Heat gain from natural sources in the ventilation of animal shelters

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HEAT GAIN FROM NATURAL SOURCES IN THE VENTILATION OF ANIMAL SHELTERS

by

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INTRODUCTION

In cold regions where the temperature falls very low for a considerable time, farmers face the problem of keeping the livestock buildings warm and at a certain range of temperature. Even when they succeed in preserving a certain range, sudden fluctuation of the temperature affect greatly the health and production of the animal or bird.

Ventilation is found to be an important consideration in the maintenance of the health of stock and in the preservation of feed kept in the shelter and house timbers. It is essential to remove the moisture, keep air in motion, furnish fresh air to replace the oxygen consumed in addition to the removal of carbon dioxide, toxic materials and odor. It is highly desirable to maintain a comfortable shelter temperature with a proportionately low relative humidity.

Restricted ventilation is followed during cold waves although full ventilation is essential for keeping good environmental conditions. Ventilation is restricted because the coming cold air needs a great amount of heat to be warmed to the house temperature and so causes the temperature of the house to fall.

The main source of heat in a livestock building is
usually that from the animal except in brooder houses for small chicks or small pigs where an artificial source of heat is likely to be used.

In this study, two ways of gaining heat are discussed which are considered to be cheap and available for each farmer.

A heat exchanger is used to warm the coming cold air by the warm exhaust air during the process of ventilation. Also heat is gained by the cold air from the warm air inside the building. Amount of heat gained depends mainly upon the quantity of air to be heated, the surface areas of pipes in the heat exchanger, the difference in temperature between the inside and outside air and the relative humidity of inside air.

The heat exchanger consists of an outside duct in which there are small pipes. Warm air flows through these small pipes while cold air runs in the space between the outside duct and the inside pipes.

This heat gained will help during cold waves in raising the temperature of the inside making possible for air to carry more moisture in ventilation which will decrease the condensation of vapor moisture on walls and roof.

Another point is discussed in this study. This point is always neglected in calculation of heat gained and heat lost in a livestock building. Soil under the floor has been
considered as a source of heat in winter and a way of losing heat in summer. The amount of heat gained or lost depends upon the temperature in the house and the material used in making the floor. A good conducting material such as concrete is preferable.

Insulation of soil encountered within the foundation walls is suggested so that heat under the floor will not be dissipated to the cold neighboring soil and will be directed toward warming the floor. Insulation placed on the inner or warm side is more efficient than if placed on the outer side. The object in the use of insulation is to stop the flow of heat outward, and the sooner the heat flow is stopped the greater the conservation.
The early pioneers built log houses for themselves while animals were left without shelter. Later barns were constructed as a rough protection against the extremes of a rigorous climate. These constructions were not perfectly made and there was much leakage of air through windows, doors and cracks of walls. This kind of construction did not prevent wide fluctuation of temperature inside the building during the winter. Realizing that this fluctuation and low temperature affected the production of the animal and consumption of food, the farmer tried to tighten the building against air leakage. Immediately, he was faced with the problem of moisture increase and condensation. Whenever shelter walls were tightly built to save heat, ventilation became necessary. Attempts made to control the rise of moisture content in air by ventilation were but partially successful. Proper ventilation affected the temperature inside the building and lowered it below the required range. Restricted ventilation was followed but this was considered a partial solution since in low temperatures the amount of air permitted was not enough to remove all the moisture. An acceptable solution was reached by insulating the building which saved a high percentage of sensible heat for warming
the coming air instead of being lost through the walls. The increase in volume of air allowed the removal of excess moisture in vapor form without permitting rapid temperature changes.

A statement made by Whitefield (45) showed that when a poultry house having 300 birds was well insulated and ventilated, the increased income amounted to $210 which covered the expense of insulation. Experience has also shown that it pays to keep cows comfortable (14). But a limit to the use of insulation has to be regarded according to the expense and benefit that will be returned. Production of animals or poultry and the consumption of food depend to a great extent upon the environmental conditions. Animal comfort is greatly affected by air temperature, air movement, humidity and temperature of the nearby surfaces, such as walls, floor and ceiling. If the physical condition of air becomes such that heat loss from the body is greatly retarded, the animal will feel uncomfortable. If the atmosphere is high in temperature, has much moisture and is stagnant, it will be a poor one for taking heat up by radiation and evaporating moisture from the body. On the contrary, if the temperature is too low, heat radiated from the animal will increase and the animal will also feel uncomfortable.

All of these factors can be regulated by ventilation, insulation of the building and artificial heating in addition to the heat produced by the animal.
For dairy cows sudden fluctuation in temperature may seriously affect milk production even if it is in the range that the cow can stand which is usually between $40^\circ$ and $65^\circ$ F. (34). At the same time a heavy producing dairy cow needs from 70 to 150 pounds of water daily and will not drink enough if the water is near the freezing point (7). The temperature of the surroundings also plays an essential part in the amount of water that the cow drinks. When she has to stand in very low temperature she is likely to drink relatively little regardless of the temperature of the water.

It has been proven by experiments with hens that they can adjust themselves to a temperature of about freezing if they are not exposed to drafts but they also are affected by sudden changes in temperature (34). In moist air combs freeze at $6^\circ$ F., and at $8^\circ$ to $10^\circ$ F., egg production is reduced 25% while at $0^\circ$ F. hens stop laying (30).

It has also been noticed that if the temperature of air inside the shelter is low, this will influence the care given to the animals by the worker. It will not be easy for the human body to stand a long time under such atmosphere. Freezing of water in bowls and pipes will also be problems that the farmer will face if the temperature drops below freezing.

Artificial heating in barns or laying houses is not accepted by farmers and they try to conserve the heat produced by the animal. The engineer, therefore, confronted
with the design of livestock buildings, must know about these animals or birds, not as food producers, but as heat and moisture producers. The amount of heat and moisture produced by the animal depends upon some factors which are: (1) the size of the animal or bird, (2) the surrounding temperature, (3) the amount of activity and the quantity of food being assimilated. To keep the temperature of the building at a certain level, the production of heat and loss of heat must be in equilibrium at this certain level. Heat is lost through (1) walls, ceiling, floor, windows and doors, (2) the infiltration of cold air and exfiltration of warm air through doors, windows and cracks, (3) the loss through ventilation.

The first and second losses can be controlled by the use of insulation of walls and roofs, double windows and doors, well and tightly built structure.

In ventilation, volume of air required can be calculated if we know the amount of moisture to be removed from the building, temperature and humidity of the incoming air and temperature and humidity required for the inside air. The following table gives an idea about some animals as producers of heat and moisture.
Table 1
Production of Heat and Moisture by Farm Animals

<table>
<thead>
<tr>
<th>Animal</th>
<th>Total heat produced Btu/hr.</th>
<th>Moisture produced grains/hr.</th>
</tr>
</thead>
<tbody>
<tr>
<td>900 lb. Jersey producing 20 lb. milk per day</td>
<td>2700.0</td>
<td>4460 (2)</td>
</tr>
<tr>
<td>1250 lb. Holstein producing 30 lb. milk per day</td>
<td>3295.0</td>
<td>5450 (2)</td>
</tr>
<tr>
<td>Hen (4 lb. leghorn) at 35° F.</td>
<td>38.85</td>
<td>20 (30)</td>
</tr>
<tr>
<td>Farm Horse, 1000 lb.</td>
<td>1794</td>
<td>2972 (2)</td>
</tr>
<tr>
<td>Sheep, 100 lb.</td>
<td>245</td>
<td>406 (2)</td>
</tr>
</tbody>
</table>

Water vapor that is given off in respiration, evaporation from the skin and evaporation of part of the moisture from the excretion of the animal and water spillage from the waterers are all causes contributory to the increase of moisture in air.

Relative humidity up to 80% is considered quite satisfactory in farm buildings (3) and is usually fixed by the temperature of the inside walls so that no condensation will occur. This temperature must be higher than the dew point of the warm air inside the buildings.
Ventilation of livestock buildings is found to be of great influence on the comfort of the animal. Excess moisture and carbon dioxide are driven out together with the odor and toxic organic compounds. The movement of air within the structure has an effect upon the temperature and relative humidity at which the animal feels comfortable. It has been found desirable to maintain a purity of 9 parts of CO₂ to 10,000 of air (30). A higher percentage of CO₂ than that mentioned above may be maintained in the hen house without apparent harm, but too high a concentration of CO₂ may mean a decreased hatch. It will also be remembered that King's maximum permissible CO₂ concentration was about 17 parts per 10,000 (40). To supply a large enough volume of air to maintain these levels has been found to be incompatible with the control of temperature inside the building. But it is found that in certain ranges of outside temperature, in an average construction, the relative humidity of the air will approach the saturation point long before the carbon dioxide content is high enough to be troublesome. We can therefore conclude that the relative humidity, temperature and air movement are the most important factors in the control of environment for comfort in an animal shelter.
The first systems of ventilation were of the gravity or natural draft type where movement of air depended upon the difference of temperature between the incoming and the outgoing air, height and cross section of the flue and outside air currents. There are three main known ways; namely King's and Rutherford's and a modification of the two. The second system for ventilation is the mechanical or forced system where circulation of air is accomplished by means of fans. The latter can be controlled automatically as desired according to the temperature or the relative humidity within the building.

Design of Ventilation System

In designing for ventilation, the volume of air computed for this purpose if governed by the following factors:

1. The sensible heat and moisture that the animal will produce under a certain environmental condition.

2. Temperature difference between the outside and inside which determines the rate of heat loss.

3. The building itself, whose size, shape and insulation characteristics limit the heat loss.

These factors can be expressed by one formula which depends upon the heat balance per animal basis:

\[ H = \frac{VD}{53} + ACD \]  (40)
in which \( H \) = sensible heat in Btu./Hr. per animal.

\[ V = \text{air needed for ventilation in cubic feet per hour per animal.} \]

\[ 53 = \text{volume of air in cubic feet raised one degree } T^* \text{ by 1 Btu.} \]

\[ = \frac{1}{\rho c} \quad \text{where } \rho \text{ is the density of air in lb./cu. ft.} \]
\[ \text{and } c \text{ is the specific heat in Btu. per pound per degree } F. \]

\[ A = \text{area of exposure to heat loss in sq. ft. per animal.} \]

\[ C = \text{average heat loss through walls, ceiling and windows. } \text{Btu.)/(hr.)(ft.}^2\text{)(deg. }F.\] \]

\[ D = \text{difference in temperature between inside and outside.} \]

In other words for equilibrium we must have:

Heat produced by the animal per hour equals heat lost by ventilation per hour plus heat lost through the building per hour. From this equation it can be seen that \( A \cdot C \) is the governing factor in the design of a structure and the ventilation if the conditions under which the building is to function are established.

In the different livestock buildings, the value \( A \) varies per animal and therefore the heat produced per square foot of exposed enclosing surface varies accordingly. This variation comes either from the difference in the number of the animals housed or the size of the animals. For example the amount of heat produced by a hen per square foot of exposed area is from 5 to 6 Btu. per hour while a cow produces
from 20 to 25 Btu per hour per square foot of enclosure surfaces (38). This means that poultry houses need more resistance to heat flow than dairy barns.

If there is little or no insulation in the construction the value of C will be large and the loss of heat will be great. If we increase, therefore, the resistance to heat flow, we will have a bigger range in the difference of temperatures between the inside and outside air, while for the same difference of temperature, the volume of air for ventilation that is available will be greater in an insulated house than in an uninsulated. In an insulated house the fluctuation of temperature inside the building will not be so rapid so that it will affect the comfort of the animal. This is due to the slow motion and storage of heat in the insulation.

Two methods of gaining heat from natural sources will be discussed in this paper.

In ventilation the heat in the exhaust air is simply thrown away. This air contains enough heat to warm the incoming cold air to a certain extent.

If the exchange of heat is made possible by the introduction of a heat exchanger then we can keep the house warmer and we can circulate more air which keeps the humidity lower; or, from the economic view, we can use a poorly insulated wall. If we can keep the humidity low, this will help in
preventing condensation which is harmful to either the animal or the material of the structure. A moist wall will decay rapidly and at the same time conducts heat faster than a dry wall.

In winter the soil under the livestock house is usually warmer than that of the floor. It is supposed that at a certain depth of about six feet the temperature does not vary rapidly and at the same time it is higher than that of the surface. Heat will flow therefore into the house from the soil. If the temperature of the house is higher than that of the floor there will be loss of heat through the floor to the soil.

One of the problems in a poultry house is to keep the litter dry. If we can keep the temperature of the floor higher than the air just above it, moisture in the air will not condense on the litter.

The formula $H = \frac{VD}{53} + ACD$ is valid if we know the right value of $H$ which the animal will give, if the value of $C$ is fairly accurate and if the difference in temperature is stationary for some time.

If we now include in our calculation the heat gained by the heat exchanger and heat gained or lost through the floor, the value of $H$ will be different for each different temperature of the house.
HEAT GAIN FROM SOIL

In all the discussions made for heat balance in livestock buildings, the item of heat gained or lost through the floor is always neglected. It is clear from the previous discussions that if the temperature of the floor is less than that of air, heat will be lost; while if temperature of the floor is higher than that of air, heat will be gained.

It has been noticed that the surface 6 in. layer of soil which is not under any construction is warmer than the air at every season of the year, while the subsoil is warmer in autumn and winter but cooler in spring and summer (42).

Since in winter the subsoil is warmer, we can make use of this character in gaining heat. Heat will flow naturally from the warm subsoil to the cold surface. If we direct this heat toward the floor and prevent it from being dissipated to the nearby cold soil, we can gain a great percentage. Insulating the foundation may be the solution.

Heat Flow in Soil

The temperature of the soil depends upon the amount of radiant energy that is received from the sun. This amount also depends upon that emitted by the sun and that absorbed by the atmosphere. The important factors that affect the
amount of sun's radiation reaching the earth are:

1. Position related to the angle at which sun rays hit.
3. Elevation and altitude.
4. Distribution of land and water currents.

Flow of heat in soil is greatly affected by the nature of the soil. Color is a factor that changes the absorption intensity, while moisture content affects the specific heat. Specific heat of mineral soils, in spite of variations in texture and organic matters, is about 0.2, but if moisture is added advanced to 20% the specific heat of wet mass becomes 0.33 while an increase to 30% of moisture raises the wet weight specific heat to 0.38 (27).

Soil Conductivity

The main factors that conductivity depends upon for a certain kind of soil are compactness and moisture content.

For dry soil, coefficient of thermal conductivity can be calculated from the formula:

\[ k = k_2P + k_1 (1-P) \]

\( k \) = thermal conductivity of dry soil.
\( k_1 \) = conductivity of soil material.
\( k_2 \) = conductivity of dry air.
\( P \) = porosity.
If we know the conductivity of a soil which is usually determined by experiment where time of flow and rate of flow can be measured, the fundamental equation for the flow of heat in soil giving these relations can be used:

\[
\frac{k}{\rho c} \left( \frac{d^2 \Theta}{dx^2} \right) = \frac{d\Theta}{dt} \tag{32}
\]

\[
\frac{d\Theta}{dt} = \text{rate at which temperature rises with time at a known distance "x" from heat source.}
\]

\[
\frac{d^2 \Theta}{dx^2} = \text{rate at which the fall of temperature decreases on passing from heated end, i.e., rate of change of temperature gradient.}
\]

\[
\Theta = \text{temperature.}
\]

\[
x = \text{distance from origin.}
\]

\[
k = \text{thermal conductivity of soil.}
\]

\[
\rho c = \text{thermal capacity = thermal capacity of soil solid portion + thermal capacity of water = density x sp. heat.}
\]

\[
\frac{k}{\rho c} = \text{coefficient of diffusivity} = a^2
\]

\[
a^2 = \text{is in cm}^2 \text{ per sec. in metric system, and ft}^2 \text{ per hour in the British Units.}
\]

From Smithsonian tables (36):

\[
a^2 = 0.005 \text{ cm}^2/\text{sec. for soils, clay or sand slightly damped.}
\]

\[
= 0.0195 \text{ ft}^2/\text{hour} (1 \text{ cm}^2/\text{sec.} = 3.87 \text{ ft}^2/\text{hour}).
\]

The amount of heat flow can be calculated at any time.

\[
\frac{d\Theta}{dt} = kA \frac{d\Theta}{dx}
\]

\[
\frac{d\Theta}{dx} = \text{heat flow gradient through the layer.}
\]
From the work of Patten (32); Smith (35); Smithsonian tables (36) and Houghten (16); a reasonable figure for k can be chosen as .72 Btu./(hr.)/(sq. ft.)/(deg. F. diff./ft.)

Temperature of Soil at Different Depths in Winter

The temperatures of air in central Iowa as a mean of years from 1873 to 1946 is shown in Table 2.

Table 2

Temperature of Air in Central Iowa
1873 - 1946 (42)

<table>
<thead>
<tr>
<th>Month</th>
<th>January</th>
<th>February</th>
<th>March</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air (average)</td>
<td>18.8</td>
<td>22.6</td>
<td>34.8</td>
</tr>
<tr>
<td>Lowest reached</td>
<td>-47 (in 1912)</td>
<td>-41 (in 1903)</td>
<td>-24 (in 1890)</td>
</tr>
</tbody>
</table>

It was found in Ames that the temperature of soil in 1946 at six foot depth varies between 43°F in January and 40.5°F in March while at one inch deep it fluctuates from about 27°F in January to about 46.5°F in March.

The soil under the structure will not be exposed to direct sun since it is covered by the house frame and roof, so the fluctuation of the surface temperature of the soil is limited.
Also the soil, here, can be considered as slightly damp or nearly dry, since no moisture is available except that which will come from the neighboring soil.

Calculation of Temperature and Heat Flow in Soil

For the penetration of the daily and yearly temperature variation into the earth, we assume the surface of the earth to undergo periodic temperature variations as a result of the fluctuating radiations to which it is exposed. The isothermal surfaces are planes parallel to the earth's surface and so the flow is normal to this surface.

Jobs (19) gives the following relation expressing the temperature $Q$ at any depth reckoned from the average temperature.

$$Q = A e^{\sqrt{\frac{w}{2a^2}} \cdot x} \cos(wt - \sqrt{\frac{w}{2a^2}} \cdot x) + B e^{\sqrt{\frac{w}{2a^2}} \cdot x} \sin(wt - \sqrt{\frac{w}{2a^2}} \cdot x)$$

Where $w = \frac{2\pi}{T}$

$T$ = period of cycle.

$t$ = time at which $Q$ is needed.

If we begin counting time from a moment when $Q$ at surface $(x = 0)$ is max, we have the following:

$$Q = Q_{\text{max}} e^{\sqrt{\frac{w}{2a^2}} \cdot x} \cos(wt - \sqrt{\frac{w}{2a^2}} \cdot x)$$

The temperature at any depth of the earth can also be represented by a Fourier Series: (46)
\[ Q = A_0 + A_1 e^{\sqrt{\frac{w}{2a^2}} x} \cos( wt - \sqrt{\frac{w}{2a^2}} x ) + A_2 e^{\sqrt{\frac{w}{2a^2}} x} \sin( wt - \sqrt{\frac{w}{2a^2}} x ) \]

\[(\cos 2wt - \frac{2w}{2a^2} x) + A_3 \ldots B e^{\sqrt{\frac{w}{2a^2}} x} \sin (wt - \sqrt{\frac{w}{2a^2}} x) + B_3 \ldots \]

\[ Q = A_0 + \sum_{n=1}^{\infty} e^{\sqrt{\frac{nw}{2a^2}} x} \left[ A_n \cos( nwt - \sqrt{\frac{nw}{2a^2}} x ) + B_n \sin( nwt - \sqrt{\frac{nw}{2a^2}} x ) \right] \]

In order to simplify the computations, the harmonics of the higher order of the series can be neglected for points below the surface, since the coefficients are small and the factor \( e^{\sqrt{\frac{nw}{2a^2}} x} \) further reduces their effect. Therefore, the change in temperature due to these coefficients will be slight.

Therefore:

\[ Q = A_0 + A_1 e^{\sqrt{\frac{w}{2a^2}} x} \cos( wt - \sqrt{\frac{w}{2a^2}} x ) + B_1 e^{\sqrt{\frac{w}{2a^2}} x} \sin( wt - \sqrt{\frac{w}{2a^2}} x ) \]

\( A_0 \) is proven by Wolfe (46) to be equal to the arithmetic mean of the temperature at the surface for the periodic function for the interval \( Q = 0 \) to \( Q = 2\pi \), therefore \( A_0 \) can be taken as the mean temperature.
\[ Q \text{ from the mean} = A_1 \sqrt{\frac{w}{2a^2}} x \cos(\omega t - \sqrt{\frac{w}{2a^2}} x) + B_1 \sqrt{\frac{w}{2a^2}} x \sin(\omega t - \sqrt{\frac{w}{2a^2}} x) \]

From the example solved by Wolfe (46) it can be seen that the second term can be eliminated for a simple sinusoidal surface variation.

\[ Q = A_1 \sqrt{\frac{w}{2a^2}} x \cos(\omega t - \sqrt{\frac{w}{2a^2}} x) \text{ Same as Joos (19)} \]

\[ Q \text{ at any depth from the mean} = \]

\[ Q_{\text{max}} \sqrt{\frac{w}{2a^2}} x \cos(\omega t - \sqrt{\frac{w}{2a^2}} x) \] (This is the second part of the Joos equation)

Putting \[ \frac{Q}{Q_{\text{max}}} = z \]

\[ \therefore z = \sqrt{\frac{w}{2a^2}} x \cos(\omega t - \sqrt{\frac{w}{2a^2}} x) \]

Stored Heat in Summer or Lost Heat in Winter

If we draw the curves of \( \Theta \) against \( x \) for different \( \text{which} = T/8, T/4, 3T/8, T/2 \)

Stored heat equals area under the curve from \( x = 0 \) to \( x = \), multiplied by the thermal capacity for a unit area

\[ H = c \int_0^\infty Q \, dx \]

\[ = c \int_0^\infty z \, dx \]

\[ = c Q_{\text{max}} \int_0^\infty z \, dx \]

\[ \theta \]

\[ \delta_{\text{mean}} \]

\[ x \]

\[ \Delta x \]
This will be max. when heat has just ceased to enter the surface and is beginning to leave, or when temperature gradient

$$\frac{dz}{dx} = 0$$

Since \( dH = k \frac{dQ}{dx} \)

Putting \( \sqrt{\frac{w}{2a}} = C \)

$$z = e^{-Cx} \cos(\omega t - Cx)$$

$$\frac{dz}{dx} = C e^{-Cx} \sin(\omega t - Cx) - C e^{-Cx} \cos(\omega t - Cx)$$

at \( x = 0 \) for the surface

$$\frac{dz}{dx} = C(\sin \omega t \cos \omega t) = 0$$

This can equal zero when

$$\sin \omega t = \cos \omega t$$

or if

$$\omega t = \frac{\pi}{4}$$

or

$$\frac{2\pi t}{T} = \frac{\pi}{4}$$

or

$$t = \frac{T}{8}$$

$$z = e^{-Cx} \cos \left(\frac{\pi}{4} - Cx\right)$$

$$H = \int_{-\infty}^{\infty} zdx = \int_{-\infty}^{\infty} e^{-Cx} \cos \left(\frac{\pi}{4} - Cx\right)dx$$

for the solution of the integral

if \( \int u \, dv = uv - \int v \, du \)

$$\therefore \int z \, dx = \int e^{-Cx} \cos \left(\frac{\pi}{4} - Cx\right)dx$$
\[-22-\]

\[
\begin{align*}
    &= -e^{C_x} \left( -\frac{1}{\alpha} \right) \sin \left( \frac{\pi}{4} - C_x \right) - \int \frac{1}{\alpha} \sin \left( \frac{\pi}{4} - C_x \right) \left( -e^{C_x} \right) \\
    &= -\frac{1}{\alpha} e^{C_x} \sin \left( \frac{\pi}{4} - C_x \right) - \left[ \frac{1}{\alpha} \cos \left( \frac{\pi}{4} - C_x \right) - \frac{1}{\alpha} \right] e^{C_x} \cos \left( \frac{\pi}{4} - C_x \right) - \int z \, dx \\
    \int z \, dx &= -\frac{1}{\alpha} e^{C_x} \sin \left( \frac{\pi}{4} - C_x \right) - \frac{1}{\alpha} e^{C_x} \cos \left( \frac{\pi}{4} - C_x \right) - \int z \, dx \\
    \therefore 2 \int z \, dx &= -\frac{1}{\alpha} e^{C_x} \left[ \sin \left( \frac{\pi}{4} - C_x \right) + \cos \left( \frac{\pi}{4} - C_x \right) \right] \\
    H &= -\rho c_0 \max \frac{1}{2\alpha} \left[ e^{C_x} \left\{ \sin \left( \frac{\pi}{4} - C_x \right) + \cos \left( \frac{\pi}{4} - C_x \right) \right\} \right]^2 \\
    \text{at } x = \infty \text{ the quantity } \left[ \ldots \right] = 0 \\
    H &= \rho c_0 \max \frac{1}{2\alpha} \left[ \sin \frac{\pi}{4} + \cos \frac{\pi}{4} \right] \\
    H &= \rho c_0 \max \frac{1}{2\alpha} \left( \frac{1}{\sqrt{2}} \frac{1}{\sqrt{2}} \right) = \rho c_0 \max \frac{1}{\sqrt{2}} \\
    H &= \rho c_0 \max \frac{1}{\sqrt{\frac{\pi}{2}}} = \rho c_0 \max \frac{1}{\sqrt{\frac{\pi}{2}}} \\
    H &= \rho c_0 \max \frac{1}{\sqrt{\frac{\pi}{2}}} \frac{1}{a^2} = \rho c_0 \max \frac{1}{\sqrt{\frac{\pi}{2}}} \frac{1}{a^2} \\
    H &= \rho c_0 \max \frac{1}{\sqrt{\frac{\pi}{2}}} \frac{1}{a^2} \text{ if } a^2 = \frac{k}{\rho c} \\
    H &= Q_{\max} \sqrt{\frac{k/\rho c T}{2\pi}} \\
\end{align*}
\]

For a certain area total heat can be calculated by multiplying \( H \) by the area \( A \).

(These equations have been solved by the help of Mr. Thorburn himself (41) and have not been published.)
Considering that \( k = .72 \) and density of soil to be 115 lb./cu. ft. and specific heat to be \( .31^* \) we will have the following results:

\[
a^2 = \frac{k}{\gamma C} = \frac{.72}{115 \times 0.31} = .02 \text{ which checks with the figure given by Smithsonian (36) as (.0195 for soils, clay or sand slightly damp).}
\]

The amount of heat stored can also be estimated if we have our complete cycle from which we can know the maximum temperature from the mean.

From Wolfe (46) the solved example gives 71.2° F. as maximum temperature and 47.7 as the mean temperature.

\[
Q_{\text{max}} = 71.2 - 47.7 = 23.5^0 \text{ F.}
\]
\[
H = Q_{\text{max}} \sqrt{\frac{k/C T}{2\pi}}
\]
\[
H = 23.5 \sqrt{\frac{.72 \times 115 \times .31 \times 365 \times 24}{2}}
\]
\[
= 23.5 \sqrt{36,000}
\]
\[
= 4450 \text{ Btu/sq. ft. per year.}
\]

This heat will be dissipated outside in half a year.

\[
\therefore h = \frac{4450 \times 2}{365 \times 24} = 1.01 \text{ Btu./(hr.)(ft.}^2\)
\]

where \( h = \text{average heat dissipated per square foot per hour from the soil surface.} \)

The following table gives the average temperature of soil at 72 inches deep in January.

\*

Assuming that the specific heat of dry soil is \( .2 \) and per cent of moisture by dry weight is varying from 10% to 20%. 

### Table 3

Average Temperature of Soil at 72 Inches in January in Iowa

<table>
<thead>
<tr>
<th>Year</th>
<th>Temp. F.</th>
</tr>
</thead>
<tbody>
<tr>
<td>1947</td>
<td>44.5</td>
</tr>
<tr>
<td>1946</td>
<td>43</td>
</tr>
<tr>
<td>1945</td>
<td>44.6</td>
</tr>
<tr>
<td>1944</td>
<td>-</td>
</tr>
<tr>
<td>1943</td>
<td>45</td>
</tr>
<tr>
<td>1942</td>
<td>46.1</td>
</tr>
<tr>
<td>1941</td>
<td>44.5</td>
</tr>
<tr>
<td>Ave.</td>
<td>44.6</td>
</tr>
</tbody>
</table>

Assuming in a poultry house that the temperature fluctuation is not wide and that we always try to keep the temperature about 35° F., we can calculate the heat coming up from the soil if we know the temperature gradient at the surface.

This gradient seems to be approximately a straight line in the first six foot layer (from experiment).

\[
\frac{dH}{dt} = \frac{k dQ}{dx} = \frac{.72(44.6-35)}{6} = 1.15 \text{ Btu.}/(\text{hr.})(\text{ft.}^2)
\]

This result should be greater if we compare it to the one obtained in the above solution by calculating the heat stored. In the first calculation the figure given is the average of the heat dissipated per hour in half a year, and it is higher in the cold days and vanishes until it reaches zero. If the soil is not covered by the structure the temperature will be lower than 35 degrees in the cold days, and accordingly the heat that comes out will be greater.
Experiment for Heat Gained from Soil

Temperatures of the soil at different depths were measured at two points. Point 1 was in a soil which was insulated from the neighboring soil by a trench of one foot width and four feet depth full of chopped corn stalk, while point 2 was in an ordinary soil.

A small structure used as a shelter was made to cover the two points. This shelter was built north of the Agricultural Engineering building and was put in a position six feet from the building and eleven feet from the western concrete fence. The walls were built one foot and six inches high so that loss of heat through walls and roofs would be small.

Description of the structure

The structure was six feet wide by twelve feet long and was divided into two compartments. The first one to the west stood over an insulated patch, while in the other the walls rested on the surface of the soil. Studs and rafters were made of 2 inch x 4 inch spaced at three feet apart. The siding and the inside lining were made of 1 inch x 8 inch boards. The space between the outside and the inside boards was filled with chopped corn stalk used as an insulating material.
The roof was made of the same section as the wall and was covered with plain bituminous roof cover. Two doors in the roof were made for entrance into the compartments. They were made of the same section as the roof.

The structure was carried on four poles where the soil was insulated under compartment No. 1. The chopped corn stalk was kept in a waterproof material (sisal paper) so that the insulating character would not be lost by moisture in the soil or the decay of the material. A partition wall separated the two compartments; its cross section was the same as that of the side walls and roof.

The two points No. 1 and No. 2 were chosen at two feet from the south wall and two feet from the partition wall. Temperatures of the soil were measured at six feet, four feet, two feet deep and at the surface. Air temperature inside the compartments was measured at two feet high from the surface of the soil. Constantan-Copper thermocouples were used in measuring these temperatures at the different points and depths. A hole of three inches in diameter was drilled in the soil by an auger to a depth of six feet and then a thermocouple wire was put in. The wire was made to have a horizontal helical part of two inches long so that there would be enough length of wire in the plane of equal temperature (isothermal plane).
Fig. 1. Shed constructed over the two plots.
## Results of experiment

### Table 4

Temperature of Soil at Different Depths Assuming January 1st to January 5th as period No. 1

<table>
<thead>
<tr>
<th>Period</th>
<th>Insulated soil (1)</th>
<th>Uninsulated soil (2)</th>
<th>Agronomy bare soil</th>
</tr>
</thead>
<tbody>
<tr>
<td>No.</td>
<td>Date</td>
<td>0&quot;</td>
<td>24&quot;</td>
</tr>
<tr>
<td>1:07/31</td>
<td>0</td>
<td>27</td>
<td>34.6</td>
</tr>
<tr>
<td>2:08/2</td>
<td>5</td>
<td>27</td>
<td>32</td>
</tr>
<tr>
<td>3:09/2</td>
<td>10</td>
<td>27</td>
<td>32</td>
</tr>
<tr>
<td>4:10/2</td>
<td>15</td>
<td>27</td>
<td>32</td>
</tr>
<tr>
<td>5:11/2</td>
<td>20</td>
<td>27</td>
<td>32</td>
</tr>
<tr>
<td>6:12/2</td>
<td>25</td>
<td>27</td>
<td>32</td>
</tr>
<tr>
<td>7:13/3</td>
<td>20</td>
<td>27</td>
<td>32</td>
</tr>
<tr>
<td>8:14/3</td>
<td>25</td>
<td>27</td>
<td>32</td>
</tr>
<tr>
<td>9:15/3</td>
<td>30</td>
<td>27</td>
<td>32</td>
</tr>
</tbody>
</table>
Thermocouples, in the same way, were put at four feet, two feet and on the surface of the soil. The terms 1-6, 1-4, 1-2, 1-0 and 2-6, 2-4, 2-2, 2-0 and 3-1 and 3-2 designate the different depths of the thermocouple wires at point 1 and 2. Point 3 indicates air temperature in compartment 1 and 2. A potentiometer which was previously calibrated was used to read the temperatures if one end of the thermocouple was immersed in a mixture of ice and water at 32° F.

Effect of heat gained from soil

The amount of thermal units that are stored in the soil during the summer time and are dissipated during the winter time account to approximately 4500 Btu. per square foot of surface area.

This amount seems to be of use if we can extract these units by a mechanical means during cold waves.

Pushing air through a tunnel dug underneath the house at a depth of 15 feet was the suggestion made by Thorburn (41) as a solution for the problem of heating or cooling for a dwelling house.

Operating a heat pump is another suggestion verified by some companies as Muncie Gear Works Inc., Muncie, Indiana. A heat pump is a reverse operation to the refrigerating system.
FIG 2A  SEC. ELEVATION

Plan showing location of structure

SHED 12' x 6'
CONSTRUCTED OVER TWO PLOTS FOR MEASURING SOIL TEMPERATURE
WEST PLOT INSULATED SOIL
EAST PLOT UNINSULATED SOIL

AGR. ENG. MACHINE LABORATORY
In this system heat is taken from the soil by water pipes and then passes through a radiator which will act as an evaporator to the refrigerant after passing an expansion valve. The refrigerant vapor will be warmed up before it goes to the compressor which will rise its temperature more by rising its pressure. This high temperature vapor will pass through another heat exchanger used as a condenser where house air will be the cooling medium. This air will be warmed up.

The performance of such a system is equal to \( \frac{T_1}{T_1 - T_2} \)

where \( T_1 \) = the house or condenser temperature.

\( T_2 \) = the water or evaporator temperature.

We can see from this equation that this performance can be increased if we increase \( T_2 \). In winter, water from the soil is a good source of heat for the evaporator.

All these mechanical means are expensive for an animal shelter.

During January when most of the severe cold waves occur, the average temperature at 72 inches is 44.6 degrees (average of 1941 - 1947 for central Iowa (42)). The amount of heat that is expected to flow out of the soil is 1.15 Btu./(ft.\(^2\))(hr.) assuming that the inside temperature of the house is in the range of 35 degrees. That amounts to 4.6 Btu./hr./bird. This amount, although it is comparatively small, can be beneficially used if it is correctly treated and
studied. It can raise the temperature of the house 2.3 degrees if it is fully utilized. \( \text{ACD} + \frac{60}{53} D = 4.8 \)

The main object of heat that comes from the soil is to keep the floor at a higher temperature than that of the air just above it so that no condensation will occur.

This can be verified if we make the floor of a good conductive material such as concrete.

Many objections may be aroused to the use of this conductive material since it will make the animal feel uncomfortable. The use of a straw bed for cows in the barns or litter in poultry houses will be the solution.

This floor must be separated from the foundation by an insulating material to prevent the flow of heat from the inside warm air, through the conductive floor to the cold foundation.

It is also noticed that condensation usually occurs on the floor at the part near the foundation. This is due to the existence of cold soil beneath the floor beside the foundation. If we insulate the inside face of the foundation by an insulating material as zonolite concrete, this will prevent the outside cold soil from affecting the inside one. If we make the zonolite concrete of the ratio 1 to 9 we will have a material of thermal conductivity of .96 Btu./(hr.)(ft.\(^2\))(deg. F./in.) and which will cost approximately
$0.50 per cubic foot. Chopped corn stalk protected by waterproof material as sisal paper or bituminous sheets will do the same job.
HEAT GAIN FROM HEAT EXCHANGER

Heat Transfer in the Heat Exchanger

The use of a heat exchanger in this problem is to warm the coming cold air for ventilation by the warm exhaust and room air. The flow of heat occurs in this heat exchanger by the three methods of heat transmission; namely, (1) conduction, (2) convection, and (3) radiation. Transfer of heat between the air flowing through inside pipes and air in the annular space occurs by the first two methods while transmission between the air in the annular space and room air occurs by all three divisions. Each method has many factors affecting the transfer of heat from one side to the other.

Conduction is known as the flow of heat from a high temperature region to a lower temperature region within the body, through simple molecular communication or from one body at higher temperature to another at lower temperature if the two bodies are in physical contact. The quantity of heat transferred depends upon the resistance of the material to heat flow, the thickness of the material and the difference of temperatures between the two surfaces.

In our case, since the pipes are made of galvanized
steel, whose thermal conductivity is high and there is practically no resistance to heat flow, we can consider that the only factor affecting the overall coefficient of transmission is the surface conductance due to convection or radiation or both.

Formulae Governing Forced Convection

Heat is carried from the higher to the lower temperature within a gas or a liquid by moving masses of the fluid. Convection will be either natural or forced. The process is called natural when circulation is caused merely by the difference in densities caused by the change in temperature. This occurs in our case between the outside pipe and room air. If the circulation is caused by a fan, then heat is transferred by forced convection as that which occurs in the inside pipe and in the annular space.

Formulae governing convection are complicated and are solved by dimensional analysis and experiments. Since air is flowing in a pipe, therefore the character of the flow is dependent upon the velocity, the density and the viscosity. Turbulence flow occurs when flow is beyond a certain critical velocity.

To get this critical velocity for flow of air in pipes, the following formula is used:
V_{critical} = \frac{2300 \mu}{\rho D}

where \( V = \) velocity ft./hr.
\( \mu = \) viscosity, lb./(ft.)(hr.)
\( \rho = \) density of fluid, lb./cu. ft.
\( D = \) diameter of pipe in ft.

In our case for six inch pipe and air at 70 degrees

\[
V_{critical} = \frac{2300 \times 0.044}{0.075 \times 0.5} = 2700 \text{ ft./hr.} = 0.75 \text{ ft./sec.}
\]

If the discharge of air is more than 100 cu. ft./min., the velocity in six inch pipe will be more than 8.5 ft./sec. and the flow can be considered turbulent. For the computation of the surface coefficient of forced convection in turbulent flow, the general formula takes the following shape:

\[
\frac{h_c D}{k} = C \left( \frac{D \nu}{\mu} \right)^b \left( \frac{c \nu}{k} \right)^d \quad \text{(6)}
\]

\[
\frac{h_c D}{k} = \text{Nusselt number}
\]
\[
\frac{D \nu}{\mu} = \text{Reynolds number}
\]
\[
\frac{c \nu}{k} = \text{Prandtl Number}
\]
\[
c = \text{specific heat at constant pressure}
\]
\[
k = \text{thermal conductivity}
\]

McAdams (29) has concluded that the following final shape of this formula is good for the heating of not only water,
oils and several liquids, but also gases in turbulent flow in horizontal pipes.

\[ h_c = 0.0225 \frac{k}{D} \left( \frac{D V c}{\mu} \right)^{0.8} \left( \frac{c_m}{k} \right)^{0.4} \text{ Btu/}(\text{hr.})(\text{sq. ft.})(\text{deg. F.}) \]

for the flow of air in pipes

\[ h_c = C \frac{(V_c)^{0.8}}{D^{0.2}} \]

where \( C = 0.0225 \frac{k}{\mu} \left( \frac{1}{\mu} \right)^{0.8} \left( \frac{c_m}{k} \right)^{0.4} = 0.0225 \frac{k}{\mu} \cdot 0.8 \cdot 0.4 \]

The constant \( C \) depends upon the temperature of the air.

Putting \( V_c = 1 \text{ ft./sec.} \) and \( D_c = 1 \text{ inch} \)

\[ h_o = 0.0863 \cdot 1 \times 3600 \cdot 0.8 \cdot \frac{12}{1} = 1150 \cdot C^{0.8} \]

Table 5
Value of Constant "C" \( \xi h_o \)

<table>
<thead>
<tr>
<th>Temperature, °F.</th>
<th>C</th>
<th>( \rho )</th>
<th>( \rho \cdot 8 )</th>
<th>( h_o )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0.00328</td>
<td>0.0863</td>
<td>0.14</td>
<td>0.53</td>
</tr>
<tr>
<td>20</td>
<td>0.00331</td>
<td>0.0827</td>
<td>0.136</td>
<td>0.52</td>
</tr>
<tr>
<td>40</td>
<td>0.00334</td>
<td>0.0794</td>
<td>0.134</td>
<td>0.516</td>
</tr>
<tr>
<td>60</td>
<td>0.00337</td>
<td>0.0763</td>
<td>0.128</td>
<td>0.497</td>
</tr>
</tbody>
</table>

In McAdams book (29) there is a simplified equation when air is at 80 degrees which is very near to this one.
Putting where

\[ h_c = 0.0144 \frac{c \cdot g^{0.8}}{D^2} \]

Therefore

\[ c = 0.24 \]

\[ h_c = 0.00345 \frac{g^{0.8}}{D^2} \]

where

\[ G = fV \]

\[ d = \text{diam. in inches} \]

\[ V = \text{vel. in ft./sec.} \]

\[ h_c = h_o \left( \frac{1}{d} \right)^2 \cdot 8 \cdot 2 \cdot V = h_o \cdot F_d F_v \]

where

Table 6
Value of Fd

<table>
<thead>
<tr>
<th>d in inches</th>
<th>Fd</th>
<th>d in inches</th>
<th>Fd</th>
<th>d in inches</th>
<th>Fd</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.</td>
<td>9</td>
<td>.644</td>
<td>17</td>
<td>.567</td>
</tr>
<tr>
<td>2</td>
<td>.87</td>
<td>10</td>
<td>.633</td>
<td>18</td>
<td>.56</td>
</tr>
<tr>
<td>3</td>
<td>.80</td>
<td>11</td>
<td>.62</td>
<td>19</td>
<td>.546</td>
</tr>
<tr>
<td>4</td>
<td>.76</td>
<td>12</td>
<td>.608</td>
<td>20</td>
<td>.55</td>
</tr>
<tr>
<td>5</td>
<td>.725</td>
<td>13</td>
<td>.59</td>
<td>21</td>
<td>.553</td>
</tr>
<tr>
<td>6</td>
<td>.70</td>
<td>14</td>
<td>.59</td>
<td>22</td>
<td>.54</td>
</tr>
<tr>
<td>7</td>
<td>.678</td>
<td>15</td>
<td>.582</td>
<td>23</td>
<td>.535</td>
</tr>
<tr>
<td>8</td>
<td>.66</td>
<td>16</td>
<td>.575</td>
<td>24</td>
<td>.53</td>
</tr>
</tbody>
</table>
Table 7
Value of $F_v$.

<table>
<thead>
<tr>
<th>V ft./sec.</th>
<th>$F_v$</th>
<th>V ft./sec.</th>
<th>$F_v$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.</td>
<td>12</td>
<td>7.3</td>
</tr>
<tr>
<td>2</td>
<td>1.74</td>
<td>14</td>
<td>8.22</td>
</tr>
<tr>
<td>4</td>
<td>3.04</td>
<td>16</td>
<td>9.2</td>
</tr>
<tr>
<td>6</td>
<td>4.2</td>
<td>18</td>
<td>10.1</td>
</tr>
<tr>
<td>8</td>
<td>5.27</td>
<td>20</td>
<td>11.0</td>
</tr>
<tr>
<td>10</td>
<td>6.3</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

In the case of the heat exchanger, to get the outside coefficient of the inner pipe, the term $D$ should be replaced by an equivalent diameter $D_e$ which equals to $\frac{4A}{P}$ (6). Where $A$ is the cross sectional area of the space between the tubes and $P$ is the number of tubes times the perimeter of each tube. In case of double pipe heater exchanger

$$D_e = \frac{D_2^2 - D_1^2}{D_1}$$

Where $D_1$ is the diameter of the inner pipe and $D_2$ is the diameter of the outer pipe. This procedure is recommended by Nusselt.

The insertion of supports in the annular space to keep the inside pipe in position will increase the turbulence which accordingly will increase the surface coefficient.
To compute the outside surface coefficient of the outside pipe, natural convection and radiation should be considered.

**Formula Governing Natural Convection**

$$h_c = Ck(\frac{a\Delta T}{L})^\frac{1}{2} \text{ Btu.}/(\text{hr.})(\text{sq. ft.})(\text{deg. F.})$$

- $C = .45$ for horizontal pipes.
- $a = 1.65 \times 10^6$ for air at $70^\circ$.
- $\Delta T = \text{difference in temperature of the surface of the pipe and air.}$

$$h_c = .23(\frac{9}{775})^\frac{1}{2} = .43 \text{ Btu.}/(\text{hr.})(\text{ft.}^2)(\text{deg. F.}).$$

**Effect of Radiation from the Building and the Surroundings**

The following formula is used in calculating coefficient of radiant heat transfer:

$$h_r = \frac{\varepsilon F_e F_a (T_s^4 - T_r^4)}{T_s - T_r} \quad (6)$$

- $T_s = \text{absolute temp. of the enclosing surface (room walls).}$
- $T_r = \text{absolute temp. of the outside wall of the pipes.}$
- $t_s$ and $t_r$ are temperatures corresponding to $T_s$ and $T_r$.
- $\varepsilon = .174 \times 10^8$ found in Stefan Boltzmann law for calculation of the total energy radiated by a black body.
- $F_a = 1$ for a completely small enclosed body compared with the enclosing body.
\[ F_e = \text{a factor depending upon emissivities of the two bodies.} \]

\[ \text{emissivity of the enclosed body } = 0.228 \text{ for galvanized sheet iron.} \]

\[ h_r = \frac{(0.174)(0.228)}{77 - 68} \left[ (5.37)^4 - (5.38)^4 \right] = \frac{0.174 \times 0.228 \times 60}{9} \]

\[ = 0.265 \text{ Btu.}/(\text{hr.})(\text{ft.}^2)(\text{deg. F.}) \]

Adding the radiation coefficient to the coefficient of natural convection we get

\[ h_{outside} = 0.43 + 0.265 = 0.695 \text{ Btu.}/(\text{hr.})(\text{ft.}^2)(\text{deg. F.}) \]

While warm air is flowing through the inside pipe, it will get in contact with the pipe wall which is already in contact with cold air. If the temperature of the wall is lower than the dew point of the warm air, condensation will occur. If condensation occurs, two results will accordingly follow:

1. Surface coefficient will increase due to the wet surface effect.
2. Latent heat is released from condensation and so more heat is gained through the heat exchanger.

A similar process may happen on the outside surface of the 9 inch pipe. When the surface temperature reaches that of the dew point of the room air, condensation will occur and we will again gain more heat.

The overall coefficient of heat transmission depends upon the conductivity, surface coefficients (inside and
outside including radiation if any) and is smaller than any one of them. Considering heat transfer between warm air in the inside pipe and cold air in the annular space, the following points should be noted:

Surface coefficient is calculated by the formulae mentioned noticing that this coefficient increases if condensation occurs. If dry air is flowing in 6 inch diameter pipe at a velocity 8.5 ft./sec. the coefficient will be about 2 Btu./(hr.)(ft.)^2(deg. F.). If pure vapor is passing and is condensing, this coefficient raises to 2000 Btu./(hr.) (ft.)^2(deg. F. diff.) (6). The outside surface coefficient of the inside pipe depends here upon the velocity of air in the annular space and the equivalent diameter and temperature difference.

$$\frac{1}{U} = \frac{1}{h_1} + \frac{1}{h_o}$$

$h_o$ is the outside surface coefficient and can be calculated. $h_1$ is the inside surface coefficient and is not definite due to condensation. From this formula we can deduce that the limiting factor for the overall coefficient is the outside surface coefficient.

Some discussion can be made for heat transfer between cold air in annular space and warm air of the room. Inside surface coefficient will be the limiting factor since condensation may occur on the outside surface.
Experiment for the Heat Exchanger

A double pipe heat exchanger was built in the Agricultural Engineering Machine laboratory in the space of the monitor. It consisted of a nine inch outside pipe and six inch as inside pipe. Pipes were of galvanized sheet metal.

Since the relative humidity of air in the laboratory was very low (20% R.H.) a humidifying chamber was made at the inlet of warm air. The chamber had the cross section of 16 in. x 16 in. and length of seven feet and was divided into sections. There were four atomizing nozzles to give a very fine spray in the air to increase the moisture content. Then there was an eliminator to free the air from drops of water. Four resistant heaters each of 660 watts were in the second section to raise the temperature of air to the required degree. Two heaters out of the four were connected to a thermostat.

An adjustable gate at the entrance and a damper in the outlet pipe were made to regulate the amount of warm air that flew in the six inch pipe. Warm air from the room was blown into the inner six inch pipe by a blower driven by a 1/4 horsepower motor. The revolutions per minute of the blower could be adjusted by a transmission pulley of 2 inch and 4 inch to be one quarter and one half that of the motor (1725 r.p.m.).
Cold air from outside was pushed through the nine inch pipe by a small propeller type fan which was changed later by a vacuum cleaner fan. Outlets of warm or cold air are shown in points 5 and 1 (Fig. 6). At point 5 a box of 12 in. x 12 in. x 12 in. was made where the six inch pipe ends with an elbow to be the outlet of the warm air of the room. At point 1, the nine inch pipe was closed and a six inch attached pipe was used to be the outlet of the coming cold air.

Constantan-Copper thermocouples were put in four points to measure the temperature of air in the inside pipe (warm air) and air in the annular space (cold air). Dry bulb temperatures and wet bulb temperatures were measured for warm air at the inlet at point 1 and at the outlet at point 5.

If the temperature of the pipe wall is below the dew point, condensation is expected to occur in this section of the pipe. The condensation of moisture can start in the surface film while the main body of air may be considerably above saturation point.

The quantity of air that flew in each pipe was known by measuring the velocity of air at the outlets where the cross sectional areas were known.

Velocity of air was measured by a thermocouple anemometer made by Mr. W. V. Hukill. In the anemometer the
Fig. 5. Heat exchanger
thermocouple is No. 24B and 8 gage copper and Constantan wire. The Constantan is one inch long and is butt-joined to the copper with silver solder making two junctions one inch apart. One of the junctions is wrapped with about 80 turns of No. 40 nichrome wire, through which an electric current of known amperage may be passed. The current through the heating coil was adjusted until a reading of 5.498 millivolts in the potentiometer was secured.

The cold junction of the thermocouple anemometer is immersed in the air stream to be measured. It will have a temperature equal to that of air. The thermocouple furnishes a measure of the rise in temperature of the heated junction due to the heating current. The c.m.f. is read and by a special calibrated curve the velocity can be known.

Effect of radiation of the walls of the pipes and effect of convection on the wires of the thermocouple used in measuring the temperatures can be considered negligible as it appears from the calculation of this effect in one of the readings under the following assumptions. The heat flow between the air and thermocouple by radiation plus heat flow between the air and thermocouple by convection equals heat flow between thermocouple and walls by radiation. Since the first item is very small, item 2 is equal to item 3.
Cold Air Inlet

Warm Air Inlet

Fig. 7. Air inlets.
\[ h_c (T_g - T_c) = 0.174p \left[ \left( \frac{T_c}{100} \right)^4 - \left( \frac{T_w}{100} \right)^4 \right] = \frac{\phi r (T_c - T_w)}{h_c} \] (29)

\[ T_g = \text{true temperature of air}. \]
\[ T_c = \text{temperature read by thermocouple}. \]
\[ T_w = \text{temperature of the wall}. \]
\[ p = \text{emissivity of wires}. \]

\[ \therefore T_g - T_c = \frac{T_c - T_w \phi r}{h_c} \]

The surface coefficient due to convection of air passing over wires can be obtained by the formula:

\[ h_c \cdot \frac{D}{K} = 0.35 \frac{(DC)}{\eta} \cdot 56 \] (McAdams) (29)

or by the Imperical formula \[ h_c = \frac{0.03 \cdot 56}{D \cdot 44} \]

\[ \phi r = 0.9 \times 1 \text{ (from curves from McAdams)} \]
\[ G = \rho V = 0.073 \times 5.9 \times 3600 = 1555 \text{ #/ (hr.) (ft.}^2 \text{)} \]

(reading No. 20, Table 9c)

Assuming diameter of wires of thermocouple to be 1/16 inch

\[ h_c = 0.03 \times \frac{(1555)^{\cdot 56}}{(\frac{1}{16 \times 12})^{\cdot 44}} = 18.48 \text{ Btu}./(\text{hr.})(\text{deg. F.}) \]

\[ T_g - T_c = \frac{(87.5 - 80.7) \times 0.9}{18.48} = 0.33 \text{ F.} \]

This result can be considered negligible and at the same time will be added for each reading. The difference between
Fig. 8. Thermocouple wire anemometer.
inlet temperatures and outlet temperatures after being corrected will be the same as that measured directly by the thermocouple.

In the calculation of heat transfer, the logarithmic mean temperature difference is used which is equal to the following quantity:

\[ \text{Mean temp. diff.} = \frac{Q_a - Q_b}{\log_e \frac{Q_a}{Q_b}} \]

where

\[ Q_a = T_2 - t_1 \]
\[ Q_b = t_2 - T_1 \]

This mean temperature difference is used under certain conditions and assumptions, mainly:

1. The outside is perfectly insulated.
2. Coefficient of heat transfer \( U \) is constant.
3. \( \omega_c p \) is constant. (\( c_p \) = sp. heat under constant pressure)

These conditions were not fulfilled in the experiment, and the mean temperature difference has been calculated graphically by drawing the temperature gradient in each side of the pipe wall. The area between the two curves divided by the length of the pipe gives the mean temperature difference. It is found that this mean differs by \( \pm 10\% \) from the logarithmic mean. This procedure gives the idea that the approximation resulting from the use of log-mean difference is quite satisfactory.
Observation and Results of Heat Exchanger Experiment

cross sectional area of 9 in. pipe = 0.442 ft.²
surface area of 9 in. pipe/ft. = 2.36 ft.²
cross sectional area of 6 in. pipe = 0.196 ft.²
surface area of 6 in. pipe/ft. = 1.57 ft.²
annular space area = 0.246 ft.²

Surface areas for heat transfer

From point 1 to point 5
outside pipe 9 in. = 2.36 x 48.1 = 113.6 ft.²
inside pipe 6 in. = 1.57 x 48.1 = 75.6 ft.²

Calculation of coefficient of heat transfer theoretically from experiment

\[ t_1 = \text{temperature of inlet for coming cold air} \]
\[ T_1 = \text{temperature of outlet for coming cold air} \]
\[ t_2 = \text{temperature of inlet of room warm air} \]
\[ T_2 = \text{temperature of outlet of room warm air} \]
\[ U_1 = \text{overall coefficient of heat transfer between warm air in 6 in. pipe and cold air in annular space. Btu.}/(\text{hr.})(\text{ft.}^2)(^\circ\text{F.}) \]

\[ U_2 = \text{overall coefficient of heat transfer between warm air in the room and cold air in annular space. Btu.}/(\text{hr.})(\text{ft.}^2)(^\circ\text{F.}) \]

\[ \Omega m_1 = \text{log mean temperature difference between warm air in 6 in. pipe and cold air} \]

\[ \Omega m_1 = \frac{(T_2 - t_1) - (t_2 - T_1)}{\log_e \frac{T_2 - t_1}{t_2 - T_1}} = \frac{Qa - Qb}{\log_e \frac{Qa}{Qb}} \]

\[ \Omega m_2 = \text{log mean temperature difference between room warm air and cold air} \]

\[ \Omega m_2 = \frac{(t - t_1) - (t - T_1)}{\log_e \frac{t - t_1}{t - T_1}} \]

The first reading in Table 8 is taken as an example for the calculation of the coefficients of heat transfer.

1. Weight of cold air = 240 lb./hr.
   Volume of cold air = 3081 cu.ft./hr.

2. Weight of warm air = 175 lb./hr.
   Volume of warm air = 2210 cu.ft./hr.

3. Average velocity in 6 in. pipe = 3.15 ft./sec.
   Average velocity in annular space = 3.5 ft./sec.
Equivalent diameter used in calculation for $h_2$ in the equation $h_c = h_0 F_d F_v$

$$D_e = \frac{D_2^2 - D_1^2}{D_1} = .625 \text{ ft.} = 7\frac{1}{2} \text{ in.}$$

**Theoretical calculation**

From Tables 5, 6 and 7, $h_c$ for forced convection can be calculated if we know the diameter, the velocity and the average temperature of air.

$$h_1 = .495 \times .7 \times 2.5$$

$$= .866 \text{ Btu./hr.}(\text{ft.}^2)(\text{deg. F. Diff.})$$

$$h_2 = .51 \times .67 \times 2.7 = .915 \text{ Btu./hr.}(\text{ft.}^2)(\text{deg. F. diff.})$$

$$\frac{1}{U_1} = \frac{1}{h_1} + \frac{1}{h_2}$$

$$U_1 = .445 \text{ Btu./hr.}(\text{ft.}^2)(\text{deg. F. diff.})$$

For the calculation of $U_2$ we have to calculate $h_4$ which consists of $h_{4c}$ due to convection and $h_{4r}$ due to radiation.

$$h_{4c} = .23 \left( \frac{9}{.75} \right) = .23 \left( \frac{9}{.75} \right)$$

$$= .43 \text{ Btu./hr.}(\text{ft.}^2)(\text{deg. F. Diff.})$$

$$h_{4c} = \frac{.174 \times .228 \left( (5.36)^2 - (5.28)^2 \right)}{77 - 68}$$

$$= 0.265 \text{ Btu./hr.}(\text{ft.}^2)(\text{deg. F.})$$
Therefore,

\[ h_4 = 0.43 + 0.265 = 0.695 \text{ Btu./}(\text{hr.})(\text{ft.}^2)(\text{deg. F.}) \]

\[ \frac{1}{U_2} = \frac{1}{h_4} + \frac{1}{h_3} \]

\[ = \frac{1}{0.695} + \frac{1}{0.915} \]

\[ U_2 = 0.4 \text{ Btu./}(\text{hr.})(\text{ft.}^2)(\text{deg. F. diff.}) \]

**Calculation from experiment**

\[ Q_{m1} = \frac{(67-47.2)-(77-71)}{\log_e \frac{19.8}{6}} = 11.6^\circ \text{ (graphically = 10.8^\circ)} \]

\[ Q_{m2} = \frac{(77-47.2)-(77-71)}{\log_e \frac{29.8}{6}} = 14.88 \text{ (graphically = 14^\circ)} \]

Total heat gained by cold air = \( M_1 \text{ cp } (T_1-t_1) \)

= 240 \times 0.24 \times 23.8

= 1370 \text{ Btu./hr.}

Loss of heat from moving warm air = \( M_2 \text{ cp } (t_2-T_2) \)

= 175 \times 0.24 \times 10

= 420 \text{ Btu./hr.}

Loss of heat from still room air = 1370 - 420

= 950 \text{ Btu./hr.}

\[ 420 = U_1 x \pi D_1 L Q_{m1} \]

\[ 420 = U_1 x 75.6 \times 10.8 \]

\[ U_1 = \frac{.51 \text{ Btu./}(\text{hr.})(\text{ft.}^2)(\text{deg. F. diff.})}{10.8} \]

\[ 950 = U_2 \pi D_2 L Q_{m2} \]

\[ 950 = U_2 x 113.8 \times 14 \]

\[ U_2 = \frac{.59 \text{ Btu./}(\text{hr.})(\text{ft.}^2)(\text{deg. F. diff.})}{14} \]
Table 8(a)
Heat Exchanger
Temperature Readings

<table>
<thead>
<tr>
<th>No.</th>
<th>Pt.1</th>
<th>Pt.2</th>
<th>Pt.3</th>
<th>Pt.5</th>
<th>#/hr.</th>
<th>temp</th>
<th>Temp</th>
<th>Pt.1</th>
<th>Pt.2</th>
<th>Pt.3</th>
<th>Pt.5</th>
<th>Remarks</th>
</tr>
</thead>
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<tr>
<td>1</td>
<td>71</td>
<td>69.2</td>
<td>62.2</td>
<td>47.2</td>
<td>240</td>
<td>34</td>
<td>77</td>
<td>77</td>
<td>76</td>
<td>74</td>
<td>67</td>
<td>175 A booster fan 8 in. diam. of blades is used in pushing cold air.</td>
</tr>
<tr>
<td>2</td>
<td>65.8</td>
<td>61.8</td>
<td>54.2</td>
<td>40.7</td>
<td>240</td>
<td>30</td>
<td>71</td>
<td>71</td>
<td>68</td>
<td>64.8</td>
<td>62</td>
<td>175 It is broken in March 19 and replaced by a vacuum cleaner fan on</td>
</tr>
<tr>
<td>3</td>
<td>70.5</td>
<td>-</td>
<td>-</td>
<td>42</td>
<td>240</td>
<td>30</td>
<td>76</td>
<td>76</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>175 Volume of cold air = 3081 cu. ft./hr.</td>
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<tr>
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<td>65.9</td>
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<td>32</td>
<td>75</td>
<td>75</td>
<td>72</td>
<td>69</td>
<td>64</td>
<td>175</td>
</tr>
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<td>58</td>
<td>44.8</td>
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<td>34</td>
<td>75.5</td>
<td>75.5</td>
<td>72.3</td>
<td>69.8</td>
<td>66</td>
<td>175 April 2, 1947</td>
</tr>
<tr>
<td>6</td>
<td>71.5</td>
<td>-</td>
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<td>45</td>
<td>240</td>
<td>34</td>
<td>76.2</td>
<td>76.2</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>175 Volume of warm air = 2210 cu. ft./hr.</td>
</tr>
<tr>
<td>7</td>
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<td>55.5</td>
<td>43</td>
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<td>32</td>
<td>75.5</td>
<td>75.5</td>
<td>72.5</td>
<td>69.2</td>
<td>65.5</td>
<td>175</td>
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Wt. lbs. Room temp: 
Vs.
### Table 9(a)
#### Heat Exchanger
**Calculated Heat Transfer Coefficients**

<table>
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<tr>
<th>No.</th>
<th>( t_1 )</th>
<th>( T_1 )</th>
<th>( T_1 - t_1 )</th>
<th>( t_2 )</th>
<th>( T_2 )</th>
<th>( t_2 - T_2 )</th>
<th>( \theta_m1 )</th>
<th>( \theta_m2 )</th>
<th>( \text{Loss from air} )</th>
<th>( \text{Loss from air} )</th>
<th>( U_1 )</th>
<th>( U_2 )</th>
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<tr>
<td>1</td>
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<td>71.0</td>
<td>23.8</td>
<td>77</td>
<td>67</td>
<td>10</td>
<td>29.8:10.8:14</td>
<td>1370</td>
<td>420</td>
<td>950</td>
<td>.51</td>
<td>.59</td>
</tr>
<tr>
<td>2</td>
<td>40.7</td>
<td>65.8</td>
<td>25.1</td>
<td>71</td>
<td>62</td>
<td>9</td>
<td>30.3:11.4:15</td>
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<td>1067</td>
<td>.44</td>
<td>.625</td>
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<td>70.5</td>
<td>28.5</td>
<td>76</td>
<td>62.7</td>
<td>13.3</td>
<td>34:11.4:16.5</td>
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<td>558</td>
<td>1082</td>
<td>.645</td>
<td>.575</td>
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<td>75</td>
<td>64</td>
<td>11</td>
<td>32:11.2:15.4</td>
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<td>.62</td>
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<td>26</td>
<td>75.5</td>
<td>66</td>
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<td>.645</td>
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<td>420</td>
<td>1135</td>
<td>.46</td>
<td>.62</td>
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Average: .502: .605

Theor.: .445: .4
### Table 8(b)

**Heat Exchanger Temperature Readings**

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<th>No.</th>
<th>Pt.1</th>
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<th>Pt.3</th>
<th>Pt.5</th>
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<th>Cold air temp.</th>
<th>Warm air temp.</th>
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FIG. 9 MEAN TEMPERATURE DIFFERENCE IN HEAT EXCHANGER. (NO HEATER)
Table 9 (b)

Heat Exchanger
Calculated Heat Transfer Coefficients

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<tr>
<th>No.</th>
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<th>T₁</th>
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<th>t₂</th>
<th>T₂</th>
<th>t₂-T₂</th>
<th>Qm₁</th>
<th>Qm₂</th>
<th>Total Loss</th>
<th>Loss from:</th>
<th>Loss:</th>
<th>Total from:</th>
<th>Mov:</th>
<th>Heat gained:</th>
<th>Still:</th>
<th>Averag</th>
<th>Theor.</th>
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<td>5.0</td>
<td>12</td>
<td>5.57</td>
<td>7</td>
<td>1018</td>
<td>521</td>
<td>497</td>
<td></td>
<td>1.25</td>
<td>.645</td>
<td></td>
<td></td>
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<td>70.6</td>
<td>12.3</td>
<td>75.5</td>
<td>68.5</td>
<td>8</td>
<td>18.2</td>
<td>8.2</td>
<td>9.5</td>
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<td>1.36</td>
<td>.7</td>
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<td>67</td>
<td>60</td>
<td>7</td>
<td>16.7</td>
<td>7.8</td>
<td>8.2</td>
<td>1390</td>
<td>740</td>
<td>650</td>
<td></td>
<td>1.25</td>
<td>.675</td>
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<td></td>
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</tbody>
</table>

Average: 1.3 : .77

Theor.: .95 : .7
Table 8 (c)

Heat Exchanger
Temperature Readings

<table>
<thead>
<tr>
<th>Read. No.</th>
<th>Cold air temp.</th>
<th>Wt.</th>
<th>Side</th>
<th>Room</th>
<th>Warm air temp.</th>
<th>Wt.</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pt. 1</td>
<td>Pt. 2</td>
<td>Pt. 3</td>
<td>Pt. 5</td>
<td>#/hr.</td>
<td>temp.</td>
<td>temp.</td>
<td>Pt. 1</td>
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<tr>
<td>17</td>
<td>70.2</td>
<td>68.2</td>
<td>63.2</td>
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<td>440</td>
<td>48</td>
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</tr>
<tr>
<td>18</td>
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<td>67.2</td>
<td>65</td>
<td>63</td>
<td>440</td>
<td>54</td>
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<td>73.5</td>
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<td>69.3</td>
</tr>
<tr>
<td>20</td>
<td>74</td>
<td>72</td>
<td>69</td>
<td>67</td>
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<td>70.5</td>
</tr>
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<td>21</td>
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<td>70</td>
<td>62</td>
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<td>70.6</td>
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<td>60</td>
<td>50.5</td>
<td>405</td>
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<td>66</td>
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<td>67</td>
<td>61</td>
<td>51</td>
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<td>405</td>
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<td>75</td>
<td>73.5</td>
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<td>405</td>
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<td>70.5</td>
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<td>27</td>
<td>70.6</td>
<td>64.2</td>
<td>62.4</td>
<td>58.4</td>
<td>405</td>
<td>49.5</td>
<td>62</td>
</tr>
</tbody>
</table>
FIG. 10 MEAN TEMPERATURE DIFFERENCE IN HEAT EXCHANGER. (USING HEATER)
### Table 9(c)

<table>
<thead>
<tr>
<th>No.</th>
<th>t1 (°F)</th>
<th>t2 (°F)</th>
<th>t1-t1</th>
<th>t2-t2</th>
<th>t2-t1</th>
<th>Total Loss</th>
<th>Loss from Heat Gained Still Air</th>
<th>Aver.</th>
<th>Theor.</th>
</tr>
</thead>
<tbody>
<tr>
<td>17</td>
<td>70.2</td>
<td>71.5</td>
<td>1.3</td>
<td>13.7</td>
<td>15.0</td>
<td>1130</td>
<td>1005</td>
<td>1.968</td>
<td>-</td>
</tr>
<tr>
<td>18</td>
<td>69.3</td>
<td>72.5</td>
<td>3.2</td>
<td>10.8</td>
<td>14.0</td>
<td>665</td>
<td>744</td>
<td>0.86</td>
<td>-</td>
</tr>
<tr>
<td>19</td>
<td>69</td>
<td>76.5</td>
<td>7.5</td>
<td>8.5</td>
<td>16.0</td>
<td>10.5</td>
<td>20.5</td>
<td>0.86</td>
<td>-</td>
</tr>
<tr>
<td>20</td>
<td>67</td>
<td>72</td>
<td>5.5</td>
<td>9.8</td>
<td>15.3</td>
<td>10.5</td>
<td>20.5</td>
<td>0.86</td>
<td>-</td>
</tr>
<tr>
<td>21</td>
<td>57</td>
<td>72</td>
<td>15</td>
<td>31.5</td>
<td>46.5</td>
<td>1.5</td>
<td>31.5</td>
<td>0.86</td>
<td>-</td>
</tr>
<tr>
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<td>52.3</td>
<td>75</td>
<td>22.7</td>
<td>38.2</td>
<td>60.9</td>
<td>2200</td>
<td>1440</td>
<td>0.95</td>
<td>-</td>
</tr>
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<td>23</td>
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<td>75</td>
<td>20.5</td>
<td>34.8</td>
<td>55.3</td>
<td>10.3</td>
<td>34.8</td>
<td>0.85</td>
<td>-</td>
</tr>
<tr>
<td>24</td>
<td>51</td>
<td>73</td>
<td>22</td>
<td>36.6</td>
<td>58.6</td>
<td>22.0</td>
<td>20.2</td>
<td>0.85</td>
<td>-</td>
</tr>
<tr>
<td>25</td>
<td>57.5</td>
<td>72.5</td>
<td>15</td>
<td>28.2</td>
<td>43.2</td>
<td>14.6</td>
<td>14.6</td>
<td>0.85</td>
<td>-</td>
</tr>
<tr>
<td>26</td>
<td>65.5</td>
<td>75</td>
<td>9.5</td>
<td>15.7</td>
<td>25.2</td>
<td>8.5</td>
<td>20.1</td>
<td>0.85</td>
<td>-</td>
</tr>
<tr>
<td>27</td>
<td>58.4</td>
<td>70.6</td>
<td>12.2</td>
<td>17.4</td>
<td>31.2</td>
<td>5.5</td>
<td>17.4</td>
<td>0.85</td>
<td>-</td>
</tr>
</tbody>
</table>
Four groups of the experiment were made for obtaining the actual coefficient of heat transfer; namely:

1. Cold air was pushed with a booster fan, and the exhaust air and room temperature were nearly the same.

2. Same as No. 1 but volume of cold air was increased by use of a vacuum cleaner fan.

3. Warm air was heated before being pushed into the inside pipe and so it was not of the same temperature as the room temperature.

4. Same as No. 3 but the volume of warm air was increased by increasing the speed of the blower, and it was heated by the use of two heaters.

Experiments 3 and 4 were done when the outside cold air temperature became near to the room temperature.

Results

Two main results were clear from the figures obtained from the experiment.

I. A. The per cent of heat transmitted from the still air of the room to the total heat obtained was high. This was due to the fact that the 9 inch pipe surface was great with respect to the 6 inch pipe surface. Since our aim is to obtain the maximum amount of heat from
the exhaust air, we have to increase the surface area of transmission between the exhaust air and cold air. This can be obtained by using several small pipes in the 9 inch pipe. If we use three 3 inch pipes, the surface will increase 150%.

B. 1. As the velocity increased in the small pipe, the coefficient of heat transfer increased too. If we replace the 6 inch pipe by three 3 inch pipes the cross sectional area will decrease by 25% (6 inch pipe area = .196 ft.$^2$, 3 pipes 3 inch area = .147 ft.$^2$).

2. Actual coefficient of heat transfer between warm exhaust air and cold air was greater than that calculated theoretically due to extra turbulence occurring in the annular space. This extra turbulence came from the supports keeping the inside pipe in place and the fact that the pipes were not perfectly straight.

The outside surface coefficient, in this experiment, differed from the theoretical one due to the variable surrounding conditions which affected convection and radiation.
For convection, in groups 3 and 4 when the difference of temperature of the coming cold air and room was small, the coefficient of heat transfer was also small.

In some cases, in a certain part of the pipe, the heat was transferred from the room to the cold air; in other parts, heat transfer was stopped; while in the third part heat was transferred from air in the annular space to the room. The opening and closing of the laboratory doors caused currents in the still air. Heat radiators near the walls had the same effect of making continuous movement in the surrounding air. All these factors combined together helped in increasing the coefficient of heat transfer by convection for the outside 9 inch pipe wall.

For heat transfer coefficient due to radiation, conditions were always variable due to the position of the pipes. The pipes were put in the space of a monitor which was surrounded by windows. On clear days a greater coefficient of radiation was expected than on cloudy days. The temperature of the pipe wall was also variable in the different sections and changed from day to day, affecting the coefficient of heat transfer. This coefficient
is directly proportional to the difference between the fourth power of the absolute temperature of the wall of the pipe and that of the surroundings, and inversely proportional to the difference between these temperatures.

II. Cold air in passing over the motor of the fan is warmed up. The amount of heat gained depends upon the mass of air blown, the size of the motor, and the way that this air passes over the motor. If the fan is of the propeller type and is put in a box where all the coming air will pass over the motor, the gain will be maximum. This can be clearly noticed in the results obtained.

For determining the heat generated by a motor, the input and efficiency must be known. This input in watts per hour multiplied by the coefficient \((1 - \text{efficiency})\) multiplied by 3.4 will give the heat output of the motor.

Heat from Booster Fan

The existence of the fan will add energy to the air flowing from the outside to the annular space. Thermodynamically, this case can be treated as a steady flow process. If we consider the two sections 1 and 2 just before and after the fan, we will get:
Where \( V_1 \) and \( V_2 \) = velocities at sec. 1 and 2 in ft./sec.

\[ \frac{V_1^2}{2g} + Jh_1 \\downarrow \downarrow + \downarrow W_2 + Q_c = \frac{V_2^2}{2g} \uparrow + \uparrow Jh_2 \]

where \( V_1 \) and \( V_2 \) = velocities at sec. 1 and 2 in ft./sec.

\( h_1 \) and \( h_2 \) = enthalpy at sec. 1 and 2 in Btu./lb.

\( W_2 \) = input energy in ft. lb.

\( Q_c \) = heat gained from the surroundings Btu./lb.

\( V_1 \) can be considered very small and negligible.

\[ \therefore W_2 = \frac{V_2^2}{2g} + J(h_2 - h_1) - Q_c \]

\[ = \frac{V_2^2}{2g} + Jc_p \delta t - Q_c \]

The input of the fan can be calculated by measuring the consumption of electricity. A part of this input is dissipated as heat losses to the air and a part as an increase in the kinetic energy.

In our experiment we can measure the quantity \((t_1 - t_2)\). \( Q_c \), therefore, can be known and from which we can evaluate the coefficient of heat transfer from the outside warm room air to the inside cold air.

Knowing this coefficient of heat transfer, velocity of air leaving the fan, mass of air circulated, static pressure of the system and the efficiency of the motor, the expected change in temperature of air before and after the fan can be calculated.
Table 10
Temperature at Outside and Point (5)

<table>
<thead>
<tr>
<th>Reading No.</th>
<th>Outside temp.</th>
<th>Temp. at 5</th>
<th>Reading No.</th>
<th>Outside temp.</th>
<th>Temp. at 5</th>
</tr>
</thead>
<tbody>
<tr>
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<td>34</td>
<td>47.2</td>
<td>4</td>
<td>32</td>
<td>43</td>
</tr>
<tr>
<td>2</td>
<td>30</td>
<td>40.7</td>
<td>5</td>
<td>34</td>
<td>44.9</td>
</tr>
<tr>
<td>3</td>
<td>30</td>
<td>42</td>
<td>6</td>
<td>34</td>
<td>45</td>
</tr>
</tbody>
</table>

Part of the change in temperature is due to the gain of heat from the fan while the other part is due to the gain from the warm room air.
Calculation of the Proposed Heat Exchanger

Heat exchanger type No. 1 with 6000 cu.ft./hr.

Volume of air = 6000 cu.ft./hr.
Length of heat exchanger = 60 ft.
3 inside pipes 3 in. diam. while the outside is 9 in. pipe.
Area of 9 in. pipe = .442 ft.²
Area of three 3 in. pipes = .147 ft.²
Area of space between pipes = .295 ft.²
Velocity of warm air in small pipes = 11.33 ft./sec.
Velocity of cold air in space between pipes = 5.65 ft./sec.
Equivalent diam. = \(\frac{4A}{\pi} = .5\) ft. = 6 in.

\[ h_c = h_0 \frac{F_d}{F_v} \]

\[ h_1 = .517 \times .8 \times 6.9 = 2.86 \quad \text{Btu.}/(\text{hr.})(\text{ft.}^2)(\text{deg. F.}) \]

\[ h_2 = .52 \times .7 \times 4 = 1.46 \quad \text{Btu.}/(\text{hr.})(\text{ft.}^2)(\text{deg. F.}) \]

Overall coefficients \( U_1 = .967 = 1.00 \quad \text{Btu.}/(\text{hr.})(\text{ft.}^2)(\text{°F.}) \)

\( h_3 = 1.46 \quad \text{Btu.}/(\text{hr.})(\text{ft.}^2)(\text{deg. F.}) \)

\( h_4 \) consists of \( h_{4c} \) due to convection and \( h_{4r} \) due to radiation.

Consider cold waves of 0°F. or below, warm air at 35°F and surrounding walls about 35°F.

\[ h_{4c} = .23\left(\frac{23}{35}\right)\frac{1}{3} = .23\left(\frac{23}{.75}\right)^{\frac{1}{3}} = .54 \quad \text{Btu.}/(\text{hr.})(\text{ft.}^2)(\text{°F.}) \]
\[ h_4r = 0.174 \times 0.228 \left[ \frac{(4.95)^4 - (4.72)^4}{35 - 12} \right] \]
\[ = 0.18 \text{ Btu/}(\text{hr.})/(\text{ft.}^2)/\text{(deg. F.)} \]
\[ h_4 = 0.54 + 0.18 = 0.72 \text{ Btu/}(\text{hr.})/(\text{ft.}^2)/\text{(deg. F.)} \]

Overall coefficient \( U_2 = \frac{1}{1.48 + 0.72} = 0.5 \text{ Btu/}(\text{hr.})/(\text{ft.}^2)/\text{(°F.)} \)

Weight of air circulated = \( 6000 \times 0.08 = 480 \text{ lb./hr.} \)

Area of heat transfer from 3 in. pipes, length 60 ft. = 141.5 ft.^2

Area of heat transfer from 9 in. pipe = 141.5 ft.^2

\[ Q_a = T_2 - t_1 \]
\[ Q_b = t_2 - t_1 \]
\[ Q_c = t_2 - t_1 \]

\( Q_{ml} = \log \text{ mean temperature difference between warm air in small pipes and cold air.} \)
\[ Q_{a} - Q_{b} \]
\[ Q_{ml} = \frac{\log Q_{a}}{\log Q_{b}} \]

\( Q_{m2} = \log \text{ mean temperature difference between house warm air and cold air.} \)
\[ \frac{Q_{c} - Q_{b}}{\log Q_{b}} \]

Total heat gained by cold air = \( 480 \times 0.24 (T_1 - t_1) \)
\[ = 115.2 (T_1 - t_1) \quad (1) \]

Heat lost from warm air in pipes = \( 115.2(t_2 - T_2) \)
\[ = U_1 \times 141.5 \times Q_{ml} \]
\[ = 141.5 Q_{ml} \quad (3) \]
Heat lost from still house air = $U_2 \times 141.5 Q_{m2}$

= $.5 \times 141.5 Q_{m2}$

= $70.75 Q_{m2}$ \hspace{1cm} (4)

### Table 11

Heat Exchanger Type I-6000 cu.ft./hr. - Counter Flow

<table>
<thead>
<tr>
<th>Outside temp.</th>
<th>Inside temp.</th>
<th>$T_1 \degree F$</th>
<th>$T_2 \degree F$</th>
<th>Total heat gained</th>
<th>Loss from warm air</th>
<th>Loss from still air</th>
</tr>
</thead>
<tbody>
<tr>
<td>-15</td>
<td>35</td>
<td>23.75</td>
<td>12.5</td>
<td>4460</td>
<td>2600</td>
<td>1860</td>
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<tr>
<td>-5</td>
<td>35</td>
<td>25.5</td>
<td>17.5</td>
<td>3515</td>
<td>2015</td>
<td>1500</td>
</tr>
<tr>
<td>0</td>
<td>35</td>
<td>27.1</td>
<td>19.3</td>
<td>3120</td>
<td>1810</td>
<td>1310</td>
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<td>10</td>
<td>35</td>
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<td>2210</td>
<td>1300</td>
<td>910</td>
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<tr>
<td>20</td>
<td>35</td>
<td>31.5</td>
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<td>547</td>
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<td>35</td>
<td>32.75</td>
<td>30.7</td>
<td>892</td>
<td>520</td>
<td>372</td>
</tr>
</tbody>
</table>

Heat exchanger type No. 1 with 4000 cu.ft./hr.

Weight of air = 4000 x .08 = 320 lb./hr.

Velocity of air in small pipes = 7.55

Velocity of air in space between pipes = 3.75 ft./sec.

$h_1 = .517 \times .8 \times 5.03 = 2.08 \text{ Btu.)/(hr.)(ft.}^2\text{)(deg. F.)}$

$h_2 = .52 \times .7 \times 2.88 = 1.05 \text{ Btu.)/(hr.)(ft.}^2\text{)(deg. F.)}$
HEAT EXCHANGER
LENGTH 60', 6000 C.U.H/HR
COUNTER FLOW

CHANGE IN TEMPERATURE DEG. F.

HEAT GAINED FROM EXHAUST AIR

TOTAL HEAT GAINED

RISE IN COLD AIR

HEAT GAINED FROM EXHAUST

DECREASE IN WARM AIR

FIG. 11 HEAT EXCHANGER TYPE 1
overall coefficient $U_1 = \frac{1}{2.08 + \frac{1}{1.05}}$

$= .7 \text{ Btu.} / (\text{hr.}) (\text{ft.}^2)(\text{deg. F.})$

overall coefficient $U_2 = \frac{1}{1.05 + \frac{1}{.72}}$

$= .43 \text{ Btu.} / (\text{hr.}) (\text{ft.}^2)(\text{deg. F.})$

Total heat gained by cold air = $320 \times .24 (T_1 - t_1)$

$= 76.8 (T_1 - t_1)$  \hspace{1cm} (1)

Heat lost from moving warm air = $76.8 (t_2 - T_2)$  \hspace{1cm} (2)

$= U_1 \times 141.5 \Omega_m$

$= 99.05 \Omega_m$  \hspace{1cm} (3)

Heat lost from still warm air = $U_2 \times 141.5 \Omega_m$

$= 60.8 \Omega_m$  \hspace{1cm} (4)

Table 12

Heat Exchanger Type I-b 4000 cu.ft./hr. Counter Flow

<table>
<thead>
<tr>
<th>Outside temp.</th>
<th>Inside temp.</th>
<th>$T_1$ °F.</th>
<th>$T_2$ °F.</th>
<th>Total heat gained</th>
<th>Loss from warm air</th>
<th>Loss from still air</th>
</tr>
</thead>
<tbody>
<tr>
<td>-15</td>
<td>35</td>
<td>26</td>
<td>13</td>
<td>3140</td>
<td>1690</td>
<td>1450</td>
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<tr>
<td>0</td>
<td>35</td>
<td>28.5</td>
<td>19.75</td>
<td>2195</td>
<td>1185</td>
<td>1010</td>
</tr>
<tr>
<td>20</td>
<td>35</td>
<td>32</td>
<td>28.5</td>
<td>924</td>
<td>500</td>
<td>424</td>
</tr>
</tbody>
</table>
Fig. 12 HEAT EXCHANGER TYPE 1

CHANGE IN TEMPERATURE

HEAT EXCHANGER
LENGTH 60' 4000 CFM/HR.
COUNTER FLOW

9" PIPE COLD
WASH

W

3" PIPE

TOTAL HEAT GAINED

GAINED FROM EXHAUST

TEMP. RISE IN COLD AIR

TEMP. DECREASE IN WARM AIR

t₂ - t₁ = HOUSE TEMP. - OUTSIDE TEMP.
Heat exchanger type No. 2 with 6000 cu.ft./hr.

Length 20 ft. having 12-2 in. inside pipes, outside duct 9 in.

Heat exchanger type 1 has a length of 60 feet. At the same time the amount of heat gained from the still room air is comparatively high with respect to that gained from the exhaust air. To simplify the heat exchanger and to lessen the amount of heat gained from the still air, type 2 is suggested. The length of the heat exchanger is 20 feet. The outside square duct side is 9 inches in dimension. Warm air flows through twelve small pipes, each 2 inches in diameter.

This type can be taken as a unit in poultry houses for each 100 birds.

For its calculation the same methods of treatment are used.

Area of small pipes = .262 ft.²
Area of space between pipes = .3 ft.²
Velocity of warm air in small pipes = \( \frac{6000}{.262 \times 3600} = 6.4 \text{ ft./sec.} \)

Velocity of cold air in space between pipes = 5.55 ft./sec.

Equivalent diameter for getting the surface coefficient for small pipes:

Wetted perimeter for 12-2 in. pipes/ft. run = 6.28 ft.²
De = $\frac{4A}{p} = \frac{4 \times 3}{8.28} = 0.191 \text{ ft.} = 2.3 \text{ in.}$

$h_c = h_0 F_d F_v$

$h_1 = 0.517 \times 0.87 \times 4.41 = 1.98 \text{ Btu/}(\text{hr.})(\text{ft.}^2)(\text{deg. F.})$

$h_2 = 0.52 \times 0.845 \times 3.94 = 1.73 \text{ Btu/}(\text{hr.})(\text{ft.}^2)(\text{deg. F.})$

$U_1 = \frac{1}{1 - \frac{1}{1.95 + 1.73}} = 0.92 \text{ say } 1.00 \text{ Btu/}(\text{hr.})(\text{ft.}^2)(\text{deg. F.})$

Equivalent diameter for getting the inside surface coefficient for the duct.

$D_e = \frac{4x3}{9} = 0.4 \text{ ft.} = 4.8 \text{ in.}$

$h_3 = 0.52 \times 0.732 \times 3.94 = 1.5 \text{ Btu/}(\text{hr.})(\text{ft.}^2)(\text{deg. F.})$

$h_4 = h_4c$ for convection and

$h_4r$ for radiation

General formula is

$h_c = Ck \left( \frac{a}{L} \right)^{1/4}$

Where $C = 0.55$ for vertical plate; $0.71$ for horizontal plate facing upwards; $0.35$ for horizontal plate facing downward.

An average value for $C$ will be $0.54$.

$L = \text{vertical or horizontal length of the plate.}$

$= \frac{9}{\sqrt{2}} = 6.35 \text{ inches} = 0.53 \text{ ft.}$

$a = 2.4 \times 10^6$ for air at $20^\circ$

$k = 0.0128$ for air at $20^\circ$
\[ h_{ac} = 0.54 \times 0.128 \times (2.6 \times 10^6)^{\frac{1}{4}} \left( \frac{Q}{L} \right)^{\frac{1}{4}} \]
\[ = 0.28 \left( \frac{Q}{L} \right)^{\frac{1}{4}} = 0.28 \left( \frac{23}{53} \right)^{\frac{1}{4}} = 0.72 \text{ Btu./hr.}(\text{ft.}^2)(\text{deg. F.}) \]

\[ h_{ar} = 0.174 \times 0.228 \sqrt{\frac{(4.95)^4 - (4.72)^4}{35 - 12}} = 0.18 \text{ Btu./hr.}(\text{ft.}^2)(\text{deg. F.}) \]

\[ h_4 = 0.72 + 0.18 = 0.9 \text{ Btu./hr.}(\text{ft.}^2)(\text{deg. F.}) \]

\[ U_2 = \frac{1}{1.5 + 0.9} = 0.57 \text{ Btu./hr.}(\text{ft.}^2)(\text{deg. F.}) \]

Weight of air circulated = 480 lb./hr.

Area of heat transfer of 12-2 in. pipes, length 20 ft. = 125.5 ft.²

Area of heat transfer of duct 9 in. side, length 20 ft. = 60 ft.²

Total heat gained by cold air = 480 \times 0.24 (T_1-t_1) 
\[ = 115.2 \times (T_1-t_1) \quad (1) \]

Heat lost from exhaust air 
\[ = 115.2 \times (t_2-T_2) \quad (2) \]

\[ = U_1 \times 125.5 \times Q_{ml} \]
\[ = 125.5 \times Q_{ml} \quad (3) \]

Heat lost from still air 
\[ = U_2 \times 60 \times Q_{m2} \]
\[ = 34.2 \times Q_{m2} \quad (4) \]
Table 13

Heat Exchanger Type 2 - 6000 cu.ft./hr. Counter Flow

<table>
<thead>
<tr>
<th>Outside temp. °F</th>
<th>Inside temp. °F</th>
<th>$T_1$ °F</th>
<th>$T_2$ °F</th>
<th>Total heat gained</th>
<th>Loss from warm air</th>
<th>Loss from still air</th>
</tr>
</thead>
<tbody>
<tr>
<td>-15</td>
<td>35</td>
<td>17.7</td>
<td>11.5</td>
<td>3770</td>
<td>2710</td>
<td>1060</td>
</tr>
<tr>
<td>0</td>
<td>35</td>
<td>22.9</td>
<td>18.5</td>
<td>2640</td>
<td>1900</td>
<td>740</td>
</tr>
<tr>
<td>20</td>
<td>35</td>
<td>29.7</td>
<td>28</td>
<td>1120</td>
<td>810</td>
<td>310</td>
</tr>
</tbody>
</table>

Effect of Condensation

In the previous calculations for the different types of heat exchangers, the effect of condensation of moisture in the air is not taken into account. Moisture carried by warm air, will condense when this air gets in contact with the cold surface. Condensation will be in the small pipes where warm air runs to the outside and around the external wall of the outside duct.

This condensation will influence the amount of heat transferred in two ways. The first result is that the coefficient of heat transmission will increase due to the effect of having a wet surface, while the second will be the release of latent heat which will add to the amount of heat exchanged.

Condensation will begin when the temperature of the wall
HEAT EXCHANGER TYPE 2A
LENGTH 20'
6000 CU.FT./HR.
COUNTER FLOW

FIG. 13 HEAT EXCHANGER TYPE 2A
is below the dew point of warm air. This condensation will begin and continue before the whole mass of air has been cooled down to its saturation point.

It is assumed that at the cooling surface there exists a film of air, through which heat and moisture are transmitted by conduction and diffusion. In this film there is less concentration of vapor than in the main body of air mixture but as the surface is wet, the air immediately in contact can be considered saturated at the surface temperature. From the studies made by Keevil and Lewis (21) and Knaus (25) the following formulae are given:

A. if $c_a$, $c_e$, $c_s$ are the concentrations of moisture in entrance, exit and surface of the wall in lb. of moisture/lb. of dry air $t_a$, $t_e$, $t_s$ are temperatures of air at entrance, exit and surface.

\[
\frac{c_a - c_s}{c_e - c_s} = \frac{t_a - t_s}{t_e - t_s}
\]

(1)

and also

\[
\frac{c_a - c_s}{t_a - t_s} = \frac{c_a - c_e}{t_a - t_c}
\]

(1a)

From which we can predict the relative humidity of air at exit if its temperature is known.

B. if $q_s$ = rate of sensible heat removal Btu./hr.
$q_e$ = rate of latent heat removal Btu./hr.
s = humid heat Btu./lb. dry air = 0.245
r = latent heat of condensation of water vapor at
  surface temperature Btu./lb. = 1066
q = total heat = qs + qe
h = sensible heat transmission coefficient Btu./(hr.)
  (ft.²)(°F.)

h' = coefficient of heat transmission of sensible and
  latent heat
Q_m = mean temp. difference

Therefore

\[ h' = h(1 + \frac{r}{s}) \frac{c_a - c_s}{t_a - t_s} \]

\[ h' = h(1 + 4350) \frac{c_a - c_s}{t_a - t_s} \]

\[ q = \frac{AhQ_m}{s(t_a - t_s)} \left[ (st_a + rc_a) - (st_s + rc_s) \right] \]  (3)

\[ st_a + rc_a = \text{total heat at the entering air Btu./lb.} \]

\[ st_s + rc_s = \text{total heat of air saturated at surface} \]

\[ \text{temp. } t_s \text{ in Btu./lb. dry air} \]

Most of the studies on condensation have been made for cooling air by extended surfaces where the cooling surface is of constant temperature or the variation is negligible. At the same time no consideration has been made if this condensate changes to frost. The probability of frost formation is expected in the pipes used for the exhaust air in poultry house ventilation. The exhaust air temperature is
about 35° F. and leaves at a temperature far below 32° F. If warm air is at 35° and 80% relative humidity, 29° F. will be its dew point; at this temperature frost will be formed.

The formation of frost has two direct influences on heat transmission; namely, liberation of more heat and the decrease of coefficient of transmission due to the low conductivity of ice.

For the above two reasons which are the variability of the temperature of the cooling surface and the formation of frost, mathematical solution of the problem will be complicated.
ANALYSIS OF A POULTRY HOUSE

20 ft. x 22 ft. for 100-4 lb. leghorn hens

A heat exchanger of type No. 1 which has been discussed previously will be introduced in a poultry house 20 ft. x 22 ft. This house has a net floor area of approximately 400 sq. ft. which allows 4.0 sq. ft. for each bird. A minimum value for the area of windows is chosen because heat is easily lost through them due to their low resistance to heat flow.

Six windows of 2 ft. x 2 ft. - 6 in. and glass lights 9 in. x 12 in. will give glass area of 18 sq. ft. which corresponds to 4.5% of the floor area. These windows are made of the double type and their coefficient of heat transfer will be 0.45 Btu./(hr.)(sq.ft.)(deg. F.). The house has one door 6 ft. x 2 ft. - 6 in. made of lumber 1 1/2 in. thick. Its coefficient for heat transfer can be considered as 0.59 Btu./(hr.)(sq.ft.)(deg. F.).

The roof is made of rafters 2 in. x 4 in. with lengths of 12 ft. and 10 ft. lined from both sides with 3/4 in. x 6 in. boards and the outside is covered with asphalt shingles. The space between the inside and outside lining is filled with chopped corn stalks used as an insulating material.

The walls are made of the same cross section which
consists of studs 2 in. x 4 in. spaced at 2 feet lined with 3/4 in. x 6 in. boards.

These boards are covered from the inside by waterproofing paper to protect the insulation from getting wet. The space between the inside and outside lining is filled with the same insulating material as the roof. The floor is made of 2 in. concrete layer over a layer of gravel or cinder.

A propeller type fan is used in the heat exchanger and it is put in a box so that air will pass over the motor before it is pushed into the heat exchanger. A thermostat is used to stop the fan when temperature inside the house reaches 35° F.

A fan with a damper of the type discussed by Strahan (40) can also be used. In this type, the damper is controlled by another thermostat. When the temperature reaches a certain limit, the damper is closed and only half of the air is pushed while when temperature reaches its minimum limit, the fan is stopped.

Arrangement of Ventilation System

Ventilation in the 20 ft. x 22 ft. poultry house will be through type 1 heat exchanger. Its length is 60 feet and the outer pipe is of galvanized steel 9 in. in diameter. Warm exhaust air will flow in three 3 in. pipes placed
Figure 14: Poultry House, 20' x 22'
inside the big pipe. Cold air will travel in the space between the small pipes and the big pipe. It will be pushed by a fan put in the east wall where its entrance is protected by a hood D, Fig. 15. The fresh air passes through the space between the pipes until it reaches point E where it is discharged in a 6 in. vertical pipe, which leads to a horizontal perforated pipe discharging in the house. A damper is used to change the amount of air passing from 6000 cu.ft./hr. to 4000 cu.ft./hr.

Exhaust air is taken from F near the ceiling. This point is chosen because the air is warm and contains high percentage of CO₂ and moisture. A heater is added which will only be used when frost accumulates in the small pipes due to condensation.

Since condensation is expected where warm air carrying moisture comes in contact with cold pipe walls, a trough board is furnished underneath the big pipe while vessels C are put in different points to drain the inside pipes. Exhaust air discharges outside at point G in the north wall.

Calculation of Heat Losses

The thermal conductivities of materials of walls and roof are taken from Carter and Foster (9) while that of windows and door are taken from the A.S.H.V.E. guide (1).
Fig. 15. GENERAL ARRANGEMENT FOR VENTILATION SYSTEM
POULTRY HOUSE 28' x 36'
Scale ½" = 1'
In both cases the velocity of wind is assumed at 15 M.P.H.

**Wall:** 3/4 in. siding + 3-5/8 in. chopped corn stalk
3/4 in. inside boards

\[
\frac{1}{U_1} = \frac{1}{1.65} + \frac{3}{4x.8} + \frac{3.625}{.27} + \frac{3}{4x.8} + \frac{1}{8}
\]

\[U_1 = 0.062 \text{ Btu.}/(\text{hr.})(\text{sq. ft.})(\text{deg. F.})\]

**Roof:** Can be considered the same neglecting the effect of asphalt shingles.

**Windows:** for double glass \(U_2 = 0.45 \text{ Btu.}/(\text{hr.})(\text{sq. ft.})(\text{°F.})\)

**Door:** 1-1/4 in. thickness \(U_3 = 0.59 \text{ Btu.}/(\text{hr.})(\text{sq. ft.})(\text{°F.})\)

Area of walls after deducting windows and door = 472 ft.\(^2\)
Area of roof = 462 ft.\(^2\)
Glass = 30 ft.\(^2\)
Wood for windows and door = 15 ft.\(^2\)
Total area = 979 ft.\(^2\)

\(A = \text{area of exposure per bird} = 9.79 \text{ ft.}^2\)

\(C = \text{loss of heat per hr. per sq. ft. per deg. F.} = 0.0845 \text{ Btu.}\)

\(AC = \text{loss of heat per bird per hr. per deg. F.} = 0.829 \text{ Btu.}\)

This house can be considered as a well insulated house.
The insulating material is a by-product of the farm, of high resistance to heat flow and of no cost value.

The Hen as a Heat and Moisture Producer

The engineer who deals with ventilation of poultry
houses should have some knowledge of the production of heat and moisture by the bird.

Mitchell and Kelley (30) have done such a study in which different kinds of birds of different weights and under variable environmental conditions were tested and discussed. The results of their work in heat and moisture production by chickens and turkeys are given in the following two tables.

Table 14 gives the total heat produced in Btu./(hr.) for different weights while Table 15 shows the per cent of heat that can be considered as latent heat under varying environmental temperatures. The formula relating latent heat to the amount of moisture given off by respiration can be used. Sensible heat used in warming the house is the net value of total heat minus latent heat.

Table 14
Heat Production by Chickens and Turkeys

<table>
<thead>
<tr>
<th>Body weight in pounds</th>
<th>Total heat in k. cal./day</th>
<th>Total heat in Btu./hr.</th>
</tr>
</thead>
<tbody>
<tr>
<td>.077</td>
<td>10</td>
<td>1.65</td>
</tr>
<tr>
<td>.5</td>
<td>82</td>
<td>13.56</td>
</tr>
<tr>
<td>1.0</td>
<td>110</td>
<td>18.18</td>
</tr>
<tr>
<td>1.5</td>
<td>132</td>
<td>21.22</td>
</tr>
<tr>
<td>2.0</td>
<td>155</td>
<td>25.62</td>
</tr>
<tr>
<td>3.0</td>
<td>194</td>
<td>32.07</td>
</tr>
<tr>
<td>4.0</td>
<td>235</td>
<td>38.85*</td>
</tr>
<tr>
<td>5.0</td>
<td>281</td>
<td>46.45</td>
</tr>
</tbody>
</table>

*This value will be taken as 40 Btu./hr. in all coming calculations.
Latent heat

\[ P = 7.14 e^{0.06438t} \]

\[ P = \text{per cent of heat produced that is dissipated as the heat of vaporization of water.} \]

\[ t = \text{temperature in centigrade}. \]

Table 15

Per Cent of Heat in Latent Form

<table>
<thead>
<tr>
<th>Temperature °F.</th>
<th>Latent heat % of total</th>
<th>Temperature °F.</th>
<th>Latent heat % of total</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>2.23</td>
<td>60</td>
<td>19.4</td>
</tr>
<tr>
<td>10</td>
<td>3.26</td>
<td>70</td>
<td>27.6</td>
</tr>
<tr>
<td>20</td>
<td>4.66</td>
<td>80</td>
<td>39.3</td>
</tr>
<tr>
<td>30</td>
<td>6.64</td>
<td>90</td>
<td>56.7</td>
</tr>
<tr>
<td>32</td>
<td>7.14</td>
<td>100</td>
<td>81.1</td>
</tr>
<tr>
<td>40</td>
<td>9.5</td>
<td>106</td>
<td>100.0</td>
</tr>
<tr>
<td>50</td>
<td>13.6</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Moisture production

\[ W = \frac{H \cdot L}{0.58} \]

where \( W = \text{grams of water/hr.} \)

\[ W_1 = \frac{H_1 L (15.43)}{2.3} \]

\( W_1 = \text{grains of water/hr.} \)

\( H \cdot L = \text{latent heat in K. Cal.} \)

\( L = \text{per cent of total heat in latent form} \)

\( H_1 L = \text{latent heat in Btu.} \)

0.58 = heat in K. cal. required to vaporize 1 gram of water at skin temperature

2.3 = heat in Btu. required to vaporize 1 gram of water at skin temperature
1 Btu. = 252 calorie 1 pound = 7000 grains
1 K. cal. = 3.97 Btu. 1 gram = 15.43 grain
1 pound = 453.6 grams

Table 16
Water Vapor Production by 4 lb. Leghorn Hen

<table>
<thead>
<tr>
<th>Temperature °F.</th>
<th>Water vapor grains/hr.</th>
<th>Temperature °F.</th>
<th>Water vapor grains/hr.</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>5.8</td>
<td>50</td>
<td>35.5</td>
</tr>
<tr>
<td>10</td>
<td>8.5</td>
<td>60</td>
<td>50.6</td>
</tr>
<tr>
<td>20</td>
<td>12.1</td>
<td>70</td>
<td>72.0</td>
</tr>
<tr>
<td>30</td>
<td>17.3</td>
<td>80</td>
<td>105.0</td>
</tr>
<tr>
<td>32</td>
<td>18.5</td>
<td>90</td>
<td>148.0</td>
</tr>
<tr>
<td>40</td>
<td>24.8</td>
<td>100</td>
<td>211.0</td>
</tr>
</tbody>
</table>

Tables 14, 15 and 16 are represented by figures 16, 17, and 18.

Evaporation of Moisture from the Litter

In a poultry house, it is important to have dry litter. A wet litter is conducive to disease and takes more man hours of labor for turning and changing.

The rate of evaporation of moisture depends upon the available heat, the rate of air movement over the litter, the frequency of turnover of the litter and its depth.

Forty per cent moisture content in the litter is considered quite satisfactory. (10).

A study made by White (44) on the production and compo-
Fig. 16. TOTAL HEAT

Fig. 17. PERCENT LATENT FORM

TRANSLATION:
- Total heat produced per bird in BTU per hour.
- Weight of bird in lbs.

Environmental temp. in deg. F.

Heat produced by the hen.

Fig. 17. Percent latent form.
Fig. 18  LATENT HEAT AND MOISTURE PRODUCTION 4-LB. LEGHORN HEN
position of poultry manure gives data on which we can base our assumptions.

A hen voids daily 171.7 grams of manure (76% moisture) which consists of 41.2 grams of dry matters and 130 grams of water. About 35.4% of this manure is deposited on dropping boards and pits from which no moisture is required to be removed and 64.6% is deposited on the litter.

The manure deposited on litter is therefore 110.9 grams per day which contains 26.6 grams of dry matters and 84.3 grams of moisture.

If the dry matter on the litter will have 40% moisture content, we will therefore keep 17.75 grams and remove 66.53 grams daily which is equivalent to 43 grains per hour.

The total amount of moisture to be removed will be the resultant of that exhaled by the bird under the certain environmental temperature and that coming out from the litter.

**Table 17**

Total Amount of Moisture to be Removed by Ventilation

<table>
<thead>
<tr>
<th>Environmental temperature °F.</th>
<th>Removed moisture grains/hr.</th>
<th>Environmental temperature °F.</th>
<th>Removed moisture grains/hr.</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>48.8</td>
<td>50</td>
<td>78.5</td>
</tr>
<tr>
<td>10</td>
<td>51.5</td>
<td>60</td>
<td>93.6</td>
</tr>
<tr>
<td>20</td>
<td>55.1</td>
<td>70</td>
<td>115.0</td>
</tr>
<tr>
<td>30</td>
<td>60.3</td>
<td>80</td>
<td>148.0</td>
</tr>
<tr>
<td>40</td>
<td>67.8</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Ventilation Design and Equilibrium of Heat Produced with Heat Lost

The amount of air necessary in ventilation to remove water vapor exhaled by the bird and that evaporated from the litter will vary with the variation of these quantities in addition to the temperature and relative humidity of the coming fresh air and that to be retained in the house. A satisfactory relative humidity inside the house can be taken as 80% (3).

In Fig. 21, if we assume that the average temperature in winter is 20°F, the heat produced by the bird will not be enough to keep the house at a temperature of 35°F. Heat produced is lost in vaporizing moisture from the bird's body, from litter, through the building and in warming air if there is ventilation.

Table 18
Required Air to Remove Moisture (Fig. 19)
Coming air at 20°F. and R.H. 80% - Capacity 0.988 grains/cu.ft. Heat required to warm air = \( \frac{VD}{53} \)

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>0.988</td>
<td>0</td>
<td>55.1</td>
<td>cc</td>
<td>cc</td>
</tr>
<tr>
<td>30</td>
<td>1.548</td>
<td>0.56</td>
<td>60.3</td>
<td>108</td>
<td>20.4</td>
</tr>
<tr>
<td>40</td>
<td>2.279</td>
<td>1.291</td>
<td>67.8</td>
<td>52.5</td>
<td>19.8</td>
</tr>
<tr>
<td>50</td>
<td>3.261</td>
<td>2.273</td>
<td>78.5</td>
<td>34.5</td>
<td>19.6</td>
</tr>
<tr>
<td>60</td>
<td>4.596</td>
<td>3.608</td>
<td>93.6</td>
<td>25.9</td>
<td>19.55</td>
</tr>
<tr>
<td>70</td>
<td>6.384</td>
<td>5.396</td>
<td>115.0</td>
<td>21.4</td>
<td>20.2</td>
</tr>
<tr>
<td>80</td>
<td>8.747</td>
<td>7.758</td>
<td>148.0</td>
<td>19.1</td>
<td>21.6</td>
</tr>
</tbody>
</table>
Heat required to evaporate one grain of moisture can be taken as .152 Btu. (.154 at 20° and .15 at 80°).

Therefore heat required to vaporize 43 grains per hour will be 6.55 Btu./hr.

Heat lost through the walls, roof and windows can be calculated from the following formula:

\[ H = .829D \]

where \( D \) = difference in temperature between inside and outside air.

\[ .829 = A_c = \text{loss of heat in Btu. per bird per hour per deg. F.} \]

Table 19
Total Heat Loss
1. Through full ventilation.
2. Through walls, roof, windows and door.

<table>
<thead>
<tr>
<th>Inside temperature (°F)</th>
<th>Loss through ventilation Btu./hr.</th>
<th>Loss through building Btu./hr.</th>
<th>Latent heat Btu./hr.</th>
<th>Heat for litter moisture Btu./hr.</th>
<th>Total Btu./hr.</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>0</td>
<td>0</td>
<td>1.86</td>
<td>6.55</td>
<td>8.41</td>
</tr>
<tr>
<td>30</td>
<td>20.4</td>
<td>8.29</td>
<td>2.66</td>
<td>6.55</td>
<td>37.2</td>
</tr>
<tr>
<td>40</td>
<td>19.8</td>
<td>16.6</td>
<td>3.8</td>
<td>6.55</td>
<td>46.35</td>
</tr>
<tr>
<td>50</td>
<td>19.6</td>
<td>24.9</td>
<td>5.44</td>
<td>6.55</td>
<td>56.49</td>
</tr>
<tr>
<td>60</td>
<td>19.55</td>
<td>33.2</td>
<td>7.76</td>
<td>6.55</td>
<td>67.06</td>
</tr>
<tr>
<td>70</td>
<td>20.2</td>
<td>41.5</td>
<td>11.0</td>
<td>6.55</td>
<td>79.25</td>
</tr>
<tr>
<td>80</td>
<td>21.6</td>
<td>49.8</td>
<td>15.7</td>
<td>6.55</td>
<td>93.65</td>
</tr>
</tbody>
</table>
AIR REQUIRED CUB. FT./HR. PER BIRD

INSIDE TEMP.

FIG. 19 TEMP. OF COMING AIR IS 30°F

AIR REQUIRED CUB. FT./HR. PER BIRD

OUTSIDE TEMP.

VOLUME OF AIR REQUIRED FOR VENTILATION

FIG. 20 INSIDE TEMP. IS 35°F
COMING AIR AT 20°F & 80% R.H.
MOISTURE REMOVED FROM LITTER = 43 GRAMS/HR.
A: LATENT HEAT
B: HEAT REQUIRED TO WARM AIR TO ROOM TEMP.
C: LOSS THROUGH BUILDING
E: HEAT TO VAPORIZE LITTER
MOISTURE, 43 GRAMS

TOTAL VAPOR TO BE REMOVED

HEAT B.T.U./HR.

WATER VAPOR EXHALED

WATER VAPOR (GRAMS/HR.)

INSIDE TEMPERATURE

FIG. 21 RELATION BETWEEN HEAT PRODUCTION & HEAT LOSS
Relation between Heat Production and Heat Loss
if Outside Temperature is Variable While
Inside Temperature is Retained at 35°F

During the cold days if we can keep the house at 35°F the
chicken can stand this condition without any harm pro-
viding that there is no draft of air.

A complete ventilation to remove all moisture is assumed.
At this condition a four pound leghorn hen will produce 40
Btu./hr., exhale 20 grains of water vapor per hour. Air at
35°F and 90% relative humidity has the capacity to hold
1.893 grains per cu.ft.

For equilibrium, the heat produced by the bird should
be equal to the heat lost due to ventilation, through the
construction, heat dissipated to vaporize water from the
bird’s body and heat required to evaporate moisture from
litter. Tables 20 and 21 still show that this condition
cannot be fulfilled if the outer temperature is lower than
22°F. This temperature is approximately the average temper-
ature in winter. The result is shown in Fig. 22.

For colder days either we have to restrict ventilation
or introduce another means of gaining heat. This new method
is the introduction of a heat exchanger.
Table 20

Air Required for Full Ventilation (Fig. 20)

House temperature is 35° F., water vapor to be removed is 63 grains per hour.

<table>
<thead>
<tr>
<th>Coming air temp °F.</th>
<th>Vapor in coming air grains per cu.ft.</th>
<th>Moisture removed by exhaust air grains/cu. ft.</th>
<th>Required air cu.ft./hr.</th>
<th>Heat to warm air to 35° Btu./hr.</th>
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<tbody>
<tr>
<td>-15 R.H. 90%</td>
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*Average R.H. for January, February, March and December is 83.5%, 82.5%, 80% and 82% (43).

Table 21

Total Heat Loss for the Same Conditions as in Table 20

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<thead>
<tr>
<th></th>
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</table>
Heat produced by bird

TEMP. OF HOUSE 35°F

A = LATENT HEAT
B = LOSS THRU VENTILATION
C = LOSS THRU BUILDING
E = HEAT TO VAPORIZE LITTER MOISTURE

4LB LEGHORN HEN ONLY
NO HEAT EXCHANGER & FULL VENTILATION

FIG. 22 RELATION BETWEEN HEAT PRODUCTION & HEAT LOSS
Introduction of Heat Exchanger

When we introduce the effect of the heat exchanger, a constant volume of air will be considered. Shier (34) recommended 60 cu.ft./hr. for each bird. This amount can be considered ample if we consider that the average temperature in winter is 20°F, Table 20.

Using a thermostat, the fan can work continuously if the inside temperature is over 35°F, while it will be intermittent or stop if the temperature drops below this range.

Tables 22 and 23 and Fig. 23 show that ventilation at this rate can give the required result if the outside air does not fall below 7°F.

Table 22

Total Sources of Heat

<table>
<thead>
<tr>
<th>Inside temperature 35°F</th>
<th>Ventilation 60 cu.ft./hr./bird</th>
<th>(Heat exchanger type No. 1)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Outside temperature of Btu./hr.</td>
<td>Hen Exchanger Total Btu./hr.</td>
</tr>
<tr>
<td>OF.</td>
<td>Btu./hr.</td>
<td>Btu./hr.</td>
</tr>
<tr>
<td></td>
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</tr>
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<td></td>
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</tr>
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</table>
Table 23
Total Heat Loss

Inside temperature 35°
Ventilation 60 cu.ft./bird/hr.
Ac = .829 Btu./(hr.)(deg. F.)

<table>
<thead>
<tr>
<th>Outside temperature (°F)</th>
<th>Latent heat (lbs. moisture)</th>
<th>Heat for litter gain (Btu/hr.)</th>
<th>Loss through ventilation (Btu/hr.)</th>
<th>Loss through construction (Btu/hr.)</th>
<th>Total Loss (Btu/hr.)</th>
</tr>
</thead>
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<td>4.2</td>
<td>19.7</td>
</tr>
</tbody>
</table>

If the type of fan discussed by Strahan (40) is not available in small units for poultry houses, a damper controlled by hand can be made at the inlet of the fan boxes of the heat exchanger. When the temperature of the outside drops below 10° the damper is lowered and is made to allow the flow of 4000 cu.ft./hr. (40 cu.ft./bird).

Tables 24, 25 and Fig. 24 show that we can keep the house at 35° while the outside temperature is 3° F. Below this temperature, ventilation will be intermittent and controlled by the thermostat.
**VOLUME OF AIR = 60 CU. FT. (Ht.) (BIRD)**

A: LATENT HEAT + HEAT REQUIRED TO VAPORIZE LITTER MOISTURE
B: LOSS THRU VENTILATION
C: LOSS THRU BUILDING
D: GAIN FROM HEAT EXCHANGER

**Fig. 23 Relation Between Heat Production & Heat Loss. (Heat Exchanger is introduced - Constant Volume of Air)**
Table 24
Total Sources of Heat
Inside temperature 35°F - Ventilation 40 cu.ft./hr./bird
(Heat exchanger type No. 1)

<table>
<thead>
<tr>
<th>Outside temperature °F.</th>
<th>Hen Btu./hr.</th>
<th>Exchanger Btu./hr.</th>
<th>Total Btu./hr.</th>
</tr>
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<tr>
<td>-15</td>
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<td>15.6</td>
<td>55.6</td>
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</table>

From 10° to 35° we use results in Table 22 - Ventilation 60 cu.ft./hr.

Table 25
Total Heat Loss
Inside temperature 35°F.
Ventilation 40 cu.ft./bird/hr.

<table>
<thead>
<tr>
<th>Outside temperature</th>
<th>Latent heat and heat for litter moisture</th>
<th>Loss through ventilation</th>
<th>Loss through construction</th>
<th>Total Btu./hr.</th>
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</thead>
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<td>-15</td>
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<td>20.7</td>
<td>49.4</td>
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</tbody>
</table>

From 10° to 35° we use results in Table 23.
VOLUME OF AIR = 60 CU FT/(HPI) (BIRD) IF OUTSIDE TEMP IS HIGHER THAN 10°.

A: LATENT HEAT + HEAT REQUIRED TO VAPOR-LITTER MOISTURE
B: LOSS THRU VENTILATION
C: LOSS THRU BUILDING
D: GAIN FROM HEAT EXCHANGER

Fig. 24 Relation between heat production & heat loss. (Heat exchanger is introduced - variable volume of air.)
Relation between Inside and Outside Temperatures

Considering a - Ventilation is 60 cu.ft. as far as the outside temperature is higher than 10° F.

b - Ventilation is 40 cu.ft. between condition "a" and until inside temperature is 35° (outside between 3° F. and 10° F.)

c - Ventilation stops if inside temperature is below 35° F.

If we fix the conditions to fit the three cases, Tables 26 and 27 will explain clearly the balance of heat production and heat loss.

In Table 27 the temperatures of walls and windows are shown to be always higher than the dew point for case "a." This means that there will be no condensation as far as we assume that only 43 grains per hour are to be removed from the litter.

For case "b" and "c" condensation is expected unless we assume from the statement of Clyde (11) that circulation of air in the house will make condensation occur when the surface temperature is lower than the dew point of the air by 3° to 8° F.
Table 28

Balance of Heat Production and Heat Loss

a - Ventilation 60 cu.ft. if outside temperature is above 10°
b - Ventilation 40 cu.ft. below condition "a" and inside temperature is higher than 35°
c - No ventilation if inside temperature is below 35°

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<td>6.55</td>
<td>21.1</td>
<td>28.9</td>
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</table>
Temperature of walls and windows

The temperature of the inside surface of walls and windows can be calculated if we assume the coefficient of heat transfer to each item and the surface coefficient.

If \( k \) is the surface coefficient for inside wall

\[
= 1.65 \text{ Btu.} / (\text{hr.})(\text{sq.ft.})(\text{deg. F.})
\]

\( U_1 \) is the total coefficient for wall

\[
= .062 \text{ Btu.} / (\text{hr.})(\text{sq.ft.})(\text{deg. F.})
\]

\( U_2 \) is the total coefficient for window

\[
= .45 \text{ Btu.} / (\text{hr.})(\text{sq.ft.})(\text{deg. F.})
\]

\( t_i \) = temperature of the inside air

\( t_o \) = temperature of the outside air

\( t_s \) = temperature of the inside surface

\[
. (t_i - t_s) \ k = U(t_i - t_o)
\]

For each 10\(^\circ\) difference between inside and outside temperature we will have

.375\(^\circ\) difference between inside temperature and inside wall surface temperature.

2.73\(^\circ\) difference between inside temperature and inside window surface temperature.
<table>
<thead>
<tr>
<th>Case No.</th>
<th>Outside temp. °F.</th>
<th>Inside temp. °F.</th>
<th>Wall temp. °F.</th>
<th>Window temp. °F.</th>
<th>Vapor moisture, grains/min.</th>
<th>Moisture to be removed, grains/hr.</th>
<th>Total moisture driven, grains/hr.</th>
<th>R.H.</th>
<th>Dew point °F.</th>
<th>Ventilation c.f.m.</th>
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<td>35</td>
<td>35</td>
<td>35</td>
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<td>1.433</td>
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<td>48.5</td>
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</table>

Table 27

Temperature of Walls and Windows and Dew Point for Conditions of Table 26
Fig. 25 Relation Between Outside and Inside Temperature
HEAT EXCHANGER IN DAIRY BARNs

The Cow as a Heat and Moisture Producer

A preliminary study for the introduction of a heat exchanger in dairy stables will be discussed.

The engineer should also know the heat and moisture production of the cow under the different environmental conditions.

Armsby (2) has done such a study, the result of which can be summarized as follows:

Table 28
Heat and Moisture Production of Cows

<table>
<thead>
<tr>
<th>Unit</th>
<th>Total heat Btu/hr</th>
<th>Sensible heat Btu/hr</th>
<th>Latent heat Btu/hr</th>
<th>Water vapor grains/hr</th>
</tr>
</thead>
<tbody>
<tr>
<td>900# Jersey producing 20# milk per day</td>
<td>2700</td>
<td>2025</td>
<td>675</td>
<td>4460</td>
</tr>
<tr>
<td>900# Jersey producing 30# milk per day</td>
<td>3025</td>
<td>2270</td>
<td>753</td>
<td>5000</td>
</tr>
<tr>
<td>1250# Holstein producing 30# milk per day</td>
<td>3295</td>
<td>2470</td>
<td>225</td>
<td>5450</td>
</tr>
<tr>
<td>1250# Holstein producing 45# milk per day</td>
<td>3705</td>
<td>2790</td>
<td>925</td>
<td>6100</td>
</tr>
</tbody>
</table>
The per cent of latent heat and the production of moisture should not be constant since they depend upon environmental conditions. Strahan (37) presented a curve having the per cent of latent heat under different stable temperatures. Assuming that the latent heat for one grain of moisture is .152 Btu, the amount of moisture can be calculated if the latent heat is known. In the coming discussion, Holstein cows are going to be considered in the stable. An average production of heat of 3500 Btu/hr will be taken. Latent heat at 50° F. will be 14% of the total and this will give 3200 grains/hr.

Volume of Air for Ventilation

King (24) recommended 3542 cu.ft. assuming that not over 3.3% of the air in stable should have been once breathed at any time. This will limit CO₂ to about 17 parts per 10,000. Armsby (2) recommends 3452 cu.ft. which checked King's result.

Strahan (37) found that this condition could not be fulfilled in theory or practice if control of temperature would be considered the practical criterion of good performance. Till 16° F. as outside temperature, volume of air should be less than 3500 cu.ft./hr.

In this study it will be shown that till 0° F., with
the introduction of a heat exchanger, the required ventilation of 3500 cu.ft./hr. can be performed.

Ventilation Through Heat Exchanger

A unit of heat exchanger of length 30 feet serving 10 cows will be discussed. Outside duct will be made square with 13 inch sides containing nine 3 inch pipes. It is calculated in the same way as type 2 and Fig. 26 is obtained.

The characteristics of the stable are taken from one solved (37) and housing 32 cows.

Table 29

<table>
<thead>
<tr>
<th>Unit</th>
<th>Area</th>
<th>U</th>
<th>Heat loss</th>
</tr>
</thead>
<tbody>
<tr>
<td>Side walls</td>
<td>1362</td>
<td>0.148</td>
<td>204</td>
</tr>
<tr>
<td>Windows</td>
<td>96</td>
<td>0.45</td>
<td>43.2</td>
</tr>
<tr>
<td>Doors</td>
<td>154</td>
<td>0.37</td>
<td>56.98</td>
</tr>
<tr>
<td>Ceiling</td>
<td>2808</td>
<td>0.125</td>
<td>351</td>
</tr>
<tr>
<td>Total</td>
<td>4440</td>
<td></td>
<td>655.18</td>
</tr>
</tbody>
</table>
HEAT EXCHANGER TYPE 2B
LENGTH 30'
35000 CUF./HR.
COUNTER FLOW

WARM AIR
COLD AIR
3' PIPE

CHANGE IN TEMPERATURE

10 20 30 40 50 60 70

t₂ - t₁ = HOUSE TEMP. - OUTSIDE TEMP.

FIG. 26 HEAT EXCHANGER TYPE 2B
\[ C = \frac{655.2}{4440} = .147 \text{ Btu.} / (\text{hr.})(\text{ft.}^2)(\text{deg. } F) \]
\[ A = \frac{4440}{32} = 138.8 \text{ ft.}^2 / \text{cow} \]

\[ AC = 20.5 \text{ Btu.} / (\text{hr.})(\text{deg. } F)(\text{cow}) \]

Relation Between Heat Gain and Heat Loss if Inside Temperature is Kept at 50° F.

Table 30
Heat Gained and Heat Lost (Inside Temp. 50°)

<table>
<thead>
<tr>
<th>Outside Temperature °F</th>
<th>Gain Btu./hr.</th>
<th>Loss Btu./hr.</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Exchanger:</td>
<td>Ventilation:</td>
</tr>
</tbody>
</table>

*At about zero degree Fahrenheit, there will be balance between heat production and heat loss if 50° F. is to be maintained at the stable.
Relation Between Inside and Outside Temperature

Table 31
Heat Gain and Heat Loss in Equilibrium State

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>-20</td>
<td>35</td>
<td>3500</td>
<td>1620</td>
<td>5120</td>
<td>355</td>
</tr>
<tr>
<td>-15</td>
<td>40</td>
<td>3500</td>
<td>1620</td>
<td>5120</td>
<td>355</td>
</tr>
<tr>
<td>-10</td>
<td>45</td>
<td>3500</td>
<td>1620</td>
<td>5120</td>
<td>355</td>
</tr>
<tr>
<td>-5</td>
<td>48</td>
<td>3500</td>
<td>1560</td>
<td>5060</td>
<td>470</td>
</tr>
<tr>
<td>0</td>
<td>52</td>
<td>3500</td>
<td>1530</td>
<td>5010</td>
<td>497</td>
</tr>
</tbody>
</table>

Above 0° F, the inside temperature will be higher than 50° and we either increase the volume of air in ventilation or open some of the windows if inside temperature exceeds 60° F.

Effect of condensation

Condensation as discussed before will increase the amount of heat gained from the exchanger. Experiments need to be made to evaluate this increase since frost will be formed if the outside air lies between -20° and 0° F.
In this range the temperature of the warm air at exit is below $32^\circ$.

### Table 32

Temperature of Cold and Warm Air at Exits

<table>
<thead>
<tr>
<th>Cold air</th>
<th>Warm air</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet</td>
<td>Exit</td>
</tr>
<tr>
<td>-20</td>
<td>5</td>
</tr>
<tr>
<td>-15</td>
<td>10</td>
</tr>
<tr>
<td>-10</td>
<td>12.8</td>
</tr>
<tr>
<td>-5</td>
<td>17</td>
</tr>
<tr>
<td>0</td>
<td>23.5</td>
</tr>
</tbody>
</table>
SUMMARY AND CONCLUSIONS

1. Ventilation has been rather generally considered to be important in the management of animal shelters, not only because of its effect upon the health and comfort of the animals but also because of the deterioration of the structure by excess moisture. The control of temperature, relative humidity and air movement and the removal of objectionable odors are considered to be the important factors to be controlled by ventilation.

2. Fan or mechanical ventilation is superior to natural or gravity systems because of the possibilities for positive control. It probably offers the additional advantage of lower cost.

3. The principal problem in ventilating an animal shelter successfully lies in the fact that animal heat alone is not produced in sufficient quantity to vaporize the moisture given off by the animal and to heat the circulating air necessary to remove it from the building. Supplementary artificial heat has been employed usually only in brooders for chicks or pigs.

4. Evidence of poor performance has been noticed in condensation on walls, ceiling and in litter or bedding.
5. Our problem, therefore, is the shortage of heat. Heat from soil underneath the floor, or heat gain by the use of a heat exchanger are the two methods I suggested in this study.

6. In an animal shelter if the temperature of the inside is lower than that of the subsoil, there will be gain of heat. This occurs usually in poultry houses where we try to keep the temperature during cold days at 35°F. In a barn kept at 50°F, there may be loss of heat since the average temperature of soil at 6 feet is expected to be less than 50°F (under bare soil average temperature at 6 feet in Iowa is 44.6°F).

7. Existence of an insulated structure over the soil will make the fluctuation of temperature below the floor not to be sharp and so the flow of heat from or to the soil will be gradual and uniform.

8. In a poultry house kept at 35°F, the gain of heat is estimated to be about 1.2 Btu./(hr.)(ft.²). This amount will raise the temperature of a house whose AC is equal to 0.83 and has ventilation of 60 cu.ft./hr./bird, about two degrees Fahrenheit.

9. To prevent heat from dissipating through the floor to the cold foundation walls, the floor should preferably be separated at the line of contact with the foundation. The gap should be filled with an insulating material.
10. To prevent soil from being cold near the foundation wall beneath the floor, an insulating layer can be added to the thickness of the foundation. This will help in keeping the floor warm at this section and prevent condensation.

11. In the ordinary ventilation systems, restricted ventilation ought to be followed in cold days so that the house temperature will not fall below the critical temperature.

12. The recommended volume of air change is 60 cu.ft./hr. per hen (34) and 3542 cu.ft./hr. per cow (24) to obtain a good standard of air purity.

13. In a poultry house with AC = 0.83 the temperature of 35°F in the house cannot be maintained under the ordinary ventilation system with the recommended volume of air for ventilation if outside temperature is below 22°F.

In a barn with AC = 20.5, the inside temperature condition of 50°F cannot be held if outside temperature is below 15°F. Below these limits ventilation will be restricted.

14. The introduction of a heat exchanger is a promising development in the ventilation of animal shelters. In the same poultry house with a type No. 1 heat exchanger
and the recommended volume of air, required minimum limit of inside temperature can be maintained even if the outside temperature reaches 70° F. and in the same previous barn with units of heat exchanger of type No. 2B, the inside temperature can be maintained at 50° F. level or higher as far as the outside temperature reaches zero.

15. Overall coefficient of heat transfer is the resultant of the two coefficients of the inside and outside walls of the pipes. One of these coefficients will increase due to condensation of vapor. The second can be increased by raising the velocity of flow of cold air in the space between inside pipes and outside duct and by increasing the turbulence of flow by the introduction of supports between the pipes. The latter coefficient will be the limiting factor for the overall coefficient.

16. The use of small diameter pipes for the flow of warm air will be in the benefit of gain of heat due to the increase of heat transfer surface area. The limiting factor for the diameter is the increase of friction to air flow which will increase the cost of operation and the formation of frost which may log the pipes.

17. To obtain maximum use of this system, the house must be well insulated and air tight. Any leakage of air
will cause the drop of temperature inside and difficulties in controlling the system. Frost will be formed around these leaky points while it is expected that neither condensation nor frost will be formed on the walls or windows.

18. When exchange of heat is made between the warm exhaust air and cold fresh air, it will be possible to keep the shelter warmer, or, circulate more air so that we can keep the humidity low, or rather we can make a poorly insulated wall and still have no condensation.

19. Flow of cold air over the fan motor will raise the air temperature few degrees.

20. In such a system, means for draining the condensate from inside the pipes must be provided. Also other precautions must be made not to let the condensate formed over the outside pipe fall on the animals or floor.

21. Series of experiments on different types of heat exchangers need to be made to evaluate the exact gain due to condensation and frost formation in the pipes where warm air passes.
LITERATURE CITED


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