it may function quite well during cold weather. It is during mild weather that the difference in specific humidity between inside and outside conditions is the least and therefore the amount of ventilation required to remove a given quantity of water vapor is greatest. Then too, if the indoor temperature rises, cattle give off a greater proportion of their heat in the latent form, requiring still more ventilation. It was then, a natural step to introduce mechanical ventilation so that a positive amount of air could be delivered at any time. Variation of air quantity could be accomplished by dampers or by shutting off one or more fans.

Considerable research work has been done during the past few years on the use of mechanical ventilation in animal shelters and much has been written about it. In brief, a review of the literature shows the following recommendations to be consistent with good practice and generally agreed upon.

1. Exhausting of the air should take place from one location in the barn. If more than one fan is used they should be placed side by side; otherwise a dead air space may result between them.
2. If only one fan is used it should be allowed
to run continuously and variation in quantity obtained by
dampers. (3) If more than one fan is used, at least one should
be allowed to run continuously while the others may be con-
trolled by thermostat or humidistat. (4) Inlets should be
evenly spaced around the perimeter of the barn and the incom-
ing air should be directed upward. (5) It makes little dif-
fERENCE whether the air is exhausted from near the ceiling or
floor. Air near the ceiling is usually warmer but also has a
higher specific humidity so the ratio of heat to vapor removed
is practically constant. King (14) argued that air should be
removed from near the floor because the concentration of car-
bon dioxide was heaviest there, carbon dioxide being heavier
than air. Carbon dioxide elimination is not the primary rea-
son for ventilation, as has been shown, so this need not be
a controlling factor in positioning outlets.

The governing factors in the design of a dairy barn ven-
tilation system have been stated to be: (1) moisture removal,
(2) balance between heat produced in the barn and heat lost
from the barn (12). Strahan (23) presented a formula for ex-
pressing this relationship:

\[ H = \frac{VD}{53} + ACD \]

in which \( H \) = total heat produced in Btu/hr per cow

\[ \frac{VD}{53} = \text{Btu lost in ventilation per cow per hour} \]

\[ ACD = \text{Btu lost by radiation and conduction per hour per cow} \]
V = cubic ft. of air per hour per cow (includes infiltration and ventilation)

D = temperature difference inside to outside

53 = \( \frac{1}{\rho c} \) where \( \rho \) is the air density in lb/cu.ft. and \( c \) is the specific heat of air.

A = the area of exposure per cow to walls, windows, doors, etc.

c = average weighted heat transmission co-efficient in Btu/hr/sq.ft./°F

The formula is not strictly accurate as the number 53 is not constant for varying temperatures and humidities, but it does give a close approximation for design purposes. For more precise heat balances, other effects such as radiation on clear days and nights should be taken into account. Preliminary calculation based on values given by Severns (21) indicate that the heat gain on a bright, sunny day may be as high as 40 Btu per hour per sq. ft. of south wall. Henderson (9) suggested that a loss of 25 Btu per hour per sq. ft. might be expected from a black body due to radiation to the sky on a clear calm night. Thermal capacity of the structure under changing temperatures is another possible source or error in obtaining heat balances. Ibrahim (11) studied soil temperatures in an attempt to discover whether heat might be gained from or lost to the soil beneath a building. In Iowa, the soil temperature at a depth of six feet is reasonably constant at 50°F during the winter months so there should be little gain or loss of heat from a barn kept at this temperature. Edge loss to foundation may be expected to have a slight effect.
These factors could account for what would otherwise appear to be discrepancies in heat balances calculated by Strahan's formula.

Clyde (3) suggested that when mechanical ventilation is used, some form of heat exchanger could conceivably be made that would conserve a fraction of the heat lost in the exhausted air. Ibrahim (11) made a simple double tube heat exchanger which appeared to be effective, from which a more complex unit was designed for a particular dairy barn. Downing (4) made and installed a very similar unit in a dairy barn in Ontario. Both investigators obtained encouraging results. Almost half the heat lost in the exhaust air was recovered in some cases, which allowed higher rates of ventilation at low outdoor temperatures than would otherwise have been possible if the barns were to remain at comfortable temperatures.

Evaluation of heat transfer coefficients proved difficult due to condensation on the surfaces. Distribution of fresh air and pick-up of exhaust air required much duct work and also increased power requirements. Variation of air quantity with outdoor temperature was accomplished by dampers; therefore the greatest quantity of air and highest rates of heat transfer occurred when the need for heat salvage was least. In brief, the value of using such a heat exchanger was demonstrated and many of the basic quantities evaluated. However, there was shown to be a need for further work in making a heat exchanger more practical and economical for public use.
It was with this goal in view that the research work here reported was initiated.
Heat may pass from a region of high temperature to one of lower temperature by conduction, convection, radiation, or by any combination of the three. Conduction involves transfer of heat by molecular bombardment, whereas actual movement of the individual particles occurs in convection. Radiation is an electro-magnetic wave phenomenon and, compared to the other two methods, is unique in that it can occur through a vacuum. Convection usually involves conduction and radiation along with the physical mixing of the fluid.

The coefficient of heat transfer for convection, usually stated as \( h \), involves many complicated phenomena such as velocity, area, shape, and mean temperature, but can be determined from empirical formulae for conditions usually met in practice. For longitudinal flow of fluids in cylinders an approximate relation is:

\[
\frac{hD}{k} = 0.0225 \left( \frac{DV}{k} \right)^8 \left( \frac{C}{k} \right)^4
\]

provided \( \frac{DV}{k} \) is larger than 3000. In the work to be considered \( \frac{DV}{k} \) lies in the range of 10,000 to 20,000. For longitudinal air flow in cylinders this formula reduces to \( h = 0.0036 \frac{C}{D} \) over a wide range of temperatures in which

\[
h = \text{unit conductance for thermal convection, Btu/hr. sq.ft.}^{\circ}\text{F}
\]
G = 3600 \( V_s \rho \) = fluid mass velocity, pounds per \( \text{hour} \cdot \text{sq.ft. of flow cross section} \)

\( V_s \) = fluid velocity, ft. per second.

\( \rho \) = air density, pounds per cubic foot.

D = cylinder diameter in feet.

This same expression is applicable for longitudinal air flow in shapes other than cylinders provided the hydraulic radius is used as the dimension parameter of the conduit. Thus for non-circular cross sections an equivalent diameter is used which is equal to four times the area of the cross section divided by the perimeter. That is, \( D_e = \frac{4A}{p} \). When the heat exchanger is composed of a number of sheet metal pipes in a rectangular duct, heat is transferred from the air in the pipes to the air in the annular space around the pipes, or vice versa. Heat must therefore be transferred through the two surface films and through the thin metal wall so that the overall coefficient of heat transfer \( U \) is:

\[
U = \frac{1}{\frac{D_0}{D_1 h_1} + \frac{1.15 D_0}{k} \log \frac{D_o}{D_1} + \frac{1}{h_0}}
\]

where \( D_1 \) and \( D_0 \) are the inside and outside pipe diameters, respectively.

\( h_1 \) and \( h_0 \) are the inside and outside surface coefficients of heat transfer.

\( k \) is the thermal conductivity of the pipe wall per inch of thickness.

Since \( D_1 \) and \( D_0 \) are practically identical and \( k \) is approximately 100 times as large as \( h_1 \) and \( h_0 \), \( U = \frac{1}{h_1 + h_0} \) for practical purposes.
The formula previously given for determining \( h_1 \) and \( h_0 \) is based on experiments with smooth pipes. Where the pipes are joined at intervals and supports introduced in the air stream, the heat transfer can be expected to be greater due to increased turbulence. If condensation occurs, the apparent coefficient of heat transfer will also be increased due to heat liberated during condensation, even though the moisture film provides an additional resistance. Downing (14) and Ibrahim (11) both found that the actual overall coefficients of heat transfer \( U \) were from 20 to 70 per cent greater than the theoretical values but were not able to determine from their experiments the proportions which were due to turbulence and condensation. The increase in heat transfer due to condensation will, of course, vary with the amount of condensation and the portion of the pipes on which condensation is occurring.

The total transfer of heat is given by the following relationship:

\[
Q = U A \theta_m
\]

where

- \( Q \) = heat transferred
- \( U \) = overall coefficient of heat transfer
- \( \theta_m \) = logarithmic mean temperature difference

\[
\theta_m = \frac{\theta_A - \theta_B}{\log_e \frac{\theta_A}{\theta_B}}
\]

[Diagram of heat transfer with cooling and heating symbols]
The logarithmic mean will not be the true mean temperature difference, but according to Ibrahim (11) it gives a very close approximation. The total heat transferred must also be equal to the change in enthalpy of the outgoing or incoming air as it passes through the exchanger. The total heat content of a pound of air and its contained moisture is often expressed as

\[ h = C_{pa}t + W\left(h_f + h_{fg} + C_{ps}(t - T')\right) \]

where
- \( h \) = heat content, Btu per lb of dry air
- \( C_{pa} \) = specific heat of air at constant pressure
- \( t \) = air dry-bulb temperature, deg. F
- \( T' \) = air wet-bulb temperature, deg. F
- \( W \) = lb of moisture per lb of dry air
- \( h_f \) = heat of the liquid at the wet-bulb temperature
- \( h_{fg} \) = heat of evaporation at the wet-bulb temperature, \( \text{Btu/lb} \)
- \( C_{ps} \) = mean specific heat of the vapor

Total heat contents may more easily be read from a psychrometric chart. If \( M \) is the weight of air moved per hour, then

\[ UA\Theta_m = M(h_1 - h_2) \]

where \( h_1 \) and \( h_2 \) are the heat contents of the air entering and leaving the exchanger. Using these equations it is possible to predict the performance of a particular heat exchanger for a known set of conditions or to determine the actual value of \( U \) from measurements of temperature, humidity and air quantity.

The moisture content of the warm exhaust air is always greater than that of the cold, fresh air, per pound of air...
moved. It follows, then, that for a unit change in heat content the temperature of the fresh air will be raised more than the temperature of the exhaust air is lowered if the weights of air moved in each case are equal. Therefore, it will never be possible to reduce the exhaust air temperature to the entrance temperature of the incoming air.

It is practically impossible in practice to have the heat gained by the incoming air exactly equal the heat lost by the exhausted air due to loss or gain of heat from the surroundings. If the heat exchanger system is placed in the barn proper, the air surrounding the outer duct will at all times be warmer than the air in the duct and there will be a heat gain to the air in the exchanger as a result. The outgoing air will be exhausted from the barn at a somewhat higher temperature and the heat salvaged will be less as a consequence. If, on the other hand, the heat exchanger is placed in the hay mow, the surrounding air will be cooler than much of the air in the system. There will be a heat loss from the warm exhaust air which means that the cold incoming air will not be able to gain as much heat. In this case, too, there is a net loss in heat salvaged. Downing and Ibrahim recognized this fact early in an attempt to increase the heat saving. Downing's system was located in the hay mow and Ibrahim's in the lower part of the barn, but in each case the addition of insulation around the system substantially increased the heat salvaged. However, it would only be theoretically possible to have the heat lost
by the exhaust air equal the heat gained by the incoming air if the thickness of insulation was infinite. Downing suggested that this could in effect be accomplished by burying the system in the hay mow under several feet of hay. Ibrahim proposed that an outer duct be used to replace the insulation. The exhaust air could be picked up in this outer duct and then passed through the tubes while the fresh air made a single pass through the annular space. The additional heat transfer might quite conceivably save as much heat as the insulation, even though heat would still be gained from the surrounding air. The outer duct would also replace the pickup duct ordinarily required and would reduce condensation problems on the outer surface of the heat exchanger. Cross sections of the three types of systems are shown in figure 2.

![Diagram of three types of heat exchangers.](image)

**Fig. 2. Three types of heat exchangers.**

Theoretical calculation of the probable heat transfer coefficients in the outer duct are rather difficult. If the heat exchanger is treated as a single shell, double pass type, the equation \( Q = U A \theta_m \) is still true but \( A \) is now the total area of heat transfer and \( \theta_m \) must be modified depending on
inlet and outlet temperatures. The factor by which $Q_m$ must be multiplied can be obtained from graphs in any text on advanced heat transfer. This is possible if the same quantity of air is passing through both the inner tubes and outer duct, but when air is picked up all along the outer duct through a narrow slot, the problem is exceedingly complicated. Integration is required to determine the total heat transfer along the outer duct. In the equation $UAQ_m = M(h_1 - h_2)$, $U$ increases as the air is picked up which makes the problem impractical to solve for the many conditions which would be encountered. An approximate solution can be obtained by assuming the coefficient of heat transfer constant and the area to be that of the tubes plus one-half that due to the outer duct. The net saving in heat is the point of interest and this can be determined with accuracy without theoretical calculation for this portion of the heat exchanger. Experimental coefficients of heat transfer on the tubes can be checked against theoretical values with comparative facility to determine whether or not experimental results are reasonable.
IV. THE INVESTIGATION

A. Outline of Experiment

1. Description of barn

The barn used for the experiment was the east wing of the north barn at the Iowa State College Dairy Farm. Attached to the west end of the experimental barn is a single story structure used as a loose housing maternity barn. The two structures together form an L shape. The experimental barn is of the conventional two-story, gable roof type and is stocked with milk cows in production and a few calves. Structurally, the barn is far from ideal. It is old, of double boarded frame construction, with no insulation and is far from air tight. Its long axis is east and west which is not the orientation recommended to make best use of the heat from the sun. The interior lay-out is such that there is about 1100 cubic feet of space per animal, whereas 500 to 600 is recommended as being desirable from the standpoint of heating and ventilation relationships. Nevertheless, it is in this quality of structure that ventilation problems are the most severe and the most trouble is encountered in keeping them dry and warm during periods of cold weather. It is felt that a heat exchanger would be most valuable and find
its widest application in this type of structure.

A view from the southeast is shown of a portion of the barn in Figure 3. Figure 5 shows a floor plan of the experimental barn. The 14' x 19' equipment room in the southeast corner contains a stove, air compressor and experimental equipment. There was usually a fire in the stove during the winter which kept this room as warm or slightly warmer than the area where the cows were stabled. Along the south side are eight stanchions and two pens in which cows were kept. Pens numbered 15 to 20 along the north side of the building were used to stable cows while small calves were kept in pens 11 to 14. During the course of the experiment there was only one change in the cows; the cow in pen number 18 was taken out on February 17 and a smaller cow brought in. The number of calves housed varied from three to six but since their average weight was about 50 pounds, the effect on results of the experiment was ignored. The average weight of the cows was 1130 pounds. The equivalent number of 1250 pound cows would be

\[
\frac{1130 \times 16 + 3 \times 50}{1250} = 15.7
\]

Replacement of the cow in pen number 18 on February 17 by a smaller cow reduced this to 15.3 equivalent 1250 pound units.

The experimental barn shown in Figure 5 has five outside doors and a door into the adjacent barn. There are ten windows 2'-5" x 3'-1" all of single glass. The wall construction consists of 2" x 6" studs with 3/4" matched lumber on the
Fig. 3. View from southeast showing inlet, outlet and auxiliary fan.

Fig. 4. General view of barn interior.
Fig. 5. Floor plan of experimental barn.
inside. The stable ceiling and hay mow floor are made of 12" joists with 3/4" matched lumber on each side. Hay and wood shavings kept in the mow served as insulation over part of the surface. Heat loss to the mow and the adjacent barn was estimated on the basis of spot temperature readings in these areas. The heat losses from the various exposed surfaces are summarized in Table 1.

Table 1
Heat Losses from Experimental Barn

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<tr>
<th>Surface</th>
<th>Area</th>
<th>U</th>
<th>Heat Loss</th>
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<td></td>
<td>sq.ft.</td>
<td>U</td>
<td>d/hr. °F</td>
</tr>
<tr>
<td>Outside Walls</td>
<td>554</td>
<td>0.269</td>
<td>149</td>
</tr>
<tr>
<td>Wall to adjacent barn</td>
<td>360</td>
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<td>54.3</td>
</tr>
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<td>Windows</td>
<td>67 1/2</td>
<td>1.13</td>
<td>76.3</td>
</tr>
<tr>
<td>Doors</td>
<td>98</td>
<td>0.69</td>
<td>67.6</td>
</tr>
<tr>
<td>Ceiling - bare</td>
<td>864</td>
<td>0.262</td>
<td>166.0</td>
</tr>
<tr>
<td>Ceiling - with hay</td>
<td>830</td>
<td>0.004</td>
<td>Ignore</td>
</tr>
<tr>
<td><strong>Totals</strong></td>
<td>2773 1/2</td>
<td></td>
<td>515.8</td>
</tr>
</tbody>
</table>

AC value = \(\frac{515.8}{15.7} = 32.85\)

The temperature of the inside surfaces determines when and where condensation will occur. The difference in temperature from the barn air to the surface is to the indoor to outdoor temperature difference as the resistance of the surface film is to the total resistance of the wall.
Expressed mathematically,
\[ t_s = t_i - \frac{R_s}{R_w} (t_i - t_o) \]
where \( t_s \) = surface temperature
\( t_i \) = indoor air temperature
\( t_o \) = outdoor air temperature
\( R_s \) = resistance of the surface film
\( R_w \) = resistance of the wall

For the outside wall
\[ t_s = t_i - .1633 (t_i - t_o) \]

For the windows
\[ t_s = t_i - .685 (t_i - t_o) \]

If the outside temperature is \(-20^\circ\) when the barn temperature is \(40^\circ\) the surface temperatures will be \(31.2^\circ\) and \(-1.1^\circ\) for the walls and windows, respectively. Air at 70 per cent relative humidity in the 40 to 70 degree range may be cooled 10\(^\circ\) before condensation will occur. Condensation on the windows appears to be unavoidable but maintaining the humidity at 70 per cent should prevent condensation on the walls or this barn under anything but the most extreme conditions.

2. Description of equipment

The heat exchanger unit consisted of four rows of 3 inch diameter pipes, four pipes in each row, making a total of sixteen pipes. The pipes were placed on 3 3/4 inch centers and supported at five foot intervals by narrow sheet metal strips.
The pipes were surrounded by a sheet aluminum duct sixteen inches square which was, in turn, surrounded by a duct twenty inches square. Plan and cross-section views are shown diagrammatically in Figure 6. The pipes and inner duct were 31 feet long; the outer duct was 25 feet long.

Air to be exhausted was drawn from the barn through a perforated 9" pipe "E" and forced by the fan into the annular space around the pipes at "F" and then out of the building at "G" through 12" diameter pipes. During periods of mild weather, when the full benefit of the heat exchanger was not required, it was possible to open the valves and force the air directly out of the building at "H". Fresh air was drawn in at "A" and passed through the small pipes to "C" and was then forced into the space between the inner and outer ducts at "D" and so out into the barn through the 1/2" slots in the outer duct. The total area of the slots was made such that the air would be discharged from them at an angle of approximately 60° with the axis of the exchanger (15). Provision was made for short circuiting the exchanger during periods of mild weather by inlet "B". During periods of extremely cold weather recirculation of a portion of the exhaust air could be accomplished by opening dampers into the 9" perforated pipes "J". Two presumably identical Buffalo Forge Company fans with 3/4 horsepower motors were placed in the hay mow to operate the system. It will be noticed that the fresh air is drawn through and the exhaust air is forced through the system.
Fig. 6 SCHEMATIC DIAGRAM OF EXPERIMENTAL EQUIPMENT
With this arrangement, any heat added to the air by the fans and motors is used to the best advantage.

The fans and motors were considerably larger than was advisable from the standpoint of economical operation. They were purchased with flexibility of operation in mind so that the heat exchanger could be tested through a full range of air velocities. Variation of air quantity was accomplished both by the manipulation of slide values and by changing fan speed with an assortment of pulley sizes.

As the warm exhaust air traveled through the annular space around the pipes, it was often cooled below the dew point which produced condensation on the outer surfaces of the small pipes. Provision was made at the south end of the inner duct for draining this condensate either into a pail to be weighed or directly outdoors. Condensate was carried to the south end by a grade of two inches in 31 feet and by the air motion in this direction.

An auxiliary propeller type fan capable of producing a flow of 1000 cubic feet per minute was installed in the south wall and controlled by both a switch and thermostat. This fan was used to provide additional flow during mild weather as an alternative to the short circuiting method.

3. Instrumentation

The quantity of air delivered by a fan varies with the speed of rotation and the characteristics of the system but is
practically independent of the density of the air flowing. Therefore, for a particular fan speed and valve opening, the volume of air moved is constant while the weight of air varies with the density. In this experiment air quantities were computed by multiplying the average velocity in the ducts by the cross sectional area for different fan speeds and valve positions. The average velocities were obtained by making twenty point traverses of the ducts with an Alnor velometer. This instrument is a bridle vane type of anemometer. The thermoanemometer gives accurate readings of velocities as low as 50 feet per minute but makes no differentiation as to direction of flow. The Alnor velometer was calibrated in a wind tunnel constructed by the physics department especially for velocities in the range of 2000 to 3000 feet per minute, and found to have a constant error over the range of use as shown in Figure 7. All velocity readings were corrected for air density.

Originally it was intended to allow ten diameters of pipe to straighten out the air flow before taking velocity readings. Preliminary measurements showed the pattern of flow across the duct to be quite irregular under these conditions, although the same pattern could be obtained on successive days. In order to increase the precision of measurement, an egg-crate type of flow straightener composed of six pieces of twenty-four gage sheet metal three inches wide was introduced in the calibration duct. Velocity measurements were then taken one
Fig. 7 Correction chart for Alnor velometer
and one-half diameters downstream from the flow straightener. Although a flow straightener does not equalize flow velocity across a duct, it does serve to increase the precision of measurement (1). Using this apparatus and procedure, the air flow was measured at various valve openings on both the fresh and exhaust air pipes and the valve positions marked.

Readings of total pressure difference created by the fans were taken for various speeds and valve openings to check the power requirements. Static pressures were read by a liquid manometer connected to two diametrically opposite points on both the intake and exhaust ducts to the fans, as shown in Figure 9. Great care was exercised in attaching the static tubes to prevent disturbing the air flow in the ducts. The manometer consisted of a glass U tube of 5/8 inch diameter bore. Water was used as the gauge fluid.

Transfer of heat by conduction may occur in the steady state or the unsteady state. In steady state heat transfer the temperature at any point is independent of time whereas the temperature at a point varies with time in the unsteady state. The thermal capacity of materials causes an appreciable lag in the time required for steady state conditions to be reached after the temperatures on the two sides of the material become constant. This time lag may be several hours in some cases. It is obvious that steady state conditions are seldom, if ever, reached in a structure such as a dairy barn.
Fig. 8. Electronic potentiometer.

Fig. 9. Fans in hay mow.
so that calculation of heat losses through the structure at a particular instant can only be a close approximation. The degree of approximation can be reduced if it is possible to obtain temperature readings at a time when the inside and outside temperatures have been known to be constant for a considerable period. In order to accomplish this, insofar as it was possible, continuous readings of inside and outside temperatures and humidities were kept on recording hygro-thermographs. Continuous readings of temperatures through the heat exchanger and in the barn were taken by means of a Brown electronic potentiometer, Figure 8. This instrument will automatically record the temperature of sixteen thermocouple junctions every four minutes, if desired.

In order to obtain the average temperature of air flowing in a duct one foot in diameter, it was felt that at least three thermocouple junctions should be used at any cross-section. In this manner it was hoped that any errors due to eddy currents, stratification, etc., would be minimized. The potentiometer was set up in the equipment room of the barn which necessitated running lead wires thirty to fifty feet to various points on the heat exchanger. In order to avoid confusion and to conserve expensive waterproof leadwire, the three thermocouples at each cross-section were joined in parallel to a single leadwire. Thermocouples connected in parallel will indicate the true average temperature of the three so long as the lengths of wire to each junction are equal. The
Fig. 10. View of south inlet and outlet.

Fig. 11. View of ducts leading to fans.
resistance of the wires to circulating currents is then equal so cancels out and the true average emf is obtained (18). Accordingly, the length of wire from the junction with the leadwire to the copper-constantan junction was made equal in each case. The three junctions were placed at the center and at the quarter points of the ducts. Three other thermocouple junctions were placed in the barn to check temperature distribution from floor to ceiling.

B. Analysis of Results

1. Water vapor removal

The principal aim of ventilation in dairy barns has been stated to be the removal of water vapor. Although ventilation provides the main avenue, small amounts of moisture are also removed by air exfiltrating through walls and around doors and windows. The volume of ventilating air can be measured with reasonable accuracy, but the quantity of air infiltrating and exfiltrating requires a degree of estimation. Following are the estimated quantities of infiltration, based on the recommendations of the A.S.H.V.E. Guide (1) for 5 and 15 miles per hour winds:

Through walls

\[
\begin{align*}
5 \text{ mph} & \quad \frac{2 \times 1068}{2} = 1068 \text{ ft}^3 \text{ per hour} \\
15 \text{ mph} & \quad \frac{8 \times 1068}{2} = 4270 \text{ ft}^3 \text{ per hour}
\end{align*}
\]
Around windows

5 mph  27 x 55 = 1485 cfh
15 mph 111 x 55 = 6100 cfh

Around doors

5 mph  54 x 60 = 3180 cfh
15 mph 222 x 60 = 13,300 cfh

Total

5 mph  5733 cfh = \( \frac{5733}{60} \) = 90.6 c.f.m.
15 mph 23,680 = \( \frac{23670}{60} \) = 384 c.f.m.

During the course of the experiment, the quantity of fresh air brought into the building was always somewhat greater than the quantity exhausted. In this way an attempt was made to have a positive quantity of exfiltration when the outdoor wind velocity was five miles per hour or less and so to produce more reliable results for moisture removal.

The moisture removed was calculated from the difference in specific humidities and the total mass of air moved by means of the equation:

\[ w = M (h_1 - h_0) \]

where \( w \) = pounds of moisture per hour

\( M \) = total mass of air changed, lbs per hour

\( h_1 \) = specific humidity in the barn, lbs. moisture per pound of air.

\( h_0 \) = specific humidity outdoors, lbs moisture per pound of air.

The water vapor removed is given in Table 2 for the data.
Table 2

Water Vapor Removed

| Date  | Time    | Fresh Air |  | Exhaust Air |  | Baro.- Air |  | Water Removed |  | Total | Per cow |
|-------|---------|-----------|  | Temp. R.H. | # w.v./# air | Temp. R.H. | # w.v./# air |  | lb/hr. | lb./hr. |
| 1/19  | 5:30pm  | 16.0:66   | 0.00115 | 42.0:63 | 0.0036 | 29.3 | 5080 | 12.43 | 0.792 |
|       | 8:00pm  | 10.0:71   | 0.00091 | 40.0:63 | 0.00332 |  | 5150 | 12.62 | 0.804 |
| 1/20  | 3:30am  | 11.0:75   | 0.00101 | 45.0:75 | 0.00475 |  | 5140 | 19.25 | 1.226 |
|       | 7:30am  | 10.0:85   | 0.0011 | 39.0:62 | 0.0031 |  | 5150 | 10.3 | 0.656 |
|       | 3:00pm  | 38.0:50   | 0.0024 | 50.5:60 | 0.00465 |  | 4850 | 10.91 | 0.695 |
| 1/23  | 7:00pm  | 28.0:92   | 0.00289 | 50.0:72 | 0.00547 | 28.5 | 4785 | 12.35 | 0.787 |
| 1/24  | 3:00am  | 29.0:94   | 0.0031 | 52.0:74 | 0.00613 |  | 4780 | 14.5 | 0.9235 |
|       | 11:00am | 32.0:95   | 0.0036 | 53.0:76 | 0.0065 |  | 4740 | 13.75 | 0.876 |
|       | 7:00pm  | 18.5:78   | 0.00157 | 50.0:65 | 0.00496 |  | 4910 | 16.7 | 1.062 |
|       | 10:00pm | 12.5:68   | 0.00101 | 44.0:64 | 0.00394 |  | 4965 | 14.55 | 0.927 |
| 1/25  | 6:15pm  | 0.0:74    | 0.0006 | 36.0:67 | 0.00297 | 29.7 | 4900 | 11.6 | 0.74 |
|       | 6:45pm  | -5.0:73   | 0.0005 | 34.0:65 | 0.00267 |  | 4910 | 10.65 | 0.678 |
| 1/26  | 11:00am | -6.0:60   | 0.00028 | 38.0:75 | 0.00364 | 29.6 | 3870 | 13.0 | 0.828 |
|       | 4:20pm  | 1.0:51    | 0.00041 | 40.0:68 | 0.00357 |  | 3820 | 12.08 | 0.769 |
|       | 5:30pm  | -3.0:51   | 0.00032 | 37.0:64 | 0.00298 |  | 3850 | 10.23 | 0.652 |

Continued on next page
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<th># w.v./# air</th>
<th>Temp R.H</th>
<th># w.v./# air</th>
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</table>
calculated from the experiment. These data are shown in Figure 12 with moisture removed per 1250 lb. cow per hour plotted against temperature. The straight line variation found by statistical analysis is

\[ y = 0.0322 + 0.01712x \]

which compares quite favorably with the results of other investigators (see Figure 1). It should be remembered that this equation applies to total moisture production, not the moisture of respiration. The moisture of respiration will be somewhat less due to evaporation from the skin, urine and feces. Had more data been taken at temperatures above 50° there would probably have been evidence of curvilinearity.

The wide scattering of data is difficult to explain. The floor in this barn was not washed but was kept clean by applying about five lbs. of lime per day which might tend to give readings which are somewhat below the actual. There was no evidence of condensation on the walls when data were taken but there was usually some condensation on the windows. This would tend to reduce the values obtained but should give no significant fluctuation. The wide variation in data must, then, be attributed to unsteady state conditions at the times when readings were taken. The results indicate a significant increase in moisture production with increasing environmental temperature; the actual values are probably slightly low.

The ventilation rate required to remove the moisture produced may be calculated for any set of inside and outside
Fig. 12 Moisture production in dairy cattle.

\[ Y = 0.0322 + 0.01712 \times X \]
conditions when values for the moisture production have been established. Figure 13 shows the ventilation rate required to remove the moisture produced by a 1250 pound cow at barn temperatures of 40°F and 50°F with varying outdoor temperatures. The indoor relative humidity was assumed at 70 per cent, outdoor at 80 per cent. Since the ventilation rate required to remove a given quantity of moisture varies with the difference in the specific humidities of the entering and leaving air, a higher rate of ventilation is required when the barn temperature is 40°F than when the barn temperature is 50°F. This is true even though the quantity of moisture to be removed is less at 40°F, as is shown by the curves. Choice of a ventilation rate is, then, related to both indoor and outdoor temperatures so that it is practically impossible to provide the exact quantity of ventilation to remove moisture under all conditions. From a practical standpoint, it would be desirable to have a maximum of four positions for the valves used to vary the quantity of flow. At these positions the quantities should be approximately 38, 45, 50 and 58 cubic feet per minute per cow. During the course of the experiment the flow was varied over this approximate range but a total of ten valve positions used. This is unnecessary for anything but experimental work for if the quantity of ventilation is insufficient to remove the moisture produced, the humidity will build up slightly until a balance is obtained.
ASSUMPTIONS:

MOISTURE PRODUCED: 50° = 0.87
40° = 0.72

RELATIVE HUMIDITY:
OUTDOOR 80%
INDOOR 70%

Fig. 13 Ventilation required to remove moisture.
2. Temperature differentials

A complete summary of the temperature readings taken is contained in Table 3. In this table, $t_{f1}$ refers to the temperature of the fresh air as it enters the tubes, $t_{f2}$ as it leaves the tubes, $t_{f3}$ after passing through the fan and $t_{f4}$ as the fresh air leaves the fan end of the outer duct. Similarly $t_{x1}$ and $t_{x2}$ refer to the temperatures of the exhaust air as it enters the annular space and leaves the annular space, respectively. A brief glance at the data shows that the fresh air was always warmed a few more degrees than the exhaust air was cooled. Figure 14 shows this quite clearly. The curves in this figure were not arrived at by statistical analysis but were drawn to indicate the trend, rather than produce exact values. The discrepancy in temperature differentials obtained is readily explained by the fact that the exhaust air has a much higher moisture content per pound of air. Therefore, more heat is released per pound of exhaust air for a specified drop in temperature than is required to heat an equal weight of fresh air through the same temperature difference. The high moisture content produces a very pronounced effect as the dew-point is approached. Condensation of the moisture from the exhaust air liberates the heat of vaporization so the air is cooled at a much slower rate per unit of heat extracted. This accounts for the rapid flattening of the exhaust curve as a temperature differential of $15^\circ$ is approached. It is gratifying to find that air at 65 to 70
Table 3

Temperature Readings

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<tr>
<th>Date</th>
<th>Time</th>
<th>Outside</th>
<th>Barn</th>
<th>Fresh Air Temps.</th>
<th>Ex. Temp.</th>
<th>Heat Transferred</th>
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<td>Btu/min.</td>
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<td>To Fresh</td>
</tr>
<tr>
<td></td>
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<th>Barn 1</th>
<th>Barn 2</th>
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<th>Mf</th>
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<td>T3</td>
<td>T4</td>
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### Table 3 (continued)

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<th>Ex. Temp.</th>
<th>M_F</th>
<th>M_X</th>
<th>Heat Transferred Btu/min</th>
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<tr>
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<td>9:00pm</td>
<td>15.0% 72.0%</td>
<td>46%</td>
<td>65% 18.0%</td>
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<td>32.0% 80.0%</td>
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<td>65% 33%</td>
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<td>40.7% 43.0%</td>
<td>49.5% 40.0%</td>
<td>80.9% 55.4</td>
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</table>
Fig. 14. Temperature differentials

Indoor to Outdoor Temperature Difference — Deg. F.
per cent relative humidity must be cooled approximately 12 degrees to reach the dew-point temperature. Further inspection of Figure 14 reveals that the fresh air was usually warmed to within 12 degrees of the indoor temperature so that there was little tendency for condensation to occur on the outer duct. Visual observation substantiates this; only once during the course of the experiment was condensation known to have occurred on the outer duct.

Heat gain from the surrounding air is another reason that the increase in fresh air temperature is always greater than the decrease in exhaust air temperature. As the exhaust air moved through the annular space it was usually cooled to a slightly lower temperature than the air in the outer duct. In this way heat was added to the exhaust air from the air in the outer duct. That is, the air in the outer duct gained heat from the surroundings and passed a small portion of it on to the exhaust air. This had the effect of decreasing the efficiency of heat salvage.

3. Heat salvage

The values of heat gained by the fresh air and lost by the exhaust air for the data calculated from the experiment are shown in Table 3. These values were calculated from the equation:

\[ H = M(h_1 - h_2) \]
where $H =$ heat transferred, Btu/minute

$M =$ mass of ventilating air, lb./minute

$h_1 \& h_2 =$ enthalpies of air entering and leaving the exchanger, Btu/lb.

The enthalpy values were read from psychrometric charts. All values of air quantity were corrected for temperature and barometric pressure. The calculations for heat lost by the exhaust air for the run of January 19 at 5:30 pm will be shown as an example.

Ventilation rate = 640 cubic ft. per minute

Entrance condition = 44.7°, 58 per cent R.H.

Exit condition = 29.5°, 100 per cent R.H.

Barometric press. = 29.30 inches of mercury

Specific volume at 44.7°, 58 per cent and 29.92 inches of mercury = 12.72 cu. ft. per lb.

Therefore, weight of air moving

$$= \frac{640}{12.72} \times \frac{29.3}{29.92}$$

= 49.3 lb/minute

Enthalpy at entrance = 14.6 Btu per lb.

Enthalpy at exit = 10.75 Btu per lb.

Therefore, the heat given up is

$$H = 49.3(14.6 - 10.75)$$

= 189.5 Btu per minute

The procedure for calculating the heat gained by the fresh air was the same.

The heat gained by the fresh air from the exhaust air
must be transferred through the walls of the pipes. The rate at which it is transferred may be expressed as

\[ H = U \times A \times \Theta_{\text{mean}} \]

where \( U \) = overall coefficient of heat transfer
\( A \) = area of the pipe surfaces
\( \Theta_{\text{mean}} \) = logarithmic mean temperature difference.

As stated previously, use of the logarithmic mean does not give the true mean temperature difference when the heat transfer coefficient is not constant, but according to Ibrahim (12) this method gives a close approximation. The value of \( U \) is the only unknown in the equation for a particular run of the experiment and hence can be found. For the run of January 19 at 5:30 pm, the heat gained by the fresh air was found to be 249.5 Btu per minute by the method outlined above. The surface area of 16 three-inch diameter pipes 31 feet long is:

\[ 16 \pi \times 0.25 \times 31 = 390 \text{ sq. ft.} \]

\( \Theta_{\text{mean}} = \frac{\Theta_A - \Theta_B}{\ln \frac{\Theta_A}{\Theta_B}} \)

\( \Theta_A = 44.7 - 29.2 = 15.5 \)
\( \Theta_B = 29.5 - 16.5 = 13.0 \)

\( \Theta_{\text{mean}} = \frac{15.5 - 13.0}{\ln \frac{15.5}{13.0}} = 14.2^\circ F \)

The theoretical value of \( U \) has been expressed as

\[ U = \frac{1}{\frac{1}{h_0} + \frac{1}{h_1}} \]
where \( h_o \) and \( h_i \) are the outside and inside surface coefficients of heat transfer, respectively. The empirical equation given for calculating \( h_o \) and \( h_i \) was

\[
h = 0.0036 \frac{G^{0.8}}{D^{0.2}}
\]

where \( G \) is the fluid mass velocity in pounds per hour per square foot of cross-section and \( D \) is the characteristic dimension of the conduit. The theoretical value of \( U \) for the run of January 19 at 5:30 pm will now be found:

In the pipes
\[
G = \frac{1035}{0.785} \times \frac{60}{12.0} \times \frac{29.3}{29.92} = 6450 \text{ lb/(hr)(sq.ft.)}
\]

\( D = 0.25 \text{ feet} \)

Therefore \( h_i = 0.0036 \times 6450^{0.8} \frac{.25}{.2} = 5.31 \text{ Btu/(hr)(sq.ft.)(°F)} \)

In the annular space
\[
G = \frac{640}{0.98} \times \frac{60}{12.72} \times \frac{29.3}{29.92} = 3010 \text{ lb/(hr)(sq.ft.)}
\]

\( D_e = \frac{4A}{P} = \frac{4 \times 0.98}{\pi \times 0.25 \times 16} = 0.312 \text{ feet} \)

Therefore \( h_o = 0.0036 \times 3010^{0.8} \frac{.312}{.2} = 2.77 \text{ Btu/(hr)(sq.ft.)(°F)} \)

\[
U_t = \frac{1}{\frac{1}{h_o} + \frac{1}{h_i}} = \frac{1}{\frac{1}{2.77} + \frac{1}{5.31}}
\]
The values of the heat transfer coefficients calculated from the experiment are given in Table 4. These data are shown graphically in Figure 15 with $U$ plotted against log mean temperature difference for three values of average velocity.

A glance at Figure 15 shows that the actual coefficient of heat transfer was always greater than the theoretical. Closer inspection reveals that the theoretical values are practically constant while the actual values found from the experiment have a definite tendency to increase with increasing values of temperature difference. Turbulence induced by joints and by supports could account for higher values of the heat transfer coefficient. The tendency to increase with increasing temperature difference is more likely to be due to condensation. As the temperature difference increases, the amount of condensation increases and the proportion of the area on which condensation is occurring increases. The film of moisture on the surface could quite conceivably reduce the rate of heat transfer but the latent heat liberated during condensation apparently increases the coefficient due to the method of calculation. One of the assumptions on which the derivation of the logarithmic mean temperature difference is based is that the coefficient of heat transfer is constant. Use of this method of determining the heat temperature for cases in which the rate of heat transfer increases due to
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<tr>
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<td>15.0</td>
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Fig. 15 Co-efficients of heat transfer.
condensation yields values of mean temperature difference which are somewhat higher than is actually true. The coefficients of heat transfer found by this method will therefore be slightly below the true values.

When this method of calculation is used in designing a similar type of system it appears that heat transfer could be expected to occur at rates 20 per cent greater than the theoretical when no condensation was expected to occur and at least 50 per cent greater when condensation was expected to occur in quantity.

The method of calculating the heat salvaged from the exhaust air has been outlined and a sample calculation shown. The values calculated from the experiment are given in the last column of Table 4. These values are shown plotted against indoor to outdoor temperature difference in Figure 16, for three values of average velocity. If the coefficient of heat transfer had been constant, the values of heat salvage would have varied linearly with temperature difference according to the equation \( H = UA \Delta t \). However, the value of \( U \) has been shown to increase rather uniformly with increasing temperature difference over the range of values tested. Therefore, the heat salvage should not vary linearly with temperature difference but should form a second degree curve. The data plotted in Figure 16 show this relationship. It is also evident that increasing the velocity shifts the curve to the right causing higher values of heat salvage to occur at lower differences of indoor to outdoor temperature. This can be explained by the increase in rate of heat trans-
Fig. 16 Heat salvaged by exchanger.
fer that occurs with increase in velocity.

Figure 17 indicates the heat that would have been lost at these same values of air velocity and temperature difference had the air been exhausted directly out of the barn without passing through the exchanger. These are practically straight line relationships. If per cent of heat salvaged is plotted against temperature difference by combining the two sets of curves, a set of parabolic curves results as shown in Figure 18. Here again there is a noticeable tendency for the curves to shift to the right as the velocity of the air in the exchanger is increased. This too can be attributed to the increase in rate of heat transfer that occurs with increase in velocity. The lowest values of per cent of heat salvaged occurred at indoor to outdoor temperature differences of 20 to 30 degrees and the highest values occurred with temperature differences in excess of 30 degrees. It appears therefore that in order to obtain high values of per cent of heat salvage it is necessary to have high velocities and high indoor to outdoor temperature differences. It was unfortunate that it was impossible to maintain temperature differences of more than 40 degrees with these quantities of air flow so that the trend could be checked. Unfortunately, the heat losses through the structure by conduction and radiation were too great to permit this. However, it does seem safe to assume that values of heat salvage well in excess of 50 per cent could be obtained in barns of better construction.
Fig. 17 Heat loss to ventilation without exchanger.
Fig. 18 Percent of heat saved by exchanger
It should be pointed out at this time that it is impossible to salvage all the heat in the exhaust air. This can best be illustrated by the following example. Let us assume that the air in the barn is at a temperature of 50 degrees and 70 per cent relative humidity and that the outside air is at 0 degrees and 100 per cent relative humidity. It requires 11.9 Btu to warm one pound of the outside air to 50 degrees but if one pound of the inside air is to be cooled to 0 degrees, 16.8 Btu must be extracted. Therefore, if equal weights of inside and outside air are passed through the exchanger, as is desirable, only $\frac{11.9}{11.8} \times 100 = 71$ per cent of the heat may be salvaged from the exhaust air even if the fresh air is warmed to 50 degrees. The ratio is practically the same for other conditions of temperature. From this it can be seen that when 50 per cent of the heat was salvaged the actual efficiency of salvage was over 70 per cent.

4. Heat balances

In Figure 13 on page 50, the quantities of ventilation required to remove the water vapor produced by a 1250 pound cow at barn temperatures of 40 degrees and 50 degrees are shown for varying outdoor temperatures. The heat required to warm these quantities of ventilating air may be calculated. If a value of building construction is estimated the total heat loss per cow can be determined for any desired conditions of outdoor temperature. This was done for barn temperatures
of 40 degrees and 50 degrees, and the results are shown graphically in Figures 19 and 20. The assumptions on which these graphs were based are stated on them. It should be noticed that no allowance was made for infiltration. The letters AC refer to the total building heat loss in Btu per cow per degree difference of indoor to outdoor temperature. The letter A may be thought of as the area of walls, doors, windows and ceiling exposed per cow and C as the average weighted heat transfer coefficient. Inspection of Figures 19 and 20 reveals that if sufficient ventilation is provided, a cow only produces enough heat to maintain a building of AC value equal to 20 to 50 degrees when the outdoor temperature is above 12 degrees. If the barn temperature is allowed to drop to 40 degrees, the balance occurs at an outdoor temperature of 6 degrees. If an AC value of 30 is taken for the building characteristic, the balance comes at 20 degrees and 12 degrees outdoor temperatures for barn temperatures of 50 degrees and 40 degrees, respectively.

The heat loss and gain factors for a barn equipped with a heat exchanger similar to the one used in the experiment are summarized in Tables 5 and 6 for two values of indoor temperature. These tables are presented graphically in Figures 21 and 22. It will be noticed that account was taken of infiltration in these figures. The infiltration rate assumed was the same as the value found for the experimental barn with a 5 mph wind. Since this barn is of poor construction, the same
Fig. 19 Dairy barn heat balance at 40° without heat exchanger.
Fig. 20 Dairy barn heat balance at 50° without heat exchanger.
Table 5

Dairy Barn Heat Balance with Exchanger

Inside conditions: 50 degrees and 70% R.H.
Outside conditions: variable temperature, 80% R.H.
Ventilation to remove 0.87 lb. moisture per hour

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<th>Outside temperature-deg. F</th>
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<td>Total heat production</td>
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<td>Loss to ventilation-no exchanger</td>
<td>2405 : 2675 : 2835 : 3110 : 3430 : 3750</td>
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<tr>
<td>Building heat loss - AC 20</td>
<td>400 : 600 : 800 : 1000 : 1200 : 1400</td>
</tr>
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Table 6

Dairy Barn Heat Balance with Exchanger

Inside conditions: 40 degrees and 70% R.H.
Outside conditions: variable temperature, 80% R.H.
Ventilation to remove 0.72 lb. moisture per hour

<table>
<thead>
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<th>Heat Factors - Btu/cow hour</th>
<th>Outside temperature-deg. F</th>
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</thead>
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<tr>
<td></td>
<td>30 : 20 : 10 : 0 : -10 : -20</td>
</tr>
<tr>
<td>Total heat production</td>
<td>3500 : 3500 : 3500 : 3500 : 3500 : 3500</td>
</tr>
<tr>
<td>Loss to ventilation-no exchanger</td>
<td>2483 : 2464 : 2727 : 3010 : 3400 : 3770</td>
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<tr>
<td>Gain from exchanger</td>
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<tr>
<td>Building heat loss - AC 20</td>
<td>200 : 400 : 600 : 800 : 1000 : 1200</td>
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</table>
Fig. 21 Dairy barn heat balance at 40°F with heat exchanger.
Fig. 22 Dairy barn heat balance at 50°F with heat exchanger.
values might very well apply to a 10 mph wind on a barn of better construction. From the figures it is seen that a barn of AC value 20, equipped with a heat exchanger, may be maintained at 50 degrees so long as the outdoor temperature does not fall below -10.5 degrees. That is, use of the exchanger would allow sufficient ventilation to remove moisture when the outdoor temperature is 23 degrees lower than could occur with no exchanger. If the AC value is 30, the allowable outdoor temperature is lowered $16\frac{1}{2}$ degrees by use of the exchanger. If the barn temperature is allowed to drop to 40 degrees, use of the exchanger produces a heat balance at -13 degrees in a barn of AC 20 and at -3 degrees in a barn of AC 30. In this case the allowable outdoor temperature has effectively been lowered by 19 degrees and 12 degrees for barns of AC values of 20 and 30, respectively.

5. Fan performance

It becomes obvious that the demand upon a heat exchanger increases as the outdoor temperature falls below a certain value but that above this value no means of saving heat is necessary. It has also been shown that a heat exchanger operates more effectively when the velocity of the air passing through it is high. However, the volume of air required to remove moisture decreases as the outdoor temperature decreases. If all the air is passed through the exchanger at all times, the highest velocities will occur when there is no necessity
for saving heat while the velocities will be very low when the demand for heat salvage is greatest. The fans used must be of a size such that they will handle the large quantities of air against considerable pressure. When small quantities of air are desired, dampers or valves are the only feasible way of reducing the flow and these incur high resistance pressures. Therefore, the power requirements are always high. It would seem desirable to size the fans and exchanger system so that high velocities are produced when the quantity of flow is small. As larger quantities are required, the heat exchanger could be gradually short circuited until, when there was no need for heat salvage, the air could pass directly in and out of the barn without any of it passing through the exchanger. Proper selection of fans and duct sizes might produce the desirable conditions over a wider range of temperatures at a much lower cost for both power and equipment.

A centrifugal fan operated at constant speed exhibits a curve of performance which is often presented by plotting static pressure against volume of air delivered. This curve is determined by measuring the static pressure and volume of delivery at several valve positions varying from fully open to fully closed while the speed of rotation is held constant. A change in position of the valve controlling the flow produces a change in resistance and the fan then operates at a different point of rating on the performance curve. If the curve of system resistance is desired for a particular valve
position, readings of static pressure and quantity are taken at several speeds. This system resistance curve will be found to vary almost directly as the square of the quantity and so will be parabolic in shape.

The performance curves of both fans used for the experiment were found for three speeds of rotation by the method described. The fans were then attached to their respective duct systems and the system resistance curves found when air was flowing through the exchanger and when the exchanger was short circuited. The resulting calibration charts are shown in Figures 23 and 24. Corrections for slight variations in fan speed were made by the well-known laws of homologous fans and all variations in air density were corrected to 0.749 pounds per cubic foot. A third system resistance curve was drawn for the fresh air fan. Fresh air was ordinarily distributed in the barn by means of the slotted outer duct and this would also normally be the case when air flow was short-circuited. However, this outer duct was found to produce a very high resistance, probably due to the manner in which the pipe from the fan was attached to it. A curve of system resistance was therefore found with the air passing directly into the barn.

When the outdoor temperature is zero degrees Figure 13 shows that between 40 and 50 cubic feet per minute of ventilation are required per cow with barn temperatures of 40 to 50 degrees F. For the 15.7 equivalent cows used in the
Fig. 23 Calibration curves for exhaust fan.

- Short-circuit resistance
- Air density = 0.0749 \#/ft^3

Air quantity — cubic feet per minute

Static pressure — inches of water

RPM values: 930, 1350, 1500, 1620
Fig. 24. Calibration curves for fresh air fan.
experiment, the total quantity of ventilation should be approximately 700 cubic feet per minute. The charts show that this quantity could be forced through the exchanger with the fans operating at a speed of approximately 950 rpm. If the fans are short-circuited at this speed to provide extra flow, approximately 1100 cfm would be delivered which is sufficient for outdoor temperatures up to 30 degrees. For temperatures between these values, the exchanger could be partially short circuited and some of the heat salvaged from the exhaust air. It seems that short circuiting could provide a satisfactory means of varying the quantity of flow.

The alternative to short circuiting is to operate the fan at higher speed and reduce the air quantity in cold weather by means of dampers or slide valves. Referring again to Figures 23 and 24 it can be seen that a speed of approximately 1350 rpm is necessary to move 1100 cfm through the exchanger and that the static pressure is more than doubled. It would seem that short circuiting requires less than half the power to produce the same quantity of flow. The horsepower output of a fan can be expressed by the relation:

\[ \text{A.H.P.} = \frac{\text{cfm} \times \text{ht}}{6356} \]

where A.H.P. = the air horsepower
cfm = cubic feet of air delivered per minute
ht = total pressure in inches of water = static pressure plus velocity pressure
The horsepower input to the fan is then the air horsepower divided by the mechanical efficiency. Assuming an efficiency of 60 per cent, calculations were made for a few points on the system resistance curves and the horsepower curves of Figure 25 were drawn. These curves show that to push 1100 cubic feet per minute through the exchanger equipped with the outer duct requires 0.595 hp. whereas only 0.278 horsepower is required if the exchanger is short circuited. That is, 2.14 times as much power is required for ventilation during mild weather when the air is passed through the exchanger as would be required if the exchanger were short circuited.

The intermediate curve was drawn in to show that addition of an outer duct around the system increased the horsepower requirements by 22\% per cent. This value could probably be reduced by use of a smooth transition piece where the outer duct is lead-off to the fan.

An alternative to the short-circuiting method of producing additional flow during periods of mild weather is to use an auxiliary fan. A conventional propeller type fan mounted in the wall and controlled by a thermostat proved quite effective in maintaining constant temperature and low humidity during periods of mild outdoor weather. With this auxiliary available it would be possible to design the heat exchanger and the fans to operate it for a minimum quantity of flow. The exchanger would be in operation at all times and the auxiliary fan would come on to provide extra flow when the indoor
temperature rose above a predetermined level. Referring once again to Figure 25, it may be seen that 0.19 horsepower are required to force 700 cfm through the heat exchanger. When the air flow is short-circuited, 0.278 horsepower are required to move 1100 cubic feet per minute. An increase in quantity of 400 cubic feet per minute required an additional 0.088 horsepower. Exact measurement of the air quantity could not be obtained but the auxiliary fan appeared to handle approximately 1200 cubic feet per minute for slightly less than 0.15 horsepower. As this fan ran intermittently it was handling air for alittle over half the cost.

Evaluation of these two methods of producing additional flow in mild weather involves consideration of the initial cost. A fan of the type and size used costs approximately $60.00 plus additional charges for installation. An on-off type damper control costs approximately $20.00 and the sheet metal work required would possibly cost another $10.00. A modulating type of damper control would be more desirable and more expensive but at present there are none on the market which are satisfactory for use under the conditions of dust and exposure found in dairy barns. An on-off type should give reasonable satisfaction due to the high thermal capacity of the structure. Since a thermostat would be required in both cases it may be neglected in the comparison. It appears, then, that the auxiliary fan might cost approximately $30.00 more initially but would operate for about three cents per day.
Fig. 25: Power required to drive fans.

- AIR THROUGH EXCHANGER & OUTER DUCT
- AIR THROUGH EXCHANGER ONLY
- AIR SHORT-CIRCUITED
while the other system would produce additional flow for about six cents per day. Additional flow might be required fifty days per year so that the difference in yearly charge for electricity might be in the order of two dollars. It would appear that the short-circuiting method of producing additional flow shows up to advantage in both the short and long term analysis. Modifications in duct design and fan placing would reduce the cost of operating the fans on short-circuit to a much lower figure.

6. Cost analysis

Any attempt to arrive at a figure for the cost of salvaging heat with a heat exchanger such as the one used for this experiment must of necessity be highly arbitrary. The initial cost of experimental models is always excessive so that choice of a value for the fixed charges against the equipment involves a high degree of estimation. A heat exchanger very similar to the one here reported, cost approximately $1500.00 installed in the barn. Almost 2/3 of this was paid for the labor required in fabricating and installing the unit. If it were possible to produce in quantity, it seems reasonable to assume that the total initial cost would not exceed $1100.00. If the equipment is depreciated over a ten-year period with interest at five per cent, the annual cost is $142.50. In a climate where the equipment was required to operate 100 days per year the hourly charge would be 5.93
cents. In a climate such that the exchanger would be required
150 days per year the hourly charge to equipment would be
3.96 cents.

It has been shown that the amount of heat an exchanger
such as this is able to salvage is highly dependent on the
indoor to outdoor temperature difference and on the velocity
of the air through the unit. In Table 7 an attempt has been
made to relate the cost of salvaging heat to the indoor to
outdoor temperature difference when the air quantities required
for moisture removal are passed through the exchanger. This
was done for fixed charges of both 5.93 cents per hour and
3.96 cents per hour. The costs may be seen to vary from $2.63
to $13.60 per million Btu when a 100 day heating season is
used and from $1.83 to $10.56 when a 150 day heating season is
used as a basis for determining fixed charges.

Considering no charge to equipment the cost of producing
heat by other means is approximately:

Electricity at 3¢ per kwh...$8.80/million Btu
Coal at $12.00 per ton......$1.35/million Btu
Furnace oil at 15¢ per gal..$1.65/million Btu

Resistance heaters might be installed for a very nominal cost
but use of any type of furnace would require considerable
original investment. In any case fans would be required for
ventilation.

In central Iowa the temperature averages 23.6 degrees
over a 100 day heating season. If the barn is kept slightly
Table 7

Cost of Salvaging Heat in Relation to Indoor-to-Outdoor Temperature Difference

<table>
<thead>
<tr>
<th>Temp. Diff.</th>
<th>Air Volume (cfm)</th>
<th>Salvage (b/hr.)</th>
<th>Hp. req'd @3¢</th>
<th>Power Costs: Total</th>
<th>100 day season Costs: Total</th>
<th>Cost per 10^6 Btu</th>
<th>150 day season Costs: Total</th>
<th>Cost per 10^6 Btu</th>
</tr>
</thead>
<tbody>
<tr>
<td>60°</td>
<td>635</td>
<td>24,600</td>
<td>0.145</td>
<td>5.93¢</td>
<td>6.47¢</td>
<td>$2.63</td>
<td>$3.96¢</td>
<td>$1.83</td>
</tr>
<tr>
<td>50°</td>
<td>675</td>
<td>21,350</td>
<td>0.17</td>
<td>6.56¢</td>
<td>$3.07</td>
<td>&quot;</td>
<td>&quot;</td>
<td>$1.15</td>
</tr>
<tr>
<td>40°</td>
<td>740</td>
<td>17,250</td>
<td>0.217</td>
<td>7.14¢</td>
<td>$3.91</td>
<td>&quot;</td>
<td>&quot;</td>
<td>$1.77</td>
</tr>
<tr>
<td>30°</td>
<td>870</td>
<td>12,800</td>
<td>0.320</td>
<td>7.16¢</td>
<td>$5.59</td>
<td>&quot;</td>
<td>&quot;</td>
<td>$1.19</td>
</tr>
<tr>
<td>20°</td>
<td>1200</td>
<td>6,520</td>
<td>0.785</td>
<td>8.85¢</td>
<td>$13.60</td>
<td>&quot;</td>
<td>&quot;</td>
<td>$10.56</td>
</tr>
</tbody>
</table>
above 50 degrees the cost of salvaging heat in this area would
be approximately $5.60 per million Btu which places the heat
exchanger in a very favorable position economically. In the
vicinity of the author's home the average temperature for 100
days in mid-winter is 4.7 degrees which would reduce the cost
of salvaging heat to slightly over $3.00 per million Btu.
For a 150 day heating season the average temperature is 10.8
degrees in the region which would reduce the cost of saving
heat to $2.75 per million Btu. Both of these values suggest
that a heat exchanger is an extremely efficient means of keep-
ing dairy barns warm in winter. The comparison between Iowa
and Saskatchewan climates points out the fact that a heat ex-
changer becomes more valuable as the climate becomes colder
and suggests that as climatic conditions become milder, a
point will be reached where it is more economical to use elec-
tricity or some other method for auxiliary heating.

It was suggested in a previous section that future ex-
changers be designed with smaller cross-sectional areas so
that high velocities could be used in the exchanger when the
need for heat salvage was greatest. Reduction of the cross-
sectional area increases the velocity for a particular quan-
tity of flow but it also decreases the total surface area of
heat transfer if the same size of pipes is used. Analytical
investigation indicates that the increased heat salvage due
to increased velocity is almost exactly compensated for by
the loss in heat salvage incident to reducing the surface.
area. If more surface is provided by the use of smaller pipes or some such method, the power required to drive the fans will be increased. The cost of providing power has been shown to be a very small proportion of the total operating cost for velocities below 100 feet per minute. It would, therefore, seem that a more economical relation of cost of operation to heat salvaged might result from use of higher velocities in conduits of smaller characteristic dimension.

Addition of the outer duct enclosing the main heat exchanger system is apparently well justified. When this duct was used to distribute the fresh air, the additional heat transfer due to its use more than offset the loss to the surrounding air which insulation prevented in the experiments of Ibrahim (12). Actual quantitative comparison with Ibrahim's work is complicated by discrepancies in the number of cows used for the experiment and therefore the quantities of air circulated. However, use of the outer duct appeared to increase the per cent of heat salvaged by 15 to 20 per cent. Had the duct been used to pick up exhaust air, the heat salvage would probably have been even greater. Condensation did not occur on the outer duct at any time during the experiment, although it might under greater differences of indoor to outdoor temperature. Use of the outer duct to pick up exhaust air would be to advantage from this standpoint also. When counter-flow of the air in the exchanger is used, there is a tendency for the fresh air leaving the exchanger to be short-circuited.
directly back into the exhaust pick-up unless ductwork is pro-
vided to distribute the fresh air to the other side of the
barn. The outer duct used in the experiment, therefore, effec-
tively replaced both the insulation and a duct to distribute
fresh air with little or no increase in investment. Power re-
quired for operation of the system was increased 21\% per cent
by addition of the outer duct. This figure could be reduced
to 10 per cent or less by means of a well-designed transition
piece. Since the power cost is a small fraction of the total
cost of operation, an increase of 10 per cent in it is easily
compensated for by the increased heat saving incident to use
of the outer duct.
V. SUMMARY AND CONCLUSIONS

1. Moisture production in cattle increases as the environmental temperature increases. At 40 degrees F., a 1250 pound cow produces a total of approximately 0.72 pounds of moisture per hour. This value increases to 0.87 pounds per hour at a temperature of 50 degrees and rises more steeply as the environmental temperature rises above 50 degrees. The moisture produced in respiration is somewhat less than these values due to evaporation from the gutter.

2. When a multiple tube heat exchanger is used to conserve the ventilation heat loss from dairy barns, the temperature of the fresh air entering the barn is raised more than the temperature of the exhaust air is lowered. The discrepancy in temperature differentials obtained is due to the heat of vaporization in the moisture the exhaust air contains in excess of that contained by the fresh air. As a consequence, it is impossible to salvage all the heat in the exhaust air. Any gain of heat from the ambient stable air serves to aggravate the situation.

3. The overall coefficient of heat transfer found in the experiment appeared to be 20 per cent greater than the theoretical values when there was no evidence of condensation. This increase may be attributed to turbulence induced by joints in the pipes and by supports. Increase in the
logarithmic mean temperature difference due to lower outdoor temperatures produced condensation. This condensation increased the apparent coefficient of heat transfer to values which exceeded the theoretical by more than 50 per cent. The per cent increase varied linearly with logarithmic mean temperature differences.

4. An outer duct enclosing a simple, multiple tube heat exchanger adequately replaces insulation around the system when it is used to distribute the fresh air. If this duct had been used to pick up the air to be exhausted it might have been even more effective in reducing heat gain from the surroundings. The increased power requirements incident to its use were more than compensated for by the additional heat transferred.

5. The per cent of heat salvaged from the exhaust air is not constant for constant values of air velocity in the exchanger. It apparently varies with both temperature difference and air velocity. The lowest values of per cent heat salvaged occurred with indoor-to-outdoor temperature differences in the 25 to 30 degree range for the velocities checked.

6. At least 50 per cent of the heat in the exhaust air can be salvaged under conditions of high velocity and high indoor-to-outdoor temperature difference.

7. Cattle do not produce sufficient sensible heat to maintain temperatures of 40 to 50 degrees in barns of ordinary
construction when the outdoor temperature falls to near zero if sufficient ventilation is provided to remove the moisture produced. Use of a heat exchanger of the type here reported can effectively lower the critical outdoor temperature by 25 degrees or more, depending on the values of building construction and indoor temperature.

8. Short-circuiting the air flow directly into and out of the barn increased the quantity delivered by approximately 45 per cent at any speed of operation. This could be increased considerably more by proper selection of fans and by increased design of ductwork. Static pressure losses in the heat exchanger proper accounted for only 70 per cent of the total.

9. Short-circuiting appears to be a more economical method of producing additional flow during periods of mild weather than providing an auxiliary fan.

10. Horsepower required to drive the fans varied as the 2.6 power of the quantity of air flowing through the exchanger.

11. The cost of salvaging a unit quantity of heat with an exchanger decreases as the average indoor-to-outdoor temperature difference increases. Economically, a heat exchanger can compete quite favorably with other methods of providing heat in dairy barns under the climatic conditions found in Iowa. In the states to the north and west, or in central Canada it would provide a very inexpensive means of maintaining comfortable temperatures in dairy barns while providing sufficient ventilation to remove the moisture produced.
VI. LITERATURE CITED


VII. ACKNOWLEDGMENTS

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