

DESIGN AND PERFORMANCE OF FAN
VENTILATION IN POULTRY LAYING HOUSES

Thesis for the Degree of M. S.
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Jacques Albert Choinière
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This is to certify that the

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Walter M. Carleton
Major professor

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FAN VENTILATION IN POULTRY
LAYING HOUSES

by

Jacques Albert Choinière

An Abstract

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ABSTRACT

The development of a solution to the satisfactory design of poultry house electric fan ventilation systems has plagued operators, poultry specialists and agricultural engineers for many years. In fact, there are still great differences of opinion concerning the most basic assumptions and principles to use in design.

This study was made in three different parts. First, an extended review was made of the literature published on the heat and moisture production of laying hens and the many other basic factors involved in the design of poultry house insulation and ventilation.

Second, charts and nomographs presented previously to simplify the design of poultry house insulation and ventilation were analyzed and their limitations pointed out. Composite charts and nomographs with new characteristics were developed. Slide rules were also prepared which are believed to be very convenient for use and at the same time more accurate than nomographs and composite charts.

Third, actual performance of fan ventilation systems in two poultry houses was studied and analyzed in order to compare results with those obtained in calorimeter tests. From the data collected, the trend in the total sensible heat available per hen was determined together with the trend in the net sensible heat coming from secondary sources (heat from litter decomposition, sun, lights, etc.). These secondary sources of heat were found to be equal to 22.43 Btu per hour, per hen at 50°F. This value agrees very well with the findings of Stapleton and Cox. Rates of ventilation of approximately four cubic feet per minute, per hen were found desirable for moisture removal in both well insulated and moderately insulated poultry houses. It was also found that the three most important factors in the successful performance of fan ventilation and the maintenance

of dry litter are a reasonably insulated house, the use of deep litter and an adequate design of the air intake system. With regard to the latter factor it should be remembered that the air exhausted by the fans has done its job by the time it reaches the exhaust point and the incoming air is the one which must be primarily considered if the system is to do the job expected of it.

Although much work was done on the design and performance of poultry house ventilation, this problem is far from being solved. Because of the lack of adequate and complete data on the heat and moisture production of birds more tests need to be done in both ten-hen and single-hen calorimeters. More work needs to be done also in smaller and better controlled poultry houses than the ones studied here. Such tests could concern the actual heat production and moisture removal under housing conditions, the necessity of ducts, the design of air intakes, the ventilation rates now available, the relative importance of the factors producing dry litters, etc.

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VITA

Jacques-A. Choindi re was born on February 7th, 1925, in Granby, Province of Quebec, Canada. He attended St. Stanislas High School in Montreal, from 1940 to 1943 and at the same time took two winters of night courses in industrial drawing. Next he spent four years studying agriculture at Montreal University and in 1947 got his B.S. degree in agriculture with a major in agricultural engineering. During this four year course, he occupied his summer vacations working for a dairy farm, an horticultural farm, a rural electrification company (Quebec Power Company, Quebec) and a dairy machinery company (De Laval Company, Montreal). He also worked part of this time in the construction field.

After his graduation from Montreal University, he did one year of field crop and statistical study at Laval University. After this, he was farm editor for the French monthly magazine "La Ferme", located in Montreal.

Being interested in farm structures, in 1949 he applied to the Farm Structure Division, Quebec Department of Agriculture. After nine months of practical experience in the drawing of plans and visiting farms he asked for a leave of absence in order to specialize in his field.

He chose Michigan State College because of the proximity and the greater similarity of climate as compared to most other states. Also because of the recognized authority of Michigan State Agricultural Engineering Department in the field of farm structures. After two years study at Michigan State College he was awarded the B.S. in Agricultural Engineering at the end of the winter term 1952. After that he started work toward the master degree. Each summer he returned to his Canadian position of employment in order to gain more practical knowledge. In conversations with extension agriculturists, agricultural engineers, and farmers, he found that one of

the most controversial subjects was that of poultry house fan ventilation during the winter. For this reason the author became interested in the poultry house ventilation problem and made it the subject of study.

The author is a member of the Quebec Corporation of Agriculturists, the Corporation of Professional Engineers of Quebec and the Agricultural Institute of Canada. He also is a student member of the Engineering Institute of Canada, and an associate member of the American Society of Agricultural Engineers.

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PURPOSES OF THE STUDY

The purposes of this study were as follow:

1-To review all the literature published on poultry house fan ventilation, in order to critically analyze each factor contributing to the success of its use, especially the maintenance of dry litters.

2-To point out the discrepancies in basic assumptions and show the wide difference of opinion existing regarding some of the most important factors affecting ventilation design.

3-To present some nomographs, composite charts and slide rules to solve the moisture removal and temperature control formulas without introducing any of the limitations contained in the different types of charts presented previous to this date.

4-To conduct performance tests on actual poultry houses in order to obtain a better understanding of the theoretical analysis and, if possible, to bring some light on the ventilation problem.

5-To indicate the need for further research on the many aspects of the poultry house ventilation problem.

I. INTRODUCTION

A. Reasons Why Successful Poultry House Ventilation is Difficult to Obtain

The design of poultry house ventilation is more difficult than that of dairy barns because of the three following reasons:

1-Excess headroom, cubage and wall exposure as compared to dairy barns;

2-Relatively small amount of heat produced by the birds;

3-Relatively large amount of moisture produced by the birds.

1. Excess headroom, cubage and wall exposure as compared to dairy barns.

The problem of heat loss in poultry houses, as compared to dairy barns, is greatly aggravated because poultry houses are constructed for convenience of the worker rather than actual requirements of the poultry.

Hill (43) illustrates the ridiculousness of the situation by considering that if human dwellings were built along similar lines ceilings would be from forty to fifty feet in height.

Kable (55) mentions that since heat tends to bank at the ceiling, the hen must heat a column of air eight to ten times her height before the benefits at floor level from a rise in temperature.

Strahan (100) calculated the ratio of space requirements to be 4.8 cubic feet per pound of bird, live weight, as compared to 1.7 cubic feet per pound of cow, live weight.

It is seen therefore, that the volume of air to be heated and the square feet of radiation surface in walls and roof are greater per animal unit than for any other type of livestock shelter. It should be noticed that the square feet of wall and roof surface per hen also increases as

the size of house diminishes, so small houses are harder to heat and to ventilate.

Two-story poultry houses (or multi-story poultry houses) are an efficient way of reducing both the construction cost and the square feet of radiation or exposed surface per bird. In one story houses, some gain can be made in house warmth and reduction of material cost by reducing front and rear walls heights.

2. Relatively small amount of heat produced.

In designing poultry house ventilation and housing we must also consider the fact that the hen is a relatively small heat-production unit.

Ashby, Miller et al (4) illustrate this by saying that twenty-five hens occupy more floor space than one dairy cow, but emit only one-third as much heat. More detailed study of the heat and moisture production of birds will be presented in a later section.

3. Relatively high amount of moisture produced.

The third factor to consider is the quantity of moisture produced. One cow produces about fifteen to twenty pounds of expired moisture per day. The moisture in the excrement is not too much of a problem since in standard dairy barns the excrement is removed from the barn every day. On the other hand, twenty-five hens will produce approximately eleven pounds of total moisture per day (the moisture eliminated in the excrement is added here since in most cases the litter is allowed to accumulate). This is less than for an equal floor area in a dairy barn, but the fact that only one-third as much heat is produced by the birds makes the problem of moisture removal more difficult in poultry houses than in dairy barns.

B. Relations Between Ventilation and Insulation

Poor insulation or lack of insulation in the poultry house may make almost useless what otherwise would be an excellent ventilation system.

The main value of insulation lies in the important protection it gives against sudden temperature changes, thereby, tending to maintain a constant temperature. The result is a greater moisture holding capacity of the air inside the poultry house and a lower relative humidity value, both factors making for a minimum of condensation on the walls and ceiling, and the maintenance of a drier litter.

There are rather wide differences of opinion as to how much insulation would prove economical in a laying house. Recent arguments seem to favor houses where temperatures are above freezing except for very short periods. It is probable that preventing a drop in egg production, plus saving in feed and labor, will combine to make the added investment in insulating materials a profitable one, particularly in the case of the large enterprise.

Otis and White (20) concluded from their tests on a commercial poultry house: "it is significant that this uniformity of egg production was done without heat other than that produced by the birds. Temperature control by means of abundant insulation was probably responsible to a greater extent than ventilation for satisfactory humidity conditions." These results and many other similar ones help to substantiate the contention that the problem of poultry house ventilation becomes simple when satisfactory temperatures are maintained by adequate insulation.

Recommended insulating values to provide the greatest possible degree of control over inside temperature and relative humidity in cold weather consistent with good building investment economy will be presented in a later section.

2. Functional Requirements of a Poultry House Ventilation System*

A poultry house ventilation system must meet at least eight functional requirements. These functional requirements are as follow:

1. Prevention of excess condensation on walls and ceilings.

A dry house is desirable because it results in an increase in the useful life of the building itself and the poultry equipment inside it.

No detailed set of rules can be given at this stage concerning the maximum relative humidity to be maintained in a poultry house if condensation on walls and ceilings must be prevented. However, it can be said that maintenance of a relative humidity below 60% generally is satisfactory in ordinary winter weather (h).

2. Maintenance of dry litter.

One of the greatest problems the poultryman has is keeping the litter dry. If he is successful in keeping it dry, he can eliminate the problem of changing the litter, reduce the number of dirty eggs, and provide a more sanitary condition for the birds. A general recommendation is to limit the moisture content of litter to below 40%, wet basis (h).

According to Oliver (78), "maintaining a low relative humidity in the house prevents the absorption of moisture from the air by the litter." He adds: "Most litter material will remain relatively dry where the relative humidity of the air next to the litter is 80% or less, and will still be quite serviceable where 90% relative humidity prevails part of the time."

3. Maintenance of reasonable indoor temperature.

The removal of excess moisture must be done without lowering the temperature sufficiently to interfere with egg production or the health of the birds,

* Ventilation system: Will be used throughout this text although some authors are using both "ventilation system" and "ventilating system" (11, page 175).

without unduly increasing feed consumption for maintaining body heat, and without making the house uncomfortable, or cold enough to freeze water pipes.

According to Ashby, Miller et al (4) in ordinary weather, the temperature of the poultry house should always be maintained above 32°F, but in extreme weather, the house can be allowed to go as low as 15°F. The writer believes that such low temperatures are undesirable and the houses should be constructed to maintain a temperature as much as possible above 32°F, in ordinary weather, and as close as possible to 32°F, in extremely cold weather.

In general, desirable indoor temperatures will be maintained only if the house is adequately insulated and if the ventilation system provides some means of partial or complete restriction during extremely cold weather.

4. Maintenance of air purity and removal of objectionable odors.

Early standards of ventilation attached great importance to carbon dioxide concentrations. However, it has since been proven (11) that birds are greatly tolerant in this respect and that no injury results from the higher concentrations of carbon dioxide produced by temporarily restricted ventilation. Therefore, it is assumed that in removing the excess moisture objectionable odors will also be removed and an adequate supply of fresh air provided.

5. Reduction in labor requirements.

Another functional requirement of a ventilation system should be to reduce the labor requirement to a minimum by eliminating: a) the necessity of opening and closing window curtains, dampers or slots one or more times a day; b) frequent litter changing or addition; c) the necessity of cleaning too many dirty eggs.

6. Avoidance of draft and excessive air change.

A draft may be defined as a stream of air of relatively higher velocity and of lower temperature than the surroundings.

One of the basic requirements of any ventilation system is to provide uniform air movement through the house without drafts, especially in the roosting area. The reason is that high air velocity accelerates the rate of heat transfer from the body by convection; if it is uneven or localized as in a "draft", the result appears to be a derangement of the homeothermic mechanism (11) with a subsequent decrease in resistance to infections such as cold and pneumonia. These infections are always accompanied by a drop in egg production and therefore draft should be avoided by proper spacing, location and size of intakes as will be discussed in a later section.

7. Reduction of the floor space requirement per bird.

General floor space recommendations are roughly three square feet for small breeds and four square feet for large breeds. However, Barre and Sammet (11) state that although age and breed are the principal factors in determining necessary floor space, the space requirement is less per bird in well insulated houses and in large pens than in poorly insulated houses and in small pens. He presents the information in Table 1 as a summary of general practice.

Table 1. Floor space required for laying hens *
(A summary of general practice)

Breed size	Number of hen per hen	Floor space, per hen, ft ²
Small breeds (White Leghorn)	25	4
	100	3.5
	200	3
	400	2.75
Large breeds (Plymouth Rock & Rhode Island Red)	25	4.5
	100	4
	200	3.5
	400	3.25

* From data by Ashby, Miller et al (4)

Lampman (65) reports that the use of good ventilation maintains drier houses and makes it possible to reduce the floor space from four square feet to as little as $2 \frac{1}{4}$ to $2 \frac{1}{2}$ square feet per bird. The production of the extra number of hens in a well insulated and ventilated pen is believed by many farmers to pay quickly for the cost of the insulation and the ventilating equipment. Overcrowding should however be avoided as an aid to disease control and as a means of reducing cannibalism.

8. Maintenance of more comfortable housing conditions during the hot summer months.

The ventilation system should also help in maintaining more comfortable housing conditions during the hot summer months.

Baldwell (15) states that the use of 4 cfm* per hen relieved bird during hot spells and reduced the loss of many of them because of the heat. He adds that "the total cost of operating these fans per month did not cost as much as two birds that would die because of the tremendous heat." Before installation of the fan, as many as fifteen birds had been lost in one night when the temperature became unbearable. Kable (55) also recommends the summer use of the fan on still, hot days and says that ventilation by windows should be used at other times. Finally Oliver (78) reports that during summer months "when the litter is quite dry"... a ventilation system helps to remove the dust from the air, again helping to minimize the disease problem."

D. Advantages and Disadvantages of Natural Gravity Systems

There was a time where natural gravity flue systems had one big advantage over forced systems, that of being cheaper.

*cfm: Will be used throughout this text as an abbreviation for cubic feet per minute.

With the actual high price of lumber, this advantage does not hold anymore and Strahan and Marsh (97) mention that forced ventilation reduces the first cost by more than half.

It is true that natural gravity flue systems were successful. But however, successful they have been, they have always been subject to a number of serious objections. Strahan and Marsh (97) point out three of the major objections:

- 1-Large awkward flues must be installed.
- 2-They are expensive, whether they be manufactured or home-built.
- 3-The season during which they render maximum service is limited by climatic variations.

Richardson and Huber (85) speaking of natural gravity systems in general mention that the greatest difficulty is in avoiding moisture and frost on floors, walls and ceilings, since "when the weather is extremely cold and the wind blows hard they 'pull' the hardest - just when they shouldn't." In other words they have the great disadvantage of depending upon a number of factors over which no control can be exercised - such as the outside temperature, the direction and velocity of the wind and the amount of heat given off by the poultry.

Lampman (65) cites a very good example of the greater labor requirement of natural ventilation as compared to forced ventilation: "The labor saving feature with the modernized improvement has been very striking. In our case at the Experiment Station, with the many small pens involved, it takes only a few minutes a week to do the necessary checking of the pens that have thermostatically controlled flue ventilation. In the same pens it was estimated that in the past with the old system it required at least twenty minutes daily, or ten hours per months, to make the frequent adjustments of the numerous windows and intakes involved. The litter will be

changed only once a year with this system as compared with cleaning at least every two weeks during winter under the old regime."

E. Advantages of Forced Ventilation

Forced ventilation has many advantages over natural ventilation. These may be summarized as follow:

1-It provides fresh dry air, removes moisture and ammonia gases, and maintains a rather constant inside temperature (when a thermostat is used), therefore, making for better producing and working conditions.

2-It helps to keep the litter dry and loose, resulting in a decrease of the labor required to remove litter and to clean eggs, and resulting also in a saving of litter material.

3-It prevents drafts and excessive air change due to wind or sudden variations in outside temperature, therefore providing healthier conditions.

4-It reduces the formation of moisture on the house walls and ceilings, with the result of a longer useful life of the building.

5-It is automatic, therefore eliminating the labor otherwise required to regulate curtains, windows, dampers and slots.

6-Flues are entirely eliminated or at least very much reduced in size and simplified in construction.

7-The fan can be adjusted to meet any window opening.

8-It may be used with profit to make the house more comfortable during the hot summer months, resulting in a decrease in mortality and an increase in summer egg production.

9-It is more effective, cheaper and more reliable than most natural gravity systems.

F. Actual Trends Toward Mechanical Ventilation of Poultry Houses

In the last few years there has been a rapid increase in the number of electrically operated ventilation systems. This trend comes from a need for a more positive and effective operation, that is, a system fulfilling to the highest degree possible the functional requirements of a ventilation system as described previously.

Because of its advantages, electric fan ventilation, although at first limited to large poultry houses, is now applied in houses of any size and fan manufacturers are providing fans in all sizes even as low as 120 cfm.*

Stapleton (94) comparing natural gravity flue ventilation to fan ventilation states: "It does not profane the attainments of Rutherford and King to say that, had exhaust fans and rural electric service been available when this dairy stable ventilation problem needed solution, they would not have looked twice at a hay mow flue. They took what was at hand and made something of it that worked, when properly installed. But if we farm building ventilation people are going to keep on worshipping the glories of Rutherford and King, it will be a little while before there will be any progress in farm ventilation. Rather shouldn't we use the same ingenuity as Rutherford and King, take the results of fifty years of progress and adopt them to this ventilation problem? I think so."

G. Ventilation and Air Conditioning Defined

Ventilation is the process of removing air from and supplying fresh air to any enclosed space by mechanical or natural means. Air conditioning, on the other hand, is the control by natural or mechanical means of

*cfm: Will be used throughout this text as an abbreviation for cubic feet per minute.

1-The composition (or purity)

2-The temperature

3-The humidity

4-The movement

of the air within an enclosed space, to produce and maintain environmental conditions which will produce desired effects upon the occupants of the room or upon materials stored or handled in it (6) (32).

According to Fairbanks (32), mechanical ventilation with its automatic control is doing more than only ventilating, it is also providing real air conditioning of animal shelters: "When A.M. Goodman and I started our work in dairy stable ventilation (that was what we called it and still do although the term is not correct), I worked at first with two thoughts in mind: air change and moisture removal. This was not satisfactory for the dairy stable. It gradually became evident that since I was not satisfied, I had better try to find what to do about it. We have found out some things and I do not hesitate today to say that we are air conditioning animal shelters. You do not always have to have a mechanical heating system or a mechanical refrigerating system to air condition an animal shelter. You can do it with animal heat, insulation, and a properly designed ventilation system. We now control the composition (or purity), temperature, humidity, and motion of the air going into animal shelters:

1-Composition or purity control: This is controlled by size of outtakes * and intakes, location and design of outtakes and intakes, and wall construction.

2-Temperature control: This is accomplished through design of outtakes and intakes, wall and ceiling construction, and animal heat.

* Outtake : Will be used throughout this text in place of exhaust, even though exhaust may be a most common term for many engineers.

3-Humidity control: This is influenced by the design of ventilation system, the wall and ceiling construction, and the animal heat and moisture production.

4-Air motion control: This is controlled by the design of the ventilation system, the wall and ceiling construction, and the animal heat distribution."

It should be noted that although there are some cases where mechanical heat is necessary to obtain the desired results, for example in brooder houses, in most other types of animal shelters artificial heat is not required.

Fairbanks (32) concludes by saying: "Let's put ventilation in the background and still use it as one of the tools with which we work. Let's not give so much publicity to ventilation of farm buildings. But let's put the emphasis on air conditioning. Let's explain clearly what air conditioning is, and we will then have more than we can do in this field."

The writer's opinion is that Mr. Fairbanks is completely right when he states that with mechanical ventilation, thermostat control, proper size and distribution of air intakes, proper wall and ceiling insulation and adequate animal population we are doing a real job of air conditioning. However, the writer believes we should still continue to use the name "ventilation" instead of the name "air conditioning", because in the mind of to many people the name "air conditioning" would take the meaning of something rather expensive like a "residence air conditioning unit."

II. DESIGN OF FAN VENTILATION IN POULTRY LAYING HOUSES

The purpose of this second part is:

- 1-To review all the literature published on the design of poultry house fan ventilation, in order to critically analyze each factor contributing to the success of its use, especially the maintenance of dry litters.
- 2-To stress the discrepancies in basic assumptions and the completely different opinions existing regarding some of the most important factors.
- 3-To show the limitations of the different types of poultry house ventilation charts published to date.
- 4-To present nomographs, composite charts and slide rules for the simplification of poultry house ventilation design and analysis.

Because of the many factors involved, the writer found it preferable to include the review of literature under the study of each factor instead of grouping it confusedly in a separate section.

The factors to be considered are:

- A. Physiological requirements of hens.
- B. Heat and moisture production of hens.
- C. Secondary sources of heat and moisture in poultry houses.
- D. Design outside weather conditions.
- E. Methods of standardization of poultry house ventilation.
- F. Poultry house ventilation formulas.
- G. Poultry house ventilation charts.
- H. Poultry house ventilation nomographs.
- I. Poultry house ventilation composite charts.
- J. Poultry house ventilation slide rules.

- K. The problem of temperature control.
- L. The problem of condensation on walls and ceiling surfaces.
- M. The problem of condensation within walls and ceilings.
- N. The problem of condensation on the interior surface of cold attic walls.
- O. The wet litter problem.
- P. Design of the intake system.
- Q. Design of the outtake system.

A. Physiological Requirements of Hens

1. Environmental temperature

a. Low critical temperature for hens. The so-called "low critical temperature" is defined as the environmental temperature at and below which the heat production of a warm blooded animal will increase to prevent a lowering of body temperature.

For laying hens on full feed and normally active, Lippincott and Card (66) give the critical temperature as being approximately 15°F. They also mention that the critical temperature of the same hens while on the roost at night is about 40°F, since the "activity increment" of heat production is not available. (For fasting birds at rest it would be about 62°F).

Discussing the economic aspect of the low critical temperature, the same authors (66) make the following statement: "Since the animal is both producing heat and losing heat to its surroundings, it is essential that there be some sort of adjustment between these two processes in order to maintain a constant body temperature. When the environmental temperature is above the critical point the heat radiating capacity of the body is varied by the so-called physical regulation; when it is below the critical point heat production is varied by the so-called chemical regulation to meet the needs of the body. The energy required for the increased heat production under such conditions can come only from the feed. Hence in the interest

both of economy and sustained egg production it is desirable to keep the temperature of laying houses above the critical point."

b. Optimum poultry house temperature. The optimum poultry house temperature is defined as the temperature or temperature range within which egg production is not affected adversely, and poultry house management is not complicated by any special problem such as freezing of water system, etc.

Summarizing the work of many researchers, Barre and Sammet (10) recommend the range of 30 to 70°F as being the desirable working range of indoor temperature. They say: "The optimum temperature for poultry production is not known. There is experimental evidence to indicate that production is not affected adversely at temperature as low as 10°F. At the lower temperature, however, there is some danger that combs and wattles may freeze. Temperatures below freezing (32°F) are a considerable disadvantage with respect to labor efficiency and the freezing of water systems. A desirable working range is assumed to be 30 to 70°F."

According to the Poultry Committee of the Rutgers University Farm Buildings Institute (93), "The laying house should be kept above freezing in the wintertime. The house should be planned, divided, and built so as to avoid sudden and severe changes in the house temperature. It is believed that hens generally produce best when the house temperature is between 50 and 60 degrees."

Ashby, Miller et al (4) state: "Hens are kept successfully under widely different climatic conditions but are affected adversely by extremes or sudden severe changes of temperature and by cold drafts. They lay best in spring when the temperature is about 50° to 60°F outdoors. Egg production is materially reduced by low or high temperatures or by extreme, sudden changes." Moyer and Blickle (68) also make a similar statement.

Shier (92) states: "Hiese and McCormick have indicated that at higher temperatures hens need considerably less food for bodily heat production and the savings in feed may justify maintenance of higher temperatures. Other considerations such as trouble with frozen water, feed, litter, and eggs, justify temperatures above freezing." Therefore, Shier adds: "It is suggested that the temperature be kept as near constant as possible in the forties or fifties. During severe cold, the temperature in the house will probably go into the thirties, but if the house is well insulated, temperature changes will be slow and the birds will have time to adjust themselves to the changing temperature gradually."

Ota, Garver and Ashby (79) found the amount of feed consumed to increase linearly at the rate of one pound per one hundred hens for each five degree drop in the environmental temperature (Table 2):

Table 2. Trend in feed consumption of 5-lb laying hens as affected by environmental temperature *

Environmental temperature (°F)	35°	40°	45°	50°	55°	60°	65°	70°	75°	80°	85°
Feed consumption (lb/day, 100 hens)	35	34	33	32	31	30	29	28	27	26	25

* From data by Ota, Garver and Ashby (79)

The increase in feed consumption for an environmental temperature of 35°F is therefore two and three pounds per day per one hundred hens as compared to environmental temperatures of 45 and 50°F. Assuming that the relation remains linear down to an environmental temperature of 0°F, the increase in feed consumption requirements would be as given in Table 3.

Table 3. Trend in feed consumption increase of 5-lb laying hens as affected by environmental temperature

Environmental temperature (°F)	0°	5°	10°	15°	20°	25°	30°	35°	40°	45°	50°
	*	*	*	*	*	*	*	*	*	*	*
Increase in feed consumption as compared to an environmental temperature of (lb/day, 100 hens)	35°	7	6	5	4	3	2	1	-	-	-
	40°	8	7	6	5	4	3	2	1	-	-
	45°	9	8	7	6	5	4	3	2	1	-
	50°	10	9	8	7	6	5	4	3	2	1

* * From data by Ota, Garver and Ashby (79)

* Extrapolated from the assumption that the increase in feed consumption remains linear for the range of 0 to 45°F as it was found for the range 35 to 85°F.

Expressed in percentage of the feeding cost at 35°, 40°, 45° and 50°F, the above values are changed to those presented in Table 4.

Table 4. Trend in feed consumption increase of 5-lb laying hens as affected by environmental temperature

Environmental temperature (°F)	0°	5°	10°	15°	20°	25°	30°	35°	40°	45°
	*	*	*	*	*	*	*	*	*	*
Percentage of increases in feed consumption as compared to an environmental temperature of	35°	20.0	17.1	14.3	11.4	8.6	5.7	2.8	-	-
	40°	26.5	23.5	17.6	14.7	11.8	8.8	5.9	2.9	-
	45°	27.2	24.2	21.2	18.2	15.1	12.1	9.1	6.0	3.0
	50°	31.2	28.1	25.0	18.7	15.6	12.5	9.4	6.2	3.1

* * From data by Ota, Garver and Ashby (79)

* Extrapolated from the assumption that the increase in feed consumption remains linear for the range of 0 to 45°F as it was found for the range 35 to 85°F.

Last year, Becker (13) found the increase in egg production for an uninsulated pen to be approximately 10% over that of an insulated pen. The temperature in the uninsulated pen was allowed to fluctuate parallel with the outside temperature and to reach as low as 0°F. The birds had the combs and wattles cut to avoid frosting of these fleshy, unprotected parts. The amount of feed consumed in each pen was not measured, making it impossible to check if the 10% increase in egg production was economically significant. Also no consideration was given of the difference in the labor requirements for egg cleaning or similar work. It should be added that the uninsulated pen had a southern exposure while the insulated pen had a northern exposure. Becker intends to repeat his experiment next year (1953-1954) and to consider the feed consumption and perhaps the labor requirement. If results of his next year's experiment do not give so high a difference in the egg production, it will probably mean that the birds were not divided properly between the two houses (Birds should be divided into two groups having similar average weight and egg production, instead of being divided at random), or else that some factors like the difference in the orientation of each pen accounted for some part in the difference obtained.

Ota, Garver and Ashby (79) calorimeter tests do not agree with Becker results. According to these authors, the highest egg production occurs at an environmental temperature between 50 and 60°F. The following table (Table 5) shows the trend they found in egg production as affected by environmental temperature.

Table 5. Trend in egg production of 5-lb laying hens as affected by environmental temperature *

Environmental temperature	35°	40°	45°	50°	55°	60°	65°	70°	75°	80°	85°	90°
% of egg production	62	69	74	77	78	77	75	73	67	62	57	50
				Max								

* From data by Ota, Garver and Ashby (79)

The Insulation Board Institute (49) citing some results obtained at Iowa State College state that hens in insulated houses laid sixteen more eggs per bird and required one pound of feed less to produce each dozen of eggs.

From the above review of literature, it is the writer's opinion that the increase in feed consumption as a result of low temperature probably offset any profit from a possible increase in egg production as a result of having poultry houses subject to very wide fluctuations of temperature. Moreover, if lower environmental temperatures really results in lower egg production as found by Ota, Garver and Ashby (79) and Iowa State College (49), the proper insulation of laying houses more than pays for itself, and should be highly recommended because of the additional benefit it may bring to the poultrymen.

c. Design poultry house temperature. The design poultry house temperature is defined as the minimum temperature below which the house will not be permitted to fall in cold weather.

Ashby, Miller et al (4) recommend to maintain the indoor temperature above the following minimum (Table 6), in houses for commercial egg production with ordinary feeding.

Table 6. Recommended design temperatures for poultry laying houses * *

Temperature zone *	Ordinary (°F)	Extreme (°F)
Zone 1	32	15
Zone 2	40	20
Zone 3	45	25
Zone 4	50	32

* A map of these temperature zones will be presented in section D dealing with design outside weather conditions.

* * From Ashby, Miller et al (4)

Strahan (100) recognizes that design poultry house temperature is a management decision, and involves a question upon which opinions vary. He recommends 35°F (a temperature just a little above freezing) as being probably the most economical temperature. It is also the opinion of the writer that houses should be constructed to maintain a temperature as much as possible above 32°F, in ordinary weather, and as close as possible to 32°F, in extremely cold weather.

The insulation values to meet these requirements will be discussed in more details in section K dealing with the problem of temperature control.

2. Environmental relative humidity

a. Optimum poultry house relative humidity. The optimum poultry house relative humidity is defined as the relative humidity or relative humidity range within which production is not affected adversely, and poultry house management is not complicated by any special problem such as excessive condensation, excessive dust, etc.

As a conclusion of his seven years of research in eight poultry houses, Smith (93) reported that "no correlation was found between humidity and winter egg production."

Gutteridge et al (39) also reported in the same way: "Restriction of ventilation to a minimum, all ventilators being completely closed with a resulting temperature of 45.6°F and very high humidity and carbon dioxide content of the air had no detrimental effect upon egg production."

Hutter et al (17) reporting on their tests state that "Relative humidity, ranging from 40%, but never remaining above 90%, longer than 24 hours, seemed to have no effect on pen conditions or on the activity of the birds throughout this test."

In their text book, Poultry Production, Lippincott and Card (66) do not agree with the above reports. They say: "There is no condition under which poultry is kept, unless it is a state of starvation, that is more surely and quickly fatal to profitable production than dampness in the roosting and scratching quarters." They add: "Damp air compels fowls to increase their already rapid respiration. It is not uncommon to see chickens confined in a damp house, panting on a day that is rather cold. That such a condition is undesirable is so obvious as to need no argument. Relative humidity in an ideal poultry house would never exceed 75, or perhaps 80%."

Barre and Sammet (10) state that: "it undoubtedly is true that poultry can adjust satisfactorily to a considerable range of relative humidity". However, they add: "If the relative humidity is too low, it may result in excessively dry and dusty litter, which may contribute to respiratory irritation." They consider a range of 70 to 80% relative humidity to be desirable during the winter months (11).

b. Design poultry house relative humidity. Design poultry house relative humidity is defined as the maximum relative humidity above which the house will not be permitted to go in cold weather.

Since the main reason of this maximum limit is the avoidance of condensation on walls and ceilings, and perhaps also in the litter, this maximum limit depends on the insulation value of the walls and ceilings. This question will be explained in more detail in section L dealing with the insulation requirements to prevent condensation and some charts and slide rules will be presented to find rapidly the value of this maximum or design relative humidity.

However, it can be said that in general relative humidity below 80% in ordinary weather and around 90% during short cold spells is not detrimental and is considered satisfactory. Ashby, Miller et al (4) state on this respect: "To minimize moisture condensation on walls and ceilings and to avoid odors that might affect egg quality there should be enough air change through the house to keep the relative humidity below 80% in ordinary winter weather."

3. Air purity

a. Carbon dioxide. Early ventilation standards attached great importance to carbon dioxide concentrations.

Discussing the work of other researchers, Barre and Sammet (11) state: "Concentrations as low as 0.02 to 0.07% were not held to be injurious but were previously considered indicative of a dangerous accumulation of "crowd poisons". It was mainly on such opinion that King (63) established standards of ventilation that were followed for many years. The physiological effect of high carbon dioxide concentrations is to increase the rate of respiration, but this result rarely is harmful. Kelley (58) reports that studies subsequent to King's indicate a high tolerance with

respect to carbon dioxide, concentrations as high as 20 to 30% having been breathed for several hours without injurious effect. This tolerance of carbon dioxide may not hold for incompletely developed organisms, chick embryos, for example, are affected adversely in concentrations of carbon dioxide greater than 1.5%."

The general conclusion of Barre and Sammet (11) is that ventilation required for other purposes usually is great enough to eliminate carbon dioxide as a significant environmental factor.

b. Odors. Very little information is available on the physiological effect of odors on hens. According to Barre and Sammet (11), large accumulations of odors affect comfort and may have an adverse effect on the quality of product, as it does for milk. Fairbanks and Goodman (31) also state that: "air that is high in moisture and ammonia and low in oxygen is unpleasant to breathe and is not healthful."

Here also it can be said that ventilation required for other purposes usually is great enough to eliminate odors as a significant environmental factor.

h. Air velocity

• High critical air velocity. High "critical air velocity" is defined as the air velocity at and above which the birds may be subject to cold or pneumonia in extremely cold weather.

High air velocity results in a dangerous draft when the outside air temperature is very cold. For this reason, it is generally recommend to use many air intakes, well distributed around the house and of a total area such as to give a reduced air velocity. The maximum air velocity used in computing the size of air intakes is considered as 700 fpm or 8 mph* by

* fpm and mph: Will be used throughout this text as an abbreviation of "feet per minute" and "miles per hour" respectively.

N. Mitchell (76) and 800 fpm by Barre and Sarnet (11). Such air velocities seem to be the maximum safe velocity for protection against cold and pneumonia.

The design of air intakes to meet these requirements and avoid dangerous drafts will be discussed in more detail in a special section.

5. Litter moisture content

a. Optimum poultry house litter moisture content. Optimum poultry house litter moisture is defined as the litter moisture content or moisture content range within which the birds enjoy the greatest comfort and are least susceptible to disease, and poultry management is complicated by no special problem (such as excessive number of dirty eggs, too frequent additions or changes of litter required, etc.).

Barre and Sarnet (10) discussing the optimum relative humidity for hens report that: "if the relative humidity is low, it may result in excessively dry and dusty litter, which may contribute to respiratory irritation."

Lippincott and Card (6') state that excessive relative humidity, even to the point of continuous wet litter, while not desirable, need not in itself produce detrimental results when the temperature is moderate and fairly constant, however, they add that: "there is no condition under which poultry is kept, unless it is in a state of starvation, that is more surely and quickly fatal to profitable production than dampness in the roosting and scratching quarters. It makes the birds uncomfortable and renders them quite susceptible to disease."

b. Design poultry house litter moisture content. Design poultry house litter moisture content is defined as the maximum litter moisture content above which the moisture content of the litter will not be permitted to go in cold weather.

Such a maximum is given in order to reduce the number of dirty eggs and the frequency of additions or changes of litter (Reduction in labor and material requirements). A general recommendation is to limit the moisture content of litter to below 40% wet basis (4).

B. Heat and Moisture Production of Hens

1. Basal heat and moisture production of birds

The basal heat production of an animal is its minimum energy expenditure. This factor determines the minimum requirements for food energy, and appears to be one of the most constant of biological measurements. Since Mitchell and Kelley's (73) estimates of the heat production of birds are based on early tests on the basal metabolism (71), a review of the literature will be made to see if there is reasonable agreement in the results obtained.

a. Mitchell, Card and Haines basal heat values. In 1927, Mitchell, Card and Haines (71) presented some basal heat data on White Leghorn and White Plymouth Rock chickens up to one year old. These data were obtained by submitting the birds to a fast of forty-eight hours and then placing them in a cylindrical animal chamber of appropriate size, where they were subjected to a respiration test of twenty-two to twenty-three hours duration. The temperature of the laboratory, unfortunately, varied widely during these tests and it may be suspected that the heat production of the birds was raised to some extent above the basal, during the coldest weather. The following table (Table 7) contains the group averages obtained from their experiments.

It is interesting to note that while the basal heat production per square meter of body surface was remarkably constant for all age groups, except the youngest cockerels, the basal heat production per kilogram of body weight decreased continuously with advancing age.

Table 7. Basal heat produced by birds of different weights*

Number of birds	Age (Days)	Weight (Grams)	Surface area (Sq. cm.)	Basal heat production, daily		
				Total (Cals)**	Per kilogram body weight (Cals)	Per square meter body surface (Cals)
COCKS						
4	37	283	326	47	166	1441
6	76	683	783	65	96	832
5	122	1324	1241	108	81	864
5	184	1928	1606	138	71	859
6	242	2705	1979	171	63	864
6	340	2728	1980	169	62	856
PULLETS						
7	94	735	824	65	88	738
6	128	1042	1030	79	75	760
6	192	1450	1290	113	77	861
5	251	1933	1523	120	62	785
6	355	2184	1640	129	60	796

* From data by Mitchell, Card and Haines (71)

* * Cals: Will be used throughout this text as an abbreviation for calories

In previous experiments with nineteen mature Rhode Island Red cocks and twenty-eight non-laying hens, Mitchell and Haines (74) found average values of 806 calories for the cocks and 703 calories for the hens, which is lower than the values obtained for the young White Plymouth Rocks (852 calories for cockerels and 805 calories for pullets). In order to verify this fact, Mitchell, Card and Haines (71) made some tests with four mature White Plymouth Rock hens. The periods of observations were short (30 to 50 minutes) and were taken when the birds were completely at rest. Several determinations were made upon each hen and an average value of 628 calories was obtained which is still lower than the value of 703 obtained with the Rhode Island Red mature non-laying hens. According to the authors, the difference is probably due to a less degree of activity. They concluded: "It appears that the values obtained for the young White Plymouth Rock

(852 and 805 calories) are subject to a considerable decrease at later age.

No data was presented on the basal production of expired moisture.

For convenience, the following table (Table 8) presents the above data in units of pounds (1 pound = 453.6 grams) and in units of Btu (1 Btu = 252 Calories):

Table 8. Basal heat produced by birds of different weights*

COCKEREELS			PULLETS		
Age (Days)	Weight (lb)	Basal heat (Total) (Btu/hr, bird)	Age (Days)	Weight (lb)	Basal heat (Total) (Btu/hr, bird)
37	0.623	7.79	94	1.62	10.75
76	1.503	10.75	128	2.294	13.07
122	2.814	17.86	192	3.194	18.70
184	4.25	22.80	251	4.26	19.85
242	5.96	28.30	355	4.82	21.33
340	6.01	27.90			

* From data by Mitchell, Card and Haines (71)

b. Dukes basal heat and moisture values. In 1937, Dukes (28) reported on a series of tests with Plymouth Rock and White Plymouth Rock hens ranging in age from ten to twenty-two months of age. He concluded: "The average basal metabolic rate of mature hens, after a fast of approximately twenty-four to thirty hours duration was not far from 57.6 calories per kilogram per day and 777.6 calories per square meter per day". This value of 777.6 which corresponds to an environmental temperature of 77°F is somewhat higher than Mitchell and Haines basal metabolism for mature Rhode Island Red (703 calories per square meter per day).

The same author found the heat loss due to vaporization of water (basal) to vary from 12 to 25% of the total heat loss, at 77°F, and to average 17%.

c. Barott and Pringle heat and moisture values. Barott and Pringle (8) presented in 1946 some of their results on the basal metabolism of Rhode Island Red female birds as obtained by the use of the respiration calorimeters of Beltsville, Maryland (Table 2). The original values were in calories per hour per gram of liveweight. The writer converted them to Btu per hour per hen by the use of the conversion formula:

$$\text{Btu/Hr, hen} = \frac{\text{Calories/Hr, gram of liveweight} \times \text{Grams of liveweight}}{252 \text{ calories/Btu}}$$

In general, these values compare reasonably well with the Mitchell, Card and Haines basal heat values (71). One very important observation is the effect of the environmental temperature on the total basal heat production.

1-For birds below two pounds, the basal heat production at the high extreme is about half of that at 40°F.

2-For birds between two and four pounds, the basal heat production at the high extreme is about two-thirds of that at 40°F.

3-For birds above four pounds, the basal heat production is nearly constant, being at the high extreme from 85 to 95% of that at 40°F.

Table 9. Basal heat produced by birds of different weights at different environmental temperatures. (Btu/Hr, bird)*

Age (Weeks)	Weight (Lb)	Temperature (°F)												
		40	45	50	55	60	65	70	75	80	85	90	95	100
0-1	0.0795	-	-	-	-	-	1.83	1.83	1.71	1.44	1.20	0.94	0.79	0.65
2	0.198	-	-	-	-	-	3.87	3.87	3.53	3.13	2.90	2.42	2.23	2.32
5	0.572	-	11.35	11.15	10.43	9.90	9.18	8.42	7.68	6.90	6.38	6.13	6.13	-
8	1.298	21.28	20.00	18.72	17.39	15.90	14.73	13.70	12.75	12.23	11.70	11.59	11.70	-
12	2.266	25.15	23.68	22.68	20.86	19.47	18.40	17.40	16.78	16.34	15.92	15.92	16.14	-
18	3.55	27.14	25.53	23.94	22.68	21.40	20.75	20.12	19.77	19.77	20.12	20.75	21.7	-
23	4.32	28.40	27.20	26.05	24.90	24.10	23.73	23.33	22.95	22.15	23.33	24.10	24.90	-
52	5.35	30.90	29.90	28.48	27.50	27.05	26.60	26.60	26.60	27.06	27.50	28.00	28.93	-
MINIMUM														

* From data by Barott and Pringle (8)

In the same paper Barott and Pringle (8) presented data on the moisture expired by birds analyzed for basal metabolism (Table 10). The original values were in milligrams of water per hour per gram of liveweight. The writer converted them to pounds per day per one hundred hens by the use of the conversion formula:

$$\text{Lb/day, 100 hens} = \frac{\text{mgs/hr, gram of liveweight} \times \text{grams of liveweight} \times 24 \times 100}{1000 \text{ mgs/gram} \times 453.6 \text{ grams/lb}}$$

One interesting observation concerns the range of temperature over which the basal expired moisture was found to remain constant:

- 1-For chickens less than one week old, the rate starts to increase at 95°F.
- 2-For chickens two weeks old, the rate starts to increase at 85°F.
- 3-For chickens from this age up to twelve weeks, the rate starts to increase at 80°F.
- 4-For birds one year old, the rate of basal expired moisture is constant from 40 to 65°F, and it starts to increase quite rapidly, after a temperature of 80°F is reached.

At these higher temperatures, the increase in expired moisture is for purpose of cooling.

It should be noted that this moisture production by the birds is for fasting birds (no feed nor water) being studied for basal metabolism. It is probable that such values do not apply readily to active birds in production.

Table 10. Basal water expired by birds of different weights at different environmental temperatures. (Lb/Day, 100 hens)*

Age (Weeks)	Weight (lb)	Temperature (°F)												
		40	45	50	55	60	65	70	75	80	85	90	95	100
0-1	0.0795	-	-	-	-	-	0.030	0.030	0.030	0.030	0.030	0.030	0.034	0.053
2	0.198	-	-	-	-	-	0.762	0.762	0.762	0.762	0.762	0.866	1.108	1.730
5	0.572	-	-	1.85	1.85	1.85	1.85	1.85	1.85	1.85	1.89	2.38	4.38	7.14
8	1.298	-	-	4.17	4.17	4.17	4.17	4.17	4.17	4.17	4.87	7.51	12.47	-
12	2.266	-	-	6.30	6.30	6.30	6.30	6.30	6.30	6.80	8.46	13.75	24.60	-
52	5.35	7.72	7.72	7.72	7.72	7.72	7.72	8.08	8.74	10.60	20.35	41.10	65.50	-
												FAST INCREASING RATE		
												CONSTANT RATE		

* From data by Barott and Pringle (8)

2. The total heat production of birds

The basal metabolism of birds has been the object of much study, but total heat production of birds has not been so widely investigated although a value of first interest for use in the design of poultry house ventilation. In the absence of test data, some estimates were made by Mitchell and Kelley (73) as early as 1933. These estimates although probably in error have been very useful to engineers in the past and are still used by many because the results from calorimeter tests are still incomplete and unsatisfactory. Both estimates and calorimeter tests data will be presented and discussed here.

a. Mitchell and Kelley values. In 1933, Mitchell and Kelley (73) presented the first practical and comprehensive estimate of the heat production of birds for use in the design of poultry house ventilation systems. The general procedure followed in making the estimates has been to factor out and to assess as accurately as possible all of the items contributing to the energy requirements and expenditures of the birds and then to integrate these factors for birds of different weights.

They considered the total heat production as equal to the sum of:

1-The energy expended in basal metabolism: These values were taken as equal to that reported by Mitchell, Card and Haines (71) in the investigation discussed previously.

2-The energy expended in muscular activity: These values were assumed as equal to one half the basal heat. The justification of this relation for chickens is discussed fully by Mitchell, Card and Hamilton (75).

3-The heat increment due to feed: The heating effect of the feed was taken as 68 calories per one hundred grams of dry matter, this being the heat increment of wheat for chickens (59) (60). In the use of this value of 68 calories, no allowance for a variable heating effect of feed as the

level of feeding is varied could be made in the absence of information on this point. However, it had been found that the heating effect on chickens per gram of corn fed is much the same whether they are eating 50 or 75 grams of corn daily (72).

4-The additional energy requirement for egg production: The three precedent heat elements are for active non-producing hens. For a female bird in production, the net energy requirements are increased as are also the consumption of food and water and the elimination of heat and moisture. These men calculated the heat production increase for a pullet producing one egg a day of average size (2 ounces), to be equal to 3.8 Btu/hr. The writer used only 3 Btu however, since it is evident that not all hens would lay each day.

The values presented by Mitchell and Kelley (73) were expressed in calories. The writer changed them to Btu by using the following conversion formula:

$$\text{Btu/Hr, bird} = \frac{\text{Calories/Day, hen}}{24 \text{ hrs} \times 0.252 \text{ Calories/Btu}}$$

The results are shown in Table 11.

It should be noted that Mitchell and Kelley assumed the total heat production of birds to be constant and not affected by the environmental temperature.

Table 11. Total heat produced by birds of different weights and breeds.*

Breed	Body weight (lb)	Age (Weeks)		Total heat production (Btu/hr. hen)	
		Males	Females	Active non-producing birds	Active birds producing eggs
A) White Leghorn chickens	0.077	0.0	0.0	1.65	-
	0.5	4.0	4.4	13.56	-
	1.0	6.0	8.2	18.19	-
	1.5	9.5	11.5	21.82	-
	2.0	12.5	14.8	25.62	-
	3.0	18.2	22.4	32.07	35.07
	4.0	25.4	38.3	38.85	41.85
	5.0	33.0	-	46.45	49.45
B) White Plymouth Rock chickens	0.077	0.0	0.0	1.65	-
	0.5	4.6	4.6	13.56	-
	1.0	6.6	6.7	18.18	-
	1.5	8.7	8.9	22.3	-
	2.0	9.7	10.7	26.3	-
	3.0	12.4	14.9	33.0	36.0
	4.0	15.7	19.4	39.3	42.3
	5.0	19.3	25.4	46.2	49.2
	6.0	23.3	-	51.2	54.2
	7.0	-	-	58.5	-

* From data by Mitchell & Kelley (73)

b. Barre and Sammet value. Barre and Sammet (11) computed the total heat production of a four pound laying hen to be 60 Btu per hour which is fifty percent more than the value of Mitchell and Kelley (73) for an active non-producing hen fed at the rate of 72 grams of dry matter per day (39.3 Btu per hour). They say: "For birds producing at the average rate of 220 eggs per year, the quantity of feed consumed would be considerably greater. It would be still greater during the periods of highest production. Since the high production occurs during the cold months, when ventilation for moisture removal is most critical, a production rate of 275 eggs per year is chosen for purpose of analysis. At this rate of production the feed requirement is estimated by Jull (53) as being 109 grams. If heat production is assumed to be proportional to the total dry matter fed (an

admittedly rough assumption), the total heat generated is equal to:

$$\frac{10 \text{ Btu} \times 100 \text{ grams}}{72 \text{ grams}} = 60.6 \text{ Btu or approximately } 60 \text{ Btu for a four pound hen}.$$

The heat increment due to feed constitutes only a part of the total heat production as mentioned previously, the other parts being the energy expended in basal metabolism, the energy expended in muscular activity and the additional energy requirement for egg production. Since eggs are collected and taken out of the poultry house every day, they do not constitute a reserve that the bird can utilize as a source of heat for its basal metabolism or its muscular activity. Therefore, the 34 grams of additional feed required to produce an egg are only partially effective in as much as the heat production of the bird is concerned. Only the heat increment value of these 34 grams of additional feed must be considered. As previously mentioned, the heat increment due to feed is equal to 68 Calories per one hundred grams of dry matter, therefore in terms of Btu the additional energy production due to the laying of an egg is:

$$\frac{34 \text{ grams} \times 68 \text{ Calories}/100 \text{ grams}}{100 \text{ grams} \times 24 \text{ hours} \times 0.252 \text{ Calories/Btu}} = 3.8 \text{ Btu/Hr (73).}$$

Barre and Sammet (11) found in their assumption a value of 60.6 - 40 = 20.6 Btu per hour as compared to this value of 3.8 Btu per hour. It should be noted however, that Barre and Sammet value (60 Btu/Hr, hen) although based on a wrong assumption may be used for design if one considers it as including not only the heat given off directly by the birds, but also the heat from secondary sources such as the litter and the sun.

c. Strahan value. In their original table Mitchell and Kelley (73) gave only the heat production of active birds not in production. Many people did not read Mitchell and Kelley's article carefully enough and thought that the heat production given was for laying hens in production. Strahan (99), who wrote extensively on poultry house ventilation, always

used the non-corrected values from Mitchell and Kelley (73) and seems to have misled many people who took his values and didn't refer back to the original article of Mitchell and Kelley.

d. Ota, Garver and Ashby values. Ota, Garver and Ashby (79) presented in March, 1953, their preliminary results from the study conducted with the poultry respiration calorimeters at Beltsville, Maryland. These calorimeters measure 5 feet by 7 feet by 6 feet high and accommodate ten hens.

The authors specify that their report must be considered as preliminary: "indicating trends rather than absolute values, since it was based on tests with a total of only twenty hens, at somewhat varying air flow rates". However, their results are presented here (Table 12) since the writer believes that they are very probably more exact than Mitchell and Kelley's estimates (73).

Table 12 . Total heat produced by 5-lb birds at different environmental temperatures *

Environmental temperature (°F)	Total heat produced by 5-lb birds (Btu/Hr, hen)
40	53.5
45	52.5
50	51.0
55	49.0
60	47.0
65	45.0
70	43.0
75	40.5
80	37.5
85	33.5

* From data by Ota, Garver and Ashby (79)

One important consideration in their findings is the considerable drop in total heat production with an increase in temperature: At 40°F, the total heat production of a five pound bird is 53.5 Btu per hour, per hen, as compared to only 33.5 Btu at 85°F. This reduction of nearly 40% is an aspect of the

heat production that not considered in Mitchell and Kelley's estimates (73), and which was not evident from Barott and Pringle's (8) basal heat data.

Another consideration is the lack of data for temperature below 40°F , although these are the temperatures with which the farm structure engineer is concerned when designing for extremely cold temperatures. It is hoped that heat and water production will be determined soon for temperatures as low as 25°F or even lower.

c. Modified Ota, Garver and Ashby values. Since Ota, Garver and Ashby values (79) were given only for five pound birds, the writer interpolated them for hens from 3 to 6.5 pounds by using the relationship that heat production is directly proportional to the 0.73 power of the liveweight (64) (16). The results are shown in Table 13.

Table 13. Total heat produced by birds of different weights at different environmental temperatures (Btu/hr, hen)

Temperature ($^{\circ}\text{F}$)	Bird liveweight (lb)							
	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5
40	36.9**	41.2**	45.4**	49.5**	53.5*	57.3**	61.1**	64.7**
45	36.2	40.4	44.6	48.7	52.5	56.2	59.9	63.5
50	35.1	39.3	43.3	47.3	51.0	54.7	58.2	61.7
55	33.8	37.8	41.6	45.4	49.0	52.5	55.9	59.2
60	32.4	36.2	39.9	43.5	47.0	50.3	53.7	56.8
65	31.0	34.7	38.2	41.7	45.0	48.2	51.4	54.4
70	29.6	33.1	36.5	39.8	43.0	46.1	49.1	52.0
75	27.9	31.2	34.3	37.5	40.5	43.4	46.2	49.0
80	25.8	28.9	31.8	34.7	37.5	40.2	42.8	45.4
85	23.1	25.8	28.4	31.0	33.5	35.9	38.2	40.5

* From data by Ota, Garver and Ashby (79)

** Calculated by the writer from the relationship that heat is directly proportional to the 0.73 power of the liveweight (64) (16)

The following table (Table 11) is presented for the purpose of comparing the modified Ota, Garver and Ashby values to the 1933 estimates of Mitchell and Kelley (73).

Table 11. Total heat produced by birds of different weights

Weight (lb)	Modified Ota, Garver and Ashby total heat values (79)		Mitchell and Kelley total heat values (73)
	At 40°F	At 60°F	At any temperature
3.0	36.9 *	32.4 *	33.0
4.0	45.4 *	39.9 *	39.3
5.0	53.5 **	47.0 **	46.2
6.0	61.1 *	53.7	51.2

* * From data by Ota, Garver and Ashby (79)

* Calculated by the writer from the relationship that heat is directly proportional to the 0.73 power of the liveweight (54) (15)

From this table, it may be concluded:

- 1-That modified Ota, Garver and Ashby total heat values at 60°F correspond very closely to that from Mitchell and Kelley (73) for any temperature.
- 2-That modified Ota, Garver and Ashby total heat values at 40°F are from 10 to 20% higher than that from Mitchell and Kelley (73) for any temperature.
- 3-That the total heat production of birds at temperatures of 35, 32, 30, 25 and 20°F is probably still higher than that given by Ota, Garver and Ashby at 40°F. Some research is required at these lower temperatures, since these are the temperatures usually used for design.

The writer believes that modified Ota, Garver and Ashby's total heat values (Table 13) based on Beltsville calorimeter tests, although not final, should probably be used instead of Mitchell and Kelley's values (Table 11) which do not consider the effect of environmental temperature.

3. Latent to total heat ratios for birds

The total heat production of birds is composed of two elements:

1-the sensible heat and 2-the latent heat of vaporization. The ratio of these two elements to the total heat production will be discussed here.

a. Mitchell and Kelley values. Analyzing experimental data on geese (40) and some data on other animals, Mitchell and Kelley (73) found a rapidly increasing percentage of heat lost as vaporized water as the environmental temperature increased. This relation appears to be a logarithmic one, and the authors derived the following formula to represent it:

$$L = 7.14 e^{0.06438 t}$$

where L = Percentage of the total heat produced by the bird
that is dissipated as the heat of vaporization of
water (%).

t = Environmental temperature (°C)

From this equation, the latent heat may be computed for the range of 32 to 106°F, using five degree intervals. The results of these computations are presented in Table 15.

Table 15. Percentage of latent heat as affected by environmental temperatures *

Environmental temperature (°F)	Corresponding temperature for use in the formula (°C)	Latent heat (L = %)	Remarks
32	0	7.14	Below 70°F the increase in latent heat is slow.
35	1.7	8.00	
40	4.4	9.47	
45	7.2	11.32	
50	10.0	13.53	
55	12.8	16.26	
60	15.6	19.47	
65	18.3	23.14	Above 70°F the increase in latent heat is fast.
70	21.1	27.70	
75	23.9	33.20	
80	26.7	39.88	
85	29.4	47.30	
90	32.2	56.60	
95	35.0	67.70	
100	37.8	81.10	
106	41.1	100.00	

* From data by Mitchell and Kelley (73)

b. Barre and Carmet value. To simplify computations Barre and Carmet (11) considered the latent heat of vaporization to be constant at 10% for all temperatures below 50°F. Here is what they say: "Since poultry house ventilation for moisture ventilation is not critical at higher temperatures, and the analysis is necessarily approximate, the assumption of a constant ratio of latent to total heat appear to be a justifiable simplification".

c. Ota, Garver and Ashby values. In the paper described in the previous section, Ota, Garver and Ashby (79) presented some latent heat figures (Table 16). However, these values include not only the heat of vaporization of the expired moisture given off by the birds but also include the heat of vaporization of the water evaporated from the litter and drinking fountain.

Table 16. Total latent heat produced by 5-lb birds at different environmental temperatures * *

Environmental temperature (°F)	Latent heat produced by 5-lb birds* (Btu/Hr, hen)
40	21.0
45	21.5
50	22.5
55	23.0
60	23.5
65	24.0
70	25.0
75	26.5
80	27.5
85	28.5

* From data by Ota, Garver and Ashby (79)

* * These latent heat figures include both the moisture from the chickens and that evaporated from the litter and drinking fountain

Expressed in percentage of the total heat production for the given environmental temperature, the total latent heat production is as follow: (Table 17).

Table 17. Total percentage of latent heat at different environmental temperatures *

Temperature	% latent heat **
40	32.3
45	41.0
50	44.1
55	47.0
60	50.0
65	53.3
70	58.1
75	65.5
80	73.4
85	85.1

* From data by Ota, Garver and Ashby (79)

** These latent heat figures include both the moisture from the chickens and that evaporated from the litter and drinking fountain

It is interesting to note that although the total heat production decreases by 37.4% in passing from 40 to 85°F (Table 12) the percentage of latent (birds, litter and drinking fountain) to total heat is increased more than two times its original value (Table 17).

It should be noted that the combined or total latent heat values (birds, litter and drinking fountain) presented by Ota, Garver and Ashby (79) are somewhat more variable than plain latent heat values (birds only), since it is probable that the amount of water evaporated from a litter at a given environmental temperature may be affected differently from one poultry house to another according to such factors as: the type of litter, the depth of litter, the age of the litter, the amount of water spilled around the waterers, the frequency of stirring of the litter, the use of

recirculating fans, the application of lime, etc. The writer feels that, if possible the production of expired moisture and that evaporated from the litter and drinking fountain should be separated in the presentation of such data. The production of expired moisture is probably less variable for a given environmental temperature and would prove more handy in computations.

d. Modified Ota, Garver and Ashby values. From the above review of literature, it can be seen that the only true latent heat production (Heat equivalent to the moisture expired by the birds) is that given by Mitchell and Kelley (73). Since, because of the assumptions on which they are based, these percentages may be in error, some actual data are still awaited from the calorimeter tests at Beltsville, Maryland.

However, for use until such data are available, the writer suggests the use of Mitchell and Kelley latent heat percentages (Table 15) in combination with the modified Ota, Garver and Ashby total heat values (Table 13).

4. Expired moisture produced by birds

Since moisture is the main problem in a poultry house, it is important to know how much of it is produced if the ventilation system is to be designed properly. Moisture is excreted by the birds in two ways: a part is evaporated or expired by the lungs and another part is evacuated in a liquid form in excrements. Ventilation systems should be of sufficient capacity to remove at all times both the expired water in the air and the liquid water in the litter. They should also be equipped with controls to restrict or completely stop the ventilation at the lowest temperatures. Some authors believe that ventilation should be restricted somewhat, but continuous even at very low temperatures, in order to remove at least the expired moisture. By so doing excessive rise in the relative humidity of the house would be avoided and would prevent condensation on the walls and ceilings. The amount of expired moisture produced by birds will be analyzed here.

a. Mitchell and Kelley's values. The total heat production and the percentage of latent heat being known, Mitchell and Kelley (73) values for the expired or evaporated water are given by the following formula:

$$W_e = \frac{q_t \times L \times 100}{1043} = \text{Lb of moisture/Hr, 100 hens}$$

$$\text{or } W_e = \frac{q_t \times L \times 100 \times 24}{1043} = \text{Lb of moisture/Day, 100 hens}$$

where W_e = Water expired per hour at a given environmental temperature
(Lb/Day, 100 hens)

q_t = Total heat produced by one bird (Btu/Hr, hen)

L = Percentage of latent heat corresponding to the given
environmental temperature (%)

1043 = Latent heat of vaporization of one pound of water at
skin temperature or approximately 86°F (Btu/Lb of water
at 86°F)

100 = To convert the value to 100 hens

24 = To convert the value from one hour to a day

Mitchell and Kelley's values (73) are shown in the following table (Table 18). The original values were expressed in grams. The writer converted these values to pounds by the use of the following conversion formula:

$$\text{Lb/Day, 100 hens} = \frac{\text{Grams/Day, hen} \times 100 \text{ hens}}{453.6 \text{ grams/Lb}}$$

Table 18. Water expired by active non-producing birds of different weights and breeds at an environmental temperature of 82°F.*

Breed	Body weight (Lbs)	Age (Weeks)		Moisture expired by birds at 82°F (L = 43.5%) (Lb/Day, 100 hens)
		Males	Females	
A) White Leghorn chickens	0.077	0.0	0.0	1.54
	0.5	4.0	4.4	12.32
	1.0	6.0	8.2	16.75
	1.5	9.5	11.5	20.05
	2.0	12.5	14.8	23.6
	3.0	18.2	22.4	29.5
	4.0	25.4	28.3	35.7
	5.0	33.0	-	42.7
B) White Plymouth Rock chickens	0.077	0.0	0.0	1.54
	0.5	4.6	4.6	12.32
	1.0	6.6	6.7	16.75
	1.5	8.7	8.9	20.5
	2.0	9.7	10.7	24.2
	3.0	12.4	14.9	30.4
	4.0	15.7	19.4	36.1
	5.0	19.3	25.4	42.5
	6.0	23.3	-	47.1
	7.0	-	-	53.7

* From data by Mitchell and Kelley (73)

Mitchell and Kelley (73) found the amount of water expired by the birds to be mainly affected by the environmental temperature. Assuming that the latent to total heat ratio holds true for birds of any age, both non-productive and productive and using the three Btu of heat increment due to egg production, the writer computed the expired moisture for active birds not in production and for active birds in production (Table 19). The range

Table 19. Water expired by birds of different weights at different environmental temperatures. (Lb/Day, 100 hens)*

Type of bird	Weight (Lb)	Environmental temperature (°F) and Percent latent heat (L-%)											
		38°	35°	30°	40°	45°	50°	55°	60°	65°	70°	75°	80°
A) Active birds not in production	0.077	0.270	0.303	0.359	0.390	0.390	0.512	0.617	0.739	0.876	1.05	1.65	3.78
	0.5	2.22	2.49	2.95	3.52	3.52	4.21	5.08	6.07	7.20	8.64	13.52	31.1
	1.0	2.97	3.33	3.96	4.70	4.70	5.64	6.80	8.15	9.67	11.58	18.12	41.7
	1.5	3.33	3.72	4.42	5.27	5.27	6.30	7.60	9.10	10.78	12.92	20.28	46.7
	2.0	4.32	4.83	5.72	6.84	6.84	8.16	9.85	11.80	13.95	16.75	26.28	60.4
	3.0	5.41	6.07	7.18	8.57	8.57	10.24	12.38	14.80	17.50	21.00	32.94	75.7
	4.0	6.45	7.20	8.56	10.21	10.21	12.42	14.72	18.10	20.86	25.00	39.3	90.3
	5.0	7.58	8.59	10.04	12.00	12.00	14.32	17.32	20.70	24.50	29.40	46.2	106.2
	6.0	8.39	9.41	11.20	13.28	13.28	15.87	19.15	20.29	27.15	32.50	51.1	117.5
	7.0	9.60	10.75	12.73	15.20	15.20	18.14	21.92	26.20	31.10	37.25	58.5	134.5
B) Active birds producing eggs	3.0	5.92	6.63	7.85	9.37	9.37	11.20	13.50	16.17	19.15	22.92	36.1	83.0
	4.0	7.30	7.80	9.24	11.03	11.03	13.17	15.90	19.03	22.75	27.00	42.4	97.6
	5.0	8.09	9.05	10.72	12.80	12.80	15.28	18.43	22.06	26.15	31.26	49.3	113.4
	6.0	8.98	9.95	11.76	14.04	14.04	16.80	20.36	24.25	28.70			

* From data by Mitchell & Kelley (73)

of 32 to 70°F was considered and intervals of five degrees were used. Since the difference between breeds is very small, for a given weight, only the weight of the birds is considered here.

It should be noted that for the purpose of winter ventilation, only the values for temperatures below 50°F are of interest. For a temperature of 50°F and for birds of 3 to 7 pounds, the amount of expired water is found to vary from approximately ten to eighteen pounds per hour per one hundred hens. For a temperature of 32°F, this amount would be in the range from five to ten pounds. The high values of forty to sixty pounds of expired moisture obtained at temperature around 62°F are either in error or else may be explained as coming from body reserves or from higher rate of water consumption. (It might also be accompanied by a smaller moisture content of the excreta, although no data was found on this subject).

b. Barre and Sammet values. Using a total heat production of 60 Btu (wrong assumption as shown previously) and a 10% latent heat (average latent heat for temperatures between 32 and 50°F), Barre and Sammet (11) consider the evaporated water produced by a four-pound laying hen to be:

$$W_e = \frac{q_t \times L \times 24 \times 100}{1013} = \frac{60 \times 10 \times 24 \times 100}{1013} \\ = 13.78 \text{ Lb/Day, 100 hens (For the range of 32 to 50°F)}$$

as compared to 7.30 (at 32°F) and 11.17 (at 50°F) as given by Mitchell and Kelley (73).

c. Strahan values. Using a total heat production of 36.89 Btu (non-corrected value as shown previously) and a 7.24 latent heat (latent heat for 35°F), Strahan (99) consider the evaporated water produced by a four-pound Leghorn bird to be:

$$W_e = \frac{q_t \times L \times 24 \times 100}{1043} = \frac{36.89 \times 7.2 \times 24 \times 100}{1043}$$

$$= 7.05 \text{ Lb/day, 100 hens (For 35°F)}$$

as compared to 7.80 (at 35°F) as given by Mitchell and Kelley (73).

d. Ota, Garver and Ashby values. As mentioned in the previous section, Ota, Garver and Ashby (79) were unable to determine the ratio of the water expired by the birds to the total water evaporated. This fact makes it impossible to compute from their results the amount of moisture expired by the birds.

However, the authors (79) have determined the total latent heat equivalent to the moisture expired by the birds plus the moisture evaporated from litter and drinking fountain. This total latent heat (birds, litter and drinking fountain) can be transformed to pounds of moisture evaporated per hour per one hundred hens, by the use of the following formula:

$$W_e = \frac{Q_1 \times 100}{1043} = \text{Lb of moisture/Hr, 100 hens}$$

where W_e = Water evaporated per hour at a given environmental temperature (birds, litter and drinking fountain)
(Lb/Hr, 100 hens)

Q_1 = Total latent heat production (moisture from birds + moisture from litter and drinking fountain) corresponding to the given environmental temperature

1043 = Latent heat of vaporization of one pound of water at skin temperature or approximately 86°F (Btu/Lb of water at 86°F)

100 = To convert the value to 100 hens.

It should be noted that this formula includes a small error of approximately 2 to 3% since the moisture evaporated from the litter and drinking fountain do not require the same amount of latent heat as that expired by the birds. In the first case, the latent heat value should be between 1066 and 1076 Btu (for a litter temperature between 50 and 32°F) as compared to 1043 Btu required for each pound of moisture expired by the birds (for a skin temperature of about 86°F). Nevertheless, in order to get an approximate idea of the total moisture evaporated under the experimental conditions, the writer used the above formula and the results are shown in Table 20. The ventilation rate used in the experiment was approximately one cfm per hen. These data are therefore of some interest since they give some idea of how much total water can be exhausted from well insulated houses having ventilation rates of one cfm of air per hen.

Table 20. Total amount of water evaporated per each 5-lb laying hen at different environmental temperatures* *

Environmental temperature (°F)	Total amount of water evaporated per each 5-lb laying hen* (Lb/Day, 100 hens)
40	48.5
45	49.7
50	51.8
55	53.1
60	54.2
65	55.5
70	57.6
75	60.9
80	63.2
85	65.5

* From data by Ota, Garver and Ashby (79)

* * This evaporated moisture include both the moisture from the birds and that evaporated from the litter and drinking fountain

e. Modified Ota, Garver and Ashby values. Since latent heat and expired moisture are dependent, the conclusion to be drawn is similar to that in the previous section: the only true value for the moisture expired by the birds is that given by Mitchell and Kelley (73); since some assumptions may have caused these percentages to be in error, some actual data are still awaited from the calorimeter's tests at Beltsville, Maryland.

Until such data are available, the percentages presented by Mitchell and Kelley (Table 15) can probably be used. It is believed that these percentages are on the high side and therefore involve a factor of safety. These percentages should probably be used with the modified Ota, Garver and Ashby total heat values (Table 13) which are based on recent calorimeter tests, instead of Mitchell and Kelley's values (Table 11) which do not consider the effect of environmental temperature.

The writer has computed these modified values for the expired moisture of birds. Values for birds having a body weight different from five pounds were interpolated by using the relationship that heat production is directly proportional to the 0.73 power of the liveweight (64) (16). The results are shown in Table 21.

Table 21. Water consumed by birds of different weights at different environmental temperatures (Lb/Day, 100 hens)

Temperature (°F)	L (%)	Bird liveweight (Lb)							
		3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5
		**	**	**	**	***	**	**	**
20	5*	4.57	5.10	5.61	6.12	6.61	7.09	7.55	8.00
25	6*	5.35	5.98	6.59	7.20	7.77	8.34	8.89	9.40
30	7*	6.05	6.88	7.58	8.26	8.94	9.58	10.2	10.8
35	8	6.92	7.72	8.5	9.3	10.0	10.7	11.4	12.1
40	9.47	8.04	9.0	9.9	10.8	11.6	12.5	13.3	14.1
45	11.32	9.45	10.5	11.6	12.7	13.7	14.7	15.6	16.6
50	13.53	10.51	12.2	13.5	14.7	15.9	17.0	18.1	19.2
55	16.26	12.6	14.1	15.1	17.0	18.3	19.7	20.9	22.2
60	19.47	14.5	16.2	17.9	19.5	21.1	22.6	24.1	25.5
65	23.14	16.5	18.5	20.3	22.7	23.9	25.6	27.4	29.9
70	27.70	18.8	21.1	23.3	25.4	27.4	29.4	31.3	33.1
75	33.20	21.3	23.9	26.2	28.7	31.0	33.2	35.3	37.5
80	39.88	23.7	26.5	29.2	31.8	34.4	36.9	39.3	41.7
85	47.30	25.2	28.1	30.9	33.8	36.5	39.1	41.7	44.1

* Values below 40°F were extrapolated by the writer for convenience until actual test data are made available

* * Calculated by the writer from the relationship that heat is directly proportional to the 0.73 power of the liveweight (64) (16)

* * * From data by Ota, Garver and Ashby (79) and by Mitchell and Kelley (73)

The following table (Table 22) shows the change brought about in Mitchell and Kelley's values (73) for five-pound hens, by using the total heat production found by Ota, Garver and Ashby (79).

Table 22. Water expired by 5-lb laying hens at different environmental temperatures (Lb/Day, 100 hens)

Temperature (°F)	Mitchell and Kelley's * values (73)	Ota, Garver and Ashby modified values (79)	Remark
20	-	6.61	Below 50°F,
25	-	7.77	Ota, Garver and
30	8.09	8.94	Ashby modified
35	9.05	10.0	values are just
40	10.72	11.6	slightly higher
45	12.80	13.7	than Mitchell
50	15.28	15.9	and Kelley's values
55	18.13	18.3	Above 50°F,
60	22.06	21.1	Ota, Garver and
65	26.15	23.9	Ashby modified
70	31.26	27.4	values are lower
75	37.30	31.0	than Mitchell
80	43.30	34.4	and Kelley's
85	53.20	36.5	values

* Modified by the writer to include the effect of egg production (Refer to Table 19)

f. Slide rule for determining the expired moisture produced by birds.

The writer has prepared two slide rules to give readily the amount of expired moisture produced by the birds, once the total heat production and the latent to total heat ratio are known. Since it was desirable to complete these two slide rules in such a way as to obtain the total moisture to be removed from the house, these slide rules will be presented in section 6 dealing with the total water eliminated by birds.

5. Excretion of liquid water by birds

To maintain dry litters, it is important to design the house and the ventilation system in such a way as to produce conditions favorable to the evaporation of the moisture evacuated daily in the litter. The amount of this water will be analyzed here.

a. Mitchell and Kelley values. Mitchell and Kelley (73) computed water contained in the excreta from the dry matter consumed on the assumption that dry matter was 20% indigestible (or 80% digestible) and that the excreta contained 27% of dry matter (or 73% of water). The latter value was obtained from a number of published digestion experiments on chickens in which the mixed excreta were analyzed for moisture. The authors (73) recognized that this value is, of course, subject to variation. The original values were expressed in grams. The writer converted these values to pounds by the use of the following conversion formula:

$$\text{Lb/Day, 100 hens} = \frac{\text{Grams/Day, hen} \times 100 \text{ hens}}{453.6 \text{ grams/Lb}} = \text{Grams/Day, hen} \times 0.2205$$

The writer also added the column for laying hens. The original values presented by Mitchell and Kelley (73) were only for active non-producing birds fed at the rate of 72 grams of dry matter per day. For egg production 34 additional grams of dry matter are required per day. Increasing the dry feed consumption brings about an increase in water consumption. Furthermore, for laying birds, there must be an additional increase of water consumption due to water storage in egg. (An average egg weighs approximately two ounces or 58 grams and contain 75% water, meaning that $58 \times 0.75 = 43$ grams of water go into each egg laid). The latter excrement of water is not a problem since eggs are collected and removed from the house daily. However, the increase in water consumption due to the increase in dry feed consumption must be added to the moisture content of the excrement given for non-producing birds.

Mitchell and Kelley, using a factor of 2.2, found this water consumption increase to be $34 \times 2.2 = 75$ grams. The results are presented in Table 23.

Table 23. Liquid water eliminated in the excrement of birds of different weights and breeds, at an environmental temperature of 82°F.

Breed	Body weight (lb)	Age (Weeks)		Moisture content of the excrement at 82°F (Lb/day, 100 hens) *	
		Males	Females	Active non-producing birds	Active birds producing eggs
A) White Leghorn	0.007	0.0	0.0	0.88	-
	0.5	4.0	4.4	4.63	-
	1.0	6.0	8.2	5.95	-
	1.5	9.5	11.5	7.27	-
	2.0	12.5	14.8	8.15	-
	3.0	18.2	22.4	9.90	26.40
	4.0	25.4	38.3	11.90	28.40
	5.0	33.0	-	13.45	29.75
B) White Plymouth Rock chickens	0.007	0.0	0.0	0.88	-
	0.5	4.6	4.6	4.63	-
	1.0	6.6	6.7	6.17	-
	1.5	8.7	8.9	7.71	-
	2.0	9.7	10.7	9.25	-
	3.0	12.4	14.9	11.68	30.2
	4.0	15.7	19.4	13.22	29.75
	5.0	19.3	25.4	14.98	31.5
	6.0	23.3	-	14.98	31.5
	7.0	-	-	16.52	-

* From data by Mitchell and Kelley (73)

b. Modified Ota, Garver and Ashby values. The effect of environmental temperature on the rate of feed and water consumption was studied recently by Ota, Garver and Ashby (79). The values presented in the following table (Table 24) are for one hundred five-pound laying hens and indicate the trend rather than exact values:

Table 24. Trend in feed and water consumption of 5-lb laying hens as affected by environmental temperatures *

Environmental temperature	Feed consumption (Lb/Day, 100 hens)	Water consumption (Lb/Day, 100 hens)
35	35	65
40	34	62
45	33	59
50	32	58
55	31	57
60	30	57
65	29	56.5 Minimum
70	28	57
75	27	57
80	26	58
85	25 Minimum**	58

* From data by Ott, Garver and Ashby (79)

** Minimum for the range of temperatures studied

From this table the amount of feed consumed appears to increase linearly with a drop in temperature, while the fountain water consumed appears to be least at temperatures between 55 and 75°F. At the lower temperatures more feed is consumed and more water is needed to utilize the nutrients. At the higher temperatures less feed is eaten and less water is used for metabolism, but more is needed to permit evaporative cooling by panting.

Goldrich (36) and Bates (12) suggest the following equation to find roughly the total amount of water eliminated by the birds.

$$W_T = W_F - W_E$$

where W_T = Total water eliminated by the birds (Lb/Day, 100 hens)

W_F = Fountain water consumed by the birds (Lb/Day, 100 hens)

W_E = Water taken out in eggs

As mentioned previously, an egg of average size weighs about two ounces and contains about 75% water. The water taken out in eggs is therefore given by the equation

$$W_E = \frac{100 \times E \times 2 \times 0.75}{16} = \frac{150}{16} E = 9.375 E$$

where W_E = Water taken out in eggs (lb/Day, 100 hens)

100 = Number of hens considered

E = Percentage of egg production (%)

2 = Average weight of an egg (ounces)

0.75 = Water content of a typical egg

16 = To convert the weight of water from ounces to pounds

The writer has computed the amount of water taken out in the eggs produced by one hundred laying hens when producing at different levels.

The results of these computations are presented here (Table 25).

Table 25. Water taken out in eggs

% of egg production	Water taken out in eggs (lb/Day, 100 hens)
30	2.01
35	3.28
40	3.75
45	4.24
50	4.69
55	5.16
60	5.62
65	6.09
70	6.56
75	7.03
80	7.50
85	7.97
90	8.44
95	8.92
100	9.38

The following table (Table 26) shows the trend in the total amount of water eliminated by one hundred 5-pound laying hens at different environmental temperatures using the water consumption values found by Ota, Garver and Ashby (79) and applying Goldrich's (36) and Bates's (12) assumption as presented above.

Table 26. Trend in the total amount of water eliminated by 5-lb laying hens at different environmental temperatures

Environmental temperature	Trend in water consumption*	Trend in egg production*	Trend in water removed in eggs**	Trend in total water eliminated**
(°F)	(lb/day, 100 hens)	(%)	(lb/day, 100 hens)	(lb/day, 100 hens)
35	65	62	5.8	59.2 Max***
40	62	59	6.5	55.5
45	59	74	6.9	52.1
50	58	77	7.2	50.8
55	57	78 Max	7.3 Max	49.7
60	57	77	7.3	49.8 Min
65	56.9 Min	75	7.0	49.9
70	57	73	6.8	50.2
75	57	67	6.3	50.7
80	58	62	5.8	52.2
85	58	57	5.3	52.7

* From data by Ota, Garver and Ashby (79)

** Calculated by the writer on the assumption that the total water eliminated by the birds is roughly equal to the amount of fountain water consumed by the birds minus the water taken out in eggs (34) (12)

*** Maximum for the range of temperatures studied

By subtracting the total amount of water eliminated (Table 26) from the amount of expired water produced by five-pound laying hens (Table 21), the amount of water excreted in the litter was computed and is presented in the following table (Table 27).

Table 27. Trend in liquid water eliminated in the droppings of 14-lb laying hens at different environmental temperatures

Environmental temperature (°F)	Trend in total water eliminated* (Lb/Day, 100 hens)	Water expired by birds** (Lb/Day, 100 hens)	Trend in liquid water excreted in the droppings (Lb/Day, 100 hens)
35	52.2 Max***	10.0 Min***	42.2 Max***
40	55.5	11.6	43.9
45	52.1	13.7	38.4
50	50.8	15.9	34.9
55	49.7 Min	18.3	31.4
60	49.8	21.1	28.7
65	49.5	23.9	25.6
70	50.2	27.4	22.8
75	50.7	31.0	19.7
80	52.2	34.4	17.8
85	52.7	36.5 Max***	16.2 Min***

* Calculated by the writer (Table 26)

** Calculated by the writer (Table 21)

*** Maximum and minimum for the range of temperatures studied

Not all the water excreted must be evaporated, however, since the droppings have to dry only from 77% moisture content, wet basis, which is the average moisture content of fresh droppings, to 35% moisture content, wet basis, which is the average value of a reasonably dry litter. The percentage of excreted moisture to be evaporated from the litter is given by the formula:

$$F = (A - E) \frac{100}{A}$$

$$F = [A - (D - C)] \frac{100}{A}$$

where A = Pounds of water excreted in the droppings of
100 hens per day

B = $\frac{A}{0.77}$ = Weight of droppings at 77% moisture content,
wet basis

C = B - A = Pounds of dry matter in droppings

$$D = \frac{C}{0.65} = \text{Weight of droppings at 35\% moisture content, wet basis}$$

$$E = D - C = \text{Pounds of water remaining in the droppings at 35\% moisture content, wet basis}$$

$$F = \frac{(A - E) 100}{A} = \text{Percentage of excreted moisture to be evaporated from the litter to maintain the litter moisture content at 35\%}$$

By substitution of the values mentioned, the percentage of excreted moisture to be evaporated from the litter (F) was found equal to 83.9%. Using this 83.9% value, the total water to be removed from the droppings of one hundred 5-pound laying hens was computed and the results are presented here (Table 28).

Table 28. Trend in the water to be evaporated from the droppings of 5-lb laying hens at different environmental temperatures, in order to maintain a litter moisture content of 35%

Environmental temperature (°F)	Trend in liquid water excreted in the droppings* (Lb/Day, 100 hens)	Trend in the water to be evaporated from the droppings to maintain a litter moisture content of 35% (Lb/Day, 100 hens)
35	49.2	41.2
40	43.9	36.8
45	38.4	32.2
50	34.9	29.3
55	31.4	26.3
60	28.7	24.1
65	25.6	21.5
70	22.8	19.1
75	19.7	16.5
80	17.8	14.9
85	16.2	13.6

* Calculated by the writer (Table 27)

Because of the absence of data on the water consumption of laying hens of liveweight other than five pounds, the writer was unable to make similar calculations for birds of other weights.

c. Other values. Considering the feeding rate of 102 grams per hen per day, Barre and Sammet (11) estimated the excreted moisture of four-pound hen to be about 0.27 pound per day or 27 pounds per one hundred hens per day. This value was obtained by assuming a litter moisture of 40%.

Oliver (78) states that: "repeated tests by leading college poultry departments, reliable textbooks on good poultry management, and personal observations have shown that six gallons of water per day is required to satisfy one hundred heavy breed hens which are laying well. Of this six gallons, that is, fifty pounds of water, approximately five to six pounds are removed in the eggs gathered. The balance, approximately 45 pounds of water must be removed by ventilation of which about 9 pounds are given off by the hens breathing, the remaining, 36 pounds is contained in the excreta".

According to Jull (53) "... at prevailing temperatures in the majority of poultry houses during winter it is safe to assume a respiratory moisture output of 11.66 pounds per day and a total output of 45 pounds per day for each one hundred hens." This represent $45 - 11.66 = 33.34$ pounds of water excreted daily in the litter.

Hill (43) reports: "Strictly fresh droppings contains approximately 85% moisture. Although influenced by body size, feed consumption and rate of lay (The higher the rate of lay, the greater the amount), fecal moisture approximates 33.3 pounds per day for each one hundred hens."

According to Goodman (37): "Observations at the Cornell Experiment Stations indicate that hens weighing 4.5 pounds each, produce approximately 144 pounds of droppings per year. This checks closely with findings at the other northeastern stations. It is agreed that droppings may safely be assumed to contain 70% water. On this basis the moisture eliminated in the dropping per day, per one hundred hens, would be: $\frac{144 \text{ lb} \times 70\% \times 100 \text{ hens}}{365 \text{ days}} = 27.6$ pounds of water per day, per one hundred hens. This corresponds to

$\frac{27.6 \text{ lb} \times 1070 \text{ (Latent heat of water at } 100^{\circ}\text{F)}}{24 \text{ hrs} \times 100 \text{ hens}} = 12.15 \text{ Btu/hr, hen.}$ The latter value represents the amount of heat required if that water is to be evaporated from the litter.

Yushok and Bear (199) give the amount of liquid voided by four-pound hens as about 0.25 pound per day or 25 pounds for each one hundred hens.

To conclude from the above review of literature, the writer believes that the modified Ota, Garver and Ashby values for five-pound laying hens (Table 28) represent the best available data since they are the only ones to consider the effect of the environmental temperature. Some research is required however, in order to give more accurate values for birds of different weights and different levels of egg production, when submitted to different environmental temperatures. The effect of the protein level of the diet should also be investigated.

6. Total water eliminated by birds and total water to be removed from the house

The total water eliminated by birds is composed of the expired moisture plus the liquid water excreted in the litter.

a. Mitchell and Kelley values. Mitchell and Kelley (73) estimated the total water excreted by active hens not in production to be as follows at an environmental temperature of 82°F (Table 29). The original values were in grams. The writer converted them to pounds by using the following conversion formula:

$$\text{Lb of water/Day, 100 birds} = \frac{\text{Grams/Day, bird} \times 100}{453.5 \text{ grams/lb}}$$

Table 29. Total water excreted by hens and total water to be removed from the house at 82°F (lb/Day, 100 hens) *

Body weight (lb)	Total water excreted by hens		Total water to be removed from the house	
	Active hens not in production	Active hens producing eggs	Active hens not in production	Active hens producing eggs
0.007	2.42	-	2.38	-
0.5	16.95	-	16.21	-
1.0	22.9	-	21.93	-
1.5	26.2	-	26.97	-
2.0	33.5	-	31.98	-
3.0	42.1	58.62	41.20	54.1
4.0	49.3	65.82	47.20	61.1
5.0	57.5	74.02	55.08	69.0
6.0	62.1	78.62	59.68	73.6
7.0	69.8	86.32	67.58	81.4

* Calculated by the writer from data by Mitchell and Kelley (73)

It should be noted that the amount of water to be removed from the house would be somewhat lower since the excrements only need to be dry from 77% to 35% (wet basis).

b. Modified Ota, Garver and Ashby values. Table 26 in the previous section shows the total amount of water eliminated by five pound laying hens at different environmental temperatures, for an egg production rate of 60%. However, a certain part of the liquid water excreted in the litter does not need to be evaporated since drying the droppings from 77% to 35% (wet basis) is usually considered satisfactory.

The writer computed the total water to be removed from the house by adding the expired moisture to the moisture to be evaporated from the droppings. The results for five-pound laying hens are presented in the following table (Table 30):

Table 30. Trend in the total amount of water to be removed from poultry houses sheltering 5-lb laying hens at different environmental temperatures, in order to maintain a litter moisture content of 35% **

Environ- mental- temper- ature (°F)	Expired mois- ture* (Lb/Day, 100 hens)	Trend in the water to be evaporated from the drop- pings to maintain a litter moisture content of 35% ** (Lb/Day, 100 hens)	Trend in the total amount of water to be removed from poultry houses to maintain a litter moisture content of 35% (Lb/Day, 100 hens)
35	9.3 Min***	41.2 Max***	50.5 Max***
40	10.8	30.8	47.6
45	12.7	32.2	44.9
50	14.7	29.3	44.0
55	17.0	26.3	43.3
60	19.5	24.1	43.6
65	22.7	21.5	43.2
70	25.4	19.1	44.5
75	28.7	16.5	45.2
80	31.8	14.9	46.7
85	33.8 Max***	13.6 Min***	47.4

* Calculated by the writer (Table 21)

** Calculated by the writer (Table 28)

*** Maximum and minimum for the range of temperatures studied

c. Other values. Jull (53) states that: "... at prevailing temperatures in the majority of poultry houses during winter it is safe to assume a respiratory moisture output of 11.66 pounds per day and a total output of 45 pounds per day for each one hundred hens."

According to Bell and Moore (14): "A flock of one hundred birds gives off at least 22 quarts of water every twenty-four hours." This represents 5.5 gallons of water per day or $5.5 \times 8.34 = 45.8$ pounds of water per day per one hundred hens.

Kable (55) presents a somewhat lower value; he considers the water vapor expired per day, at 50°F, as being one-eighth per pound per day and the water excreted in the droppings as one-fourth pound per day, which gives a total of three-eighths pounds per day, per hen or 37.5 pounds per day per one hundred hens.

According to these three references, it seems therefore, that under ordinary conditions, i.e. for house temperatures below 50°F, the amount of total water eliminated by birds is around 40 to 45 pounds. The amount of total moisture to be removed is somewhat lower, since a part of the liquid water in the excreta remains in the litter, as shown in the previous section.

Concluding from the above review of literature the writer believes that the modified Ota, Garver and Ashby values for five-pound laying hens represent the best available data since they are the only ones to consider the effect of the environmental temperature. Some research is required however, in order to get more accurate values for birds of different weights and different levels of egg production, when submitted to different environmental temperatures. The effect of the protein level of the diet should also be investigated.

d. Slide rule for determining the expired moisture produced by birds and the total moisture to be removed from the house. The writer has prepared two slide rules to find the expired moisture produced by the birds and the total moisture to be removed from the house. The first slide rule is on an hourly basis and is derived from the following equations:

$$1) W_{e1} = \frac{q_t \times L \times 100}{1043}$$

$$\text{and } 2) W_{eT} = W_{e1} + W_{e2}$$

where W_{e1} = Water expired by the birds (Lb/Hr, 100 hens)

q_t = Total heat produced by the birds (Btu/Hr, hen)

L = Percentage of latent heat produced by the hens (%)

1043 = Latent heat of vaporization of one pound of water

at skin temperature or approximately 86°F

(Btu/Lb of water)

100 = To convert the value to 100 hens

W_{e2} = Liquid moisture to be evaporated from the litter

(Lb/Hr, 100 hens)

$W_{eT} = W_T$ = Total moisture to be removed from the house

(Lb/Hr, 100 hens)

The first equation was changed to the following logarithmic form:

$+(\log q_t) - (0) + (\log L) - (\log 1043 W_{e1}) = (0)$. A middle support was required for the conversion of $\log (1043 W_{e1})$ to ordinary values of W_{e1} .

The second equation was given the form: $+(W_{e1}) - (0) + (W_{e2}) - W_{eT} = (0)$.

The final value obtained for W_{eT} (or W_T) is that to be used in the formula for moisture removal:

$$V_L = \frac{W_T}{w_1 - w_0} \quad (\text{To be explained later})$$

The second slide rule is derived in a similar way, except that the first equation is multiplied by 24 to convert the value to a daily basis.

As an illustration of the use of these two slide rules, consider the following applications:

Application No. 1 (Slide rule No. 1)

Given: Total heat production of the birds = 35 Btu/Hr, hen.

Percentage of latent heat produced by the hens = 20%

Liquid moisture to be evaporated from the litter = 1.1 Lb/Hr, 100 hens.

Wanted: The total weight of moisture to be removed from the house, hourly.

Solution: (Use of slide rule No. 1)

1-Put the first arrow opposite $q_t = 35$ and read $W_{e1} = 0.67$ opposite

$L = 20\%$.

2-Put the second arrow opposite $W_{e1} = 0.67$ and read $W_T = 1.77$

opposite $W_{e2} = 1.1$.

Application No. 2 (Slide rule No. 2)

Given: Total heat production of the birds = 42 Btu/Hr, hen.

Percentage of latent heat produced by the birds = 17%

Liquid moisture to be removed from the litter = 23.5 Lb/Day, 100 hens.

Wanted: The total weight of moisture to be removed from the house, daily.

Solution: (Use of slide rule No. 2)

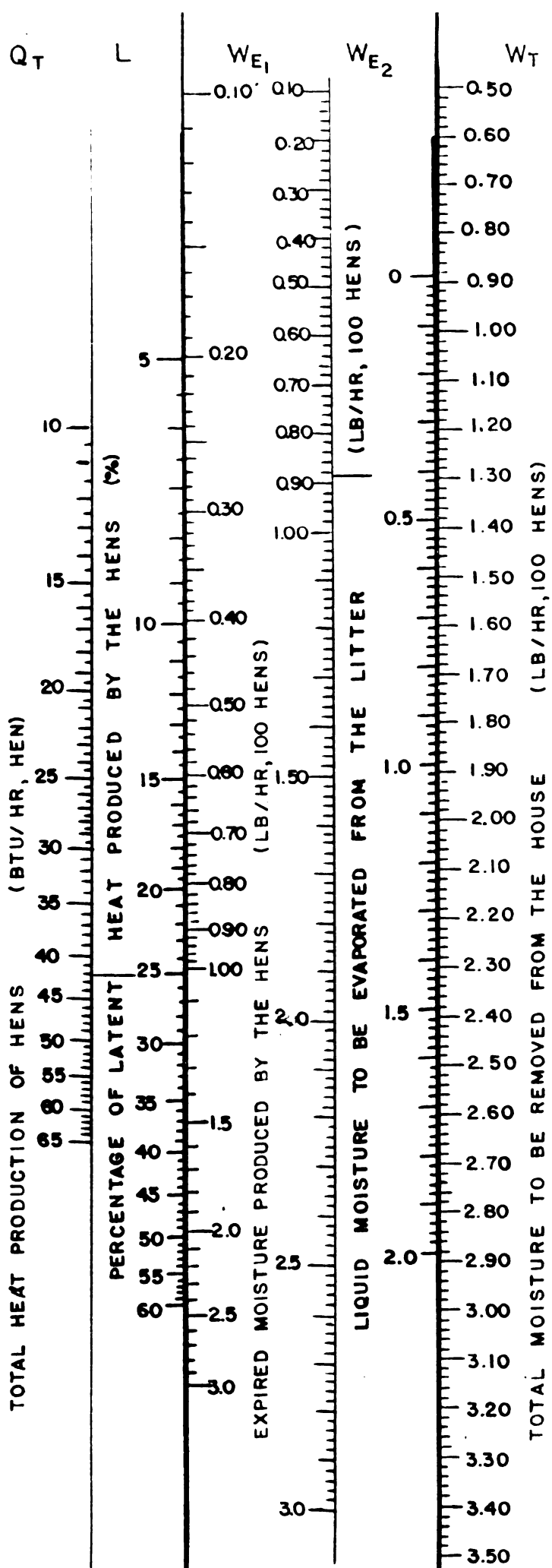
1-Put the first arrow opposite $q_t = 42$ and read $W_{e1} = 16.43$

opposite $L = 17\%$.

2-Put the second arrow opposite $W_{e1} = 16.43$ and read $W_T = 39.93$

opposite $W_{e2} = 23.5$

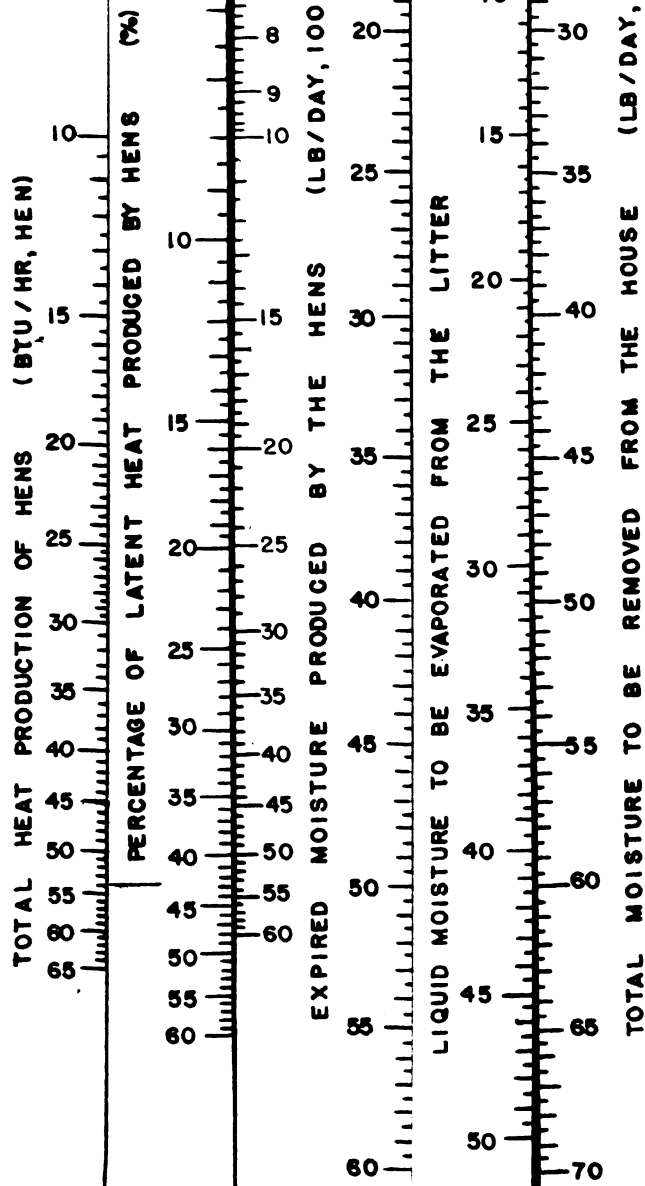
The two above slide rules could easily be combined with the two slide rules prepared to solve the moisture removal equation. These two slide rules will be presented in a later section.



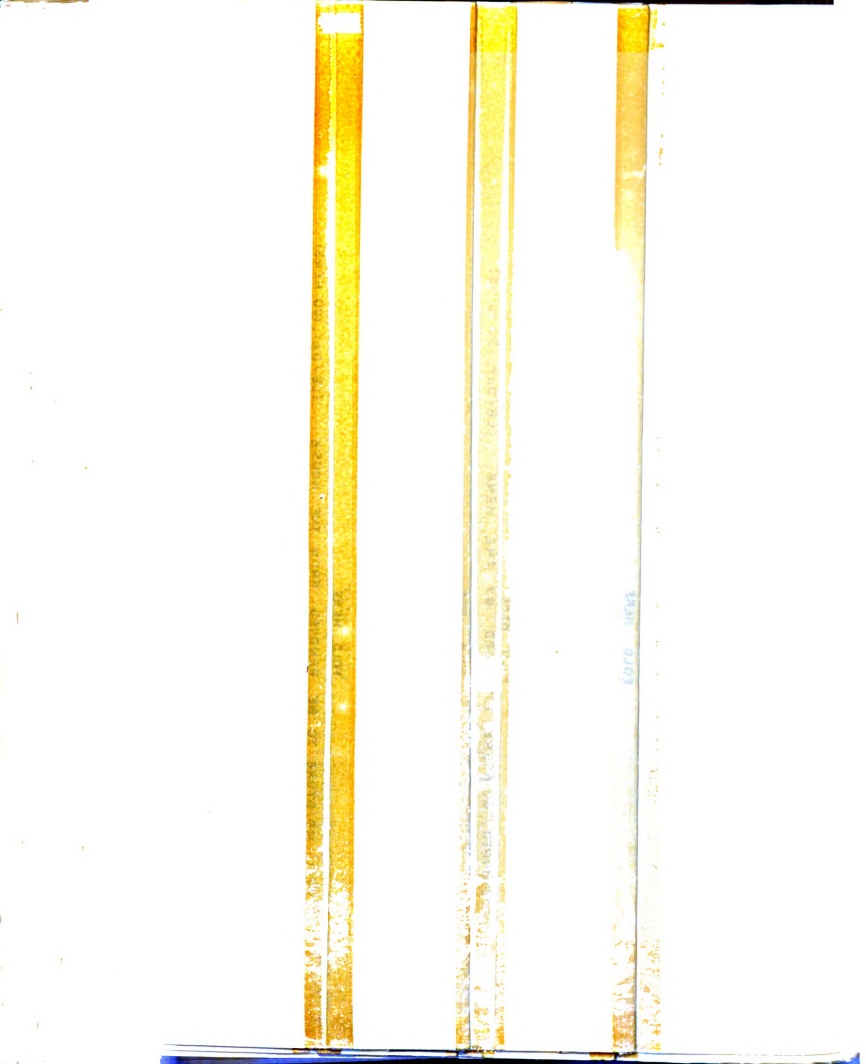
() SLIDE RULE FOR DETERMINING THE TOTAL
 MOISTURE TO BE REMOVED FROM THE HOUSE (HOURLY)

1070

CH. 4202



SLIDE RULE FOR DETERMINING THE TOTAL
MOISTURE TO BE REMOVED FROM THE HOUSE (DAILY)



7. Net sensible heat of birds and net available sensible heat in the house

The net sensible heat of birds is that part of the total heat produced by the birds which can be used for (1) evaporation of moisture from the litter and drinking fountains, (2) compensation for the heat loss through the building materials, and (3) heating of the cold incoming air (ventilation and infiltration). When deep litters are used it is customary to consider the heat produced by litter decomposition to be sufficient to evaporate the moisture from the droppings. In such cases, the net sensible heat of birds is therefore used only for purpose of (1) compensating the heat loss through the building and (2) compensating the heat loss through the ventilation system and through the infiltration by the cracks.

It is also customary to add a certain amount of heat to take care of the heat gain from the sun and other secondary heat sources. The total of the net sensible heat from the birds and these secondary heat sources is called the total available heat produced in the house and is represented by Q_s or Q_{total} in the formula for temperature control:

$$Q_{\text{total}} = \frac{60 V_s}{t_2} - ACD \quad (\text{To be explained later})$$

a. Mitchell and Kelley values. Mitchell and Kelley values (73) for the net sensible heat of birds may be obtained by subtracting from the total heat production, the product of the total heat production and the percentage of latent heat for the given temperature. By referring to sections 2 and 3, these values will be found. The writer will present at the end of the present section a slide rule which gives readily the amount of net sensible heat produced by the birds. This slide rule can be used not only for Mitchell and Kelley values (73) but also for Ota, Garver and Ashby values (79) or for any data to be obtained in the future.

b. Giese and McCormick values. In 1933, Giese and McCormick (34) reported on the heat production of poultry under housing conditions as found at the Iowa Agricultural Experiment Station. The experiments referred to were conducted in two tiers of five pens each. Each pen was 7 feet by 5 feet 6 inches in plan and 3 feet 6 inches in height. Ten White Leghorn birds were confined in each pen or one bird for each 3.85 square feet of floor space. This battery was entirely inclosed within another building to eliminate the effect of sunlight. It was possible to add feed and water and to remove the eggs through stoppered pipes and small felt-sealed doors with practically no effect on the interior conditions. Five of the ten pens were open to natural ventilation. The quantities applied to the five pens with controlled air supply were 0.054, 0.15, 0.70, 2.63 and 9.93 cfm per hen per minute.

According to their computations based on daily, weekly and monthly averages of the intake and outtake air heat and moisture content, (a) the net sensible heat available would vary considerably with the outside temperature and (b) the amount of water vaporized from the birds and the litter would be reduced considerably at lower outside temperature because of the greater amount of heat used for the building and ventilating losses. Their results are summarized here (Table 31).

Table 31. Net available heat production (Btu/Hr, °F, hen)*

Outside temperature range (°F)	Total heat characteristic of the house = Building loss + Ventilation loss (Btu/hr, °F, hen)						
	1	1.5	2.0	2.5	3.0	3.5	4.0
0-10	36.0	52.0	63.0	72.5	77.0	-	-
11-20	32.0	46.5	58.5	67.0	71.5	-	-
21-30	28.5	40.0	52.0	60.0	64.5	67.0	-
31-40	25.0	36.0	45.5	53.0	57.0	60.0	-
41-50	18.5	30.0	39.0	47.0	51.5	54.5	56.5
51-66	15.5	26.0	33.5	39.0	45.0	48.0	51.0

* From data by Giese and McCormick (34)

c. Ota, Garver and Ashby values. Ota, Garver and Ashby (39), in their paper described previously, presented the net sensible heat production of birds at different environmental temperatures (Table 32). These values differ from the total sensible heat production which includes in addition, the heat produced as sensible heat by the birds but later used for the vaporization of water from the litter and drinking fountains. This net sensible heat usable for temperature control is for the ten hen calorimeter used in the test and may not be representative of actual poultry houses conditions, or at least not representative of all of them.

The net sensible heat of a five-pound bird is found to decrease from 32.5 to 5.0 Btu/Hr, hen, for an increase in temperature from 40 to 85°F.

Table 32. Net sensible heat produced by 5-lb laying hens at different environmental temperatures *

Temperature (°F)	Net sensible heat produced by 5-lb laying hens ** (Btu/Hr, hen)
40	32.5
45	31.0
50	28.5
55	26.0
60	23.5
65	21.0
70	18.0
75	14.0
80	10.0
85	5.0

* From data by Ota, Garver and Ashby (79)

* * This net sensible heat was obtained by subtracting the total latent heat (birds, litter and drinking fountains) from the total heat production

d. Slide rule for determining the net sensible heat of birds and the net sensible heat available in the house. The writer has prepared a slide rule to find the net sensible heat produced by the birds and the total sensible heat available in the house. This slide rule is based on the following formulas:

$$1) \quad q_{s1} = q_t - q_t \times L = q_t (1 - L) = q_t \times S$$

$$\text{and } 2) \quad Q_s = q_{s1} + q_{s2}$$

where q_{s1} = Sensible heat produced by the birds (Btu/Hr, hen)

q_t = Total heat produced by the birds (Btu/hr, hen)

L = Percentage of latent heat produced by the birds (%)

S = Percentage of sensible heat produced by the birds (%)

q_{s2} = Secondary sources of net sensible heat produced in the house: Litter, Sun, etc. (Btu/Hr, hen)

Q_s = Total available net sensible heat produced in the house (Btu/Hr, hen)

The first equation was changed to the following logarithmic form:

$+(\log q_t) - (0) + (\log S) - (\log q_{s1}) = (0)$. Values of $\log S$ were plotted and values of $\log L$ were substituted for them. A middle support was required for the conversion of $\log q_{s1}$ to ordinary values of q_{s1} .

The second equation was given the form: $+(q_{s1}) - (0) + (q_{s2}) - (Q_s) = (0)$. The final value obtained for Q_s is that to be used in the formula for temperature control:

$$Q_s = \frac{60 V_s}{52} - ACD \quad (\text{To be explained later})$$

As an illustration of the use of this slide rule, consider the two following applications.

Application No. 1

Given: Total heat production of the birds = 42 Btu/Hr, hen.
Percentage of latent heat produced by the birds = 13%
Secondary sources of net available sensible heat (sun, litter, etc.) = 7.5 Btu/Hr, hen.

Wanted: The total available sensible heat produced in the house

Solution:

- 1-Put the first arrow opposite $q_t = 42$ and read $q_{s1} = 36.50$
opposite $L = 13\%$
- 2-Put the second arrow opposite $q_{s1} = 36.50$ and read $Q_s = 44.0$
opposite $q_{s2} = 7.5$.

Application No. 2

Given: Total heat production of the birds = 33 Btu/Hr, hen.
Percentage of latent heat produced by the birds = 7.5%
Secondary sources of net sensible heat (sun, litter, etc.) = 5.5 Btu/Hr, hen.

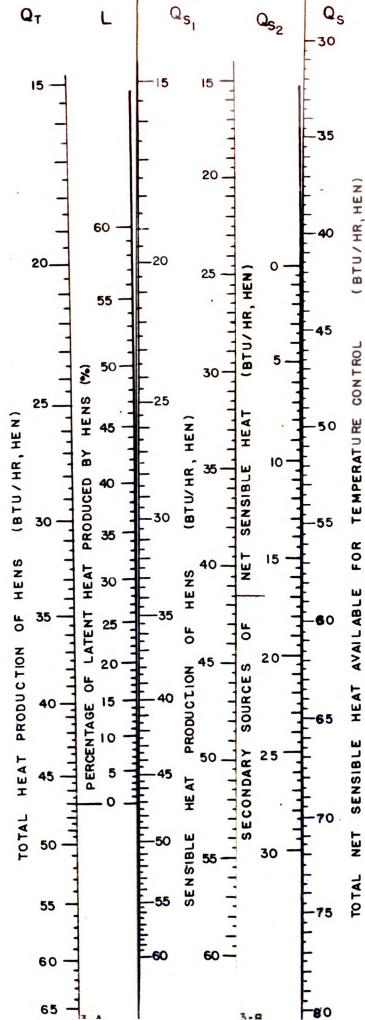
Wanted: The total available sensible heat produced in the house

Solution:

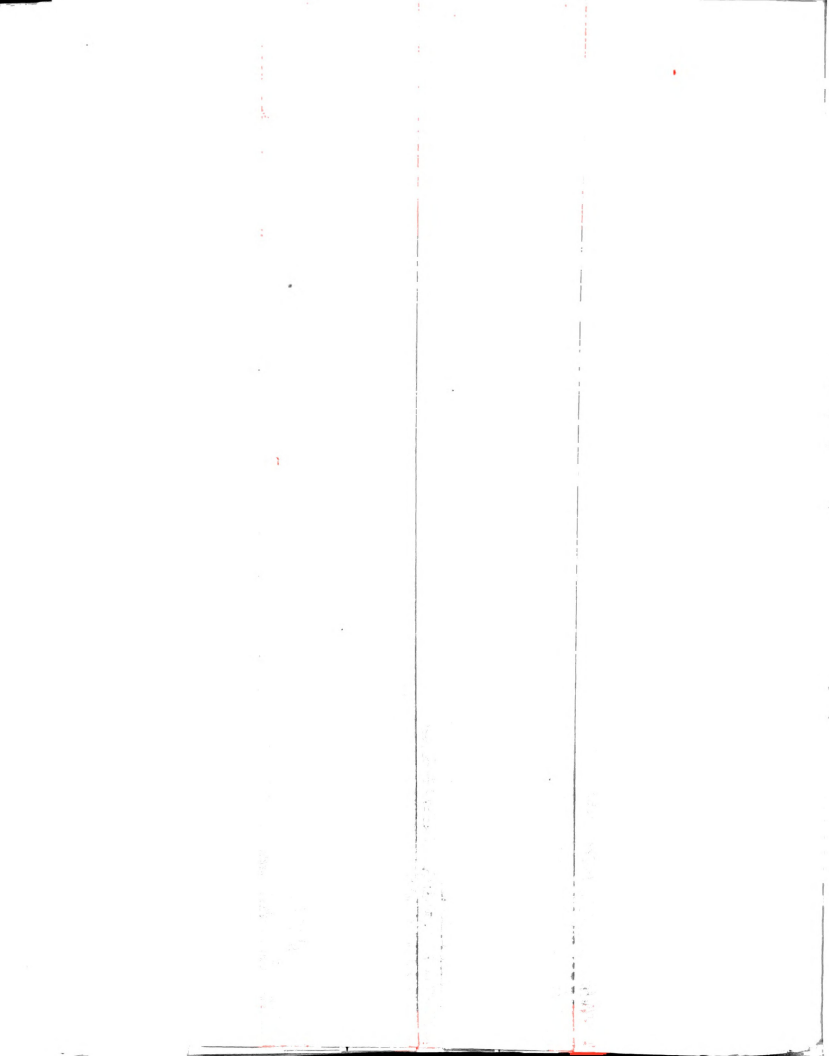
1-Put the first arrow opposite $q_t = 38$ and read $q_{s1} = 30.50$
opposite $L = 7.5$.

2-Put the second arrow opposite $q_{s1} = 30.50$ and read $q_s = 36.0$
opposite $q_{s2} = 5.5$.

The above slide rule could be easily combined with the slide rule prepared to solve the temperature control formula. This second slide rule will be presented in a later section.



SLIDE RULE FOR DETERMINING THE TOTAL SENSIBLE
HEAT AVAILABLE FOR TEMPERATURE CONTROL



C. Secondary Sources of Heat and Moisture

1. Secondary sources of heat

Although the birds are the main source of heat in poultry houses, there are numerous secondary sources of heat gain. These are:

- 1-Solar heat through the windows
- 2-Solar heat gain of the walls and roofs
- 3-Heat generated in the litter
- 4-Heat transferred from earth through floor
- 5-Heat stored in the building materials and the equipment
- 6-Latent heat from condensing moisture
- 7-Heat from light bulbs and other equipment
- 8-Heat from personnel
- 9-Heat from the water, feed and equipment entering the house.

All of these secondary sources of heat will be discussed here.

a. Solar heat through windows. Discussing solar heat radiation as a source of secondary heat for the coldest design period in poultry houses, Parker (82) states: "Since the coldest periods usually occur just before sunrise, solar radiation will not be considered as a source of heat. Solar heat stored during sunny days will help maintain higher night temperatures, especially in masonry structures (Writer's note = Or in other structures having large thermal capacity). Since solar radiation is uncertain for a given period and the effect of solar lag is not well established, solar heat will not be considered in this type of problem."

Whatever be the unknowns in solar heat radiation, there is however one certain thing: Insulating glass windows can help quite appreciably in solving the problem of temperature control, especially in areas of much winter sunshine.

Winkle (44) using the solar radiation data collected by the Michigan Hydrologic Research Station on a horizontal surface made some theoretical computations for East Lansing to determine the net heat gain or the net heat loss through fixed insulating windows as compared to a single glazed ordinary window. He found that for the winter of 1952-53 (December 10 to March 11) the average daily solar heat gain was equal to 380 Btu per square foot of insulating glass which, when spread over a twenty-four hour period, corresponds to about 14 Btu per square foot of insulating glass per hour. These results compared favorably with the results obtained for an industrial atmosphere by the use of a Sun Angle Calculator published by the Libbey-Owens-Ford Glass Company. With this instrument, the result for an average day for an industrial atmosphere was 309 Btu per square foot of insulating glass which corresponds to 13 Btu per square foot per hour. For a clear atmosphere the daily average value given by this instrument was 421 Btu per square foot of insulating glass per day which corresponds to 20 Btu per square foot per hour.

The heat loss through insulating windows is mostly by conduction since these windows are fixed and the heat losses by infiltration or crack leakage are negligible. Taking the U value for insulating glass as equal to 0.58 and using an average temperature difference of 20 degrees during the winter of 1952-53, Winkle (44) found the average heat loss to be 11.6 Btu per square foot of insulating glass per hour. He therefore calculated the average net heat gain as being equal to $14 - 11.6 = 2.4$ Btu per square foot of insulating glass per hour. For a clear atmosphere the net heat gain would be $20 - 11.6 = 8.4$ Btu per square foot per hour.

According to the same author (44) if fixed single glazed windows are used instead of insulating glass windows, the amount of heat intake is increased a small amount while the heat loss is about doubled. The solar

radiation heat gain through such a window for an industrial atmosphere would be 16 Btu per square foot per hour and the heat loss through the same window would equal about 22.8 Btu per square foot per hour. The result would therefore be a net heat loss of $22.8 - 16 = 5.2$ Btu per square foot of fixed single glazed window per hour as compared to a net gain of 2.4 Btu per square foot per hour for the insulating glass. For a clear atmosphere the heat gain of a fixed single glazed window would be approximately equal to $\frac{20 \times 16}{13} = 24.6$ Btu per square foot per hour, leaving a net gain of $24.6 - 22.8 = 1.8$ Btu per square foot per hour.

In conclusion, with fixed insulating glass in an industrial or equivalent atmosphere, an hourly average of 7.6 additional Btu are made available for litter drying and temperature control for each square foot of glass as compared to fixed single glazed windows. For a clear atmosphere, the extra heat made available would be equal to only 6.6 Btu per square foot of insulating glass per hour as compared to fixed single glazed windows.

For ordinary glazed windows (not fixed), the difference would be still greater because in such a case the main source of heat loss is due to infiltration of cold air around the windows (as much as 110 cubic feet of air per hour per foot of crack). However, with thermostatically controlled fans, this factor is of little consequence since it only results in having the fan operation reduced by an amount equal to the infiltration volume.

Stapleton and Cox (95) conducting a heat balance test on a poultry house computed an average of 29.2 Btu of solar gain for each square foot of ordinary glass: "The solar radiation was determined from readings at the Blue Hills Observatory located at Milton, Massachusetts. These showed a total of 1680 Btu/Hr. actually incident on the outside surface of the

glass. This is based on an average of 250.1 g. cal. per sq. c. per hour for the period December 14, 1948 to February, 1949. Of the 1600 Btu, part was considered to be absorbed by the glass, part transmitted through the glass and part reflected. Handbook sources indicate that for the prevailing conditions, 1462 Btu per hour would have penetrated a clean glass. Light meter readings were used to determine the reduction in transmissivity due to dust. These gave a final determination of 1276 Btu per hour input from solar sources." The glass area was 43.7 square feet, which made an average of 29.2 Btu of solar gain for each square foot of ordinary glass. This is about 1.8 times more than the 16 Btu per square foot of ordinary glass per hour computed by Hinkle (44) for an industrial atmosphere, and about 1.2 times more than this value of 24.6 Btu for a clear atmosphere.

In a personal letter to the writer, Hinkle made the following statements: "The area around the Blue Hills Observatory is usually snow covered during parts of the winter and the atmosphere is clear. Both of these factors will increase the amount of radiation coming into a south facing window. Lansing on the other hand has more clouds and the overall results give an atmosphere comparable to that of Chicago or Pittsburg. Added to this was the fact that even though my calculations can be extended to fit a clear atmosphere as you did, snow cover during the winter of my tests was about one half of average. Thus if there had been an average snow cover, my results would have approached the 29 Btu per square foot per hour obtained by Stapleton and Cox. On this basis I say that their calculations are as valid as mine. They both have the same fault, however, that they truly apply to just one particular winter. The Sun Angle Calculator was designed primarily for summer use for air conditioning calculations and makes no compensation for a snow cover. Thus the results from this instrument are also only approximate. The agreement between the three of us is very good however."

The conclusions of Winkle (44) from his summer and winter tests on the use of insulating glass in hog houses were:

- 1-During summer months, the fixed windows in the solar wing did not cause higher temperatures than the regular wing where ventilating windows were used.
- 2-Where proper ventilation is supplied by other means ventilating windows should offer no advantage over fixed windows.
- 3-During winter months, fixed windows would have an advantage because of the negligible heat loss due to crack leakage.
- 4-When a small number or size of pigs were housed in both wings, solar energy gains became more important in determining the temperature of the solar wing.

Conclusions 3 and 4 would apply readily to poultry houses. However, conclusions 1 and 2 would probably be different, since hog houses are usually constructed with adjacent concrete yards and the hogs are allowed to go out part of the day, therefore reducing the heat and moisture production inside of the house. In commercial poultry houses, the birds are generally confined at all times and the use of fixed windows would require considerable amount of forced ventilation unless some of them are removed for the purpose of summer ventilation.

b. Solar heat gain and heat loss through walls and roofs. While it is true that the majority of the heat gain due to solar radiation will be through large south facing windows, other places of solar heat gain bear mentioning. This would include the other parts of the structure such as the walls, roofs and windows oriented in other directions.

Saia (90) made some estimations of the heat which could be gained during the winter months through different types of walls. He found that for a black wall facing south, about 60 Btu per square foot will be gained for

every 1000 Btu of heat gain per square foot of insulating glass. For a wall which faces east or west, the heat gain would probably be as low as 20 Btu per square foot.

The winter solar heat gain through a roof is even harder to estimate because of the slope of the roof. Winkler (14) states that: "an estimation based on the estimations for solar heat through a south facing wall would probably locate the heat gain through a black surface gable roof running north and south at about 25 Btu per square foot for every 1000 Btu per square foot of insulating glass windows facing south. A black surface shed roof opening to the south would probably transmit less than 5 Btu per square foot for every 1000 Btu per square foot of insulating glass facing south. When all the other approximations of heating calculations are considered, this heat gain through a shed roof could be neglected."

Recent investigations have shown that, on a clear winter night, radiation to the sky from the surface may take place at a rate sufficient to cool a building surface, particularly the roof, to a temperature 10 to 15°F below the air temperature" (11). According to Angstrom (3) the average radiation loss at night for a clear sky is in the range of 30 to 45 Btu per square foot per hour. It should be noted that the amount of cloud covering on radiation losses at night can be just as important as the amount of cloud covering on total radiation gains during the day (3) (14).

Because of the lack of exact data, the solar heat gain through walls and roofs during the day and the radiation heat loss of these areas to the sky at night are usually not considered for design purposes.

c. Heat generated in the litter. Parker's opinion (82) concerning the heat generated in the litter is as follows: "Little information is available on the heat produced in deep litter. Undoubtedly this factor would vary considerably with litter management. New shallow litter would produce very

little heat if any. Therefore, this heat source, where available, is considered as a factor of safety." He also adds: "During cold snaps, unless the cold outside air is allowed to infiltrate under the floor, the air temperature in the house will be cooler than the temperature of the litter; therefore, no allowance should be made for transfer of heat to the litter when designing for critical cold periods. Thus, the sensible heat available for the coldest design periods is taken as the sensible heat produced by the birds."

Stapleton and Cox (91) conducting a heat balance test on a poultry house concluded that at an average temperature of 48.5°F., 22.46 Btu was produced in the litter for each six-pound hen in the house of which 12.10 was utilized as latent heat to dry the litter and 10.36 was utilized for temperature control. In their computations they used a total heat production of 52 Btu per six-pound bird as given by Mitchell and Kelley (73) for active non-producing birds. Recent data by Oto, Garver and Ashby (79), however, indicates a probable heat production of 50.2 Btu per hour by a six-pound bird at 50°F. It is therefore possible that their finding be a little too high although the writer in his own experiment (to be discussed later) found the total heat from secondary sources to be very close to that found by Stapleton and Cox.

Goldrich (36) considers the heat production of deep litters as generally sufficient to evaporate the moisture in the excrements and even considers it as giving out some extra heat to help in the maintenance of the inside temperature.

White and Schwantes (107) observed that in four well insulated Minnesota laying houses the litter became objectionably wet until built-up litter was used. After this the litter remained in good condition as compared to similar houses using shallow litter, where the litter had to be changed every three to five weeks.

The writer took some readings of the litter and soil temperature in the center of Mr. Jacob Schwilke's poultry house, Clinton, Dakota. The readings were taken with a standard thermometer and the results are shown here:

Depth of the litter or soil (inches)	Air temperature at floor level (°F)	
	13	27
Temperature of the litter (°F)		
1 1/2	56	57
2	56	57
3	56	57
4	56	57
5	56	57
6	56	57
7	56	57
Temperature of the dry soil (°F)		
1	54	59
3	54	59
7	54	59

A thermometer placed at a different spot in the same house indicated temperatures varying from 56 to 60°F, three inches below the top of the litter. The temperature of the litter at this depth seemed to increase with an increase in the temperature of the air at the floor level:

Air temperature at floor level (°F)	Litter temperature 3 inches below the top of the litter (°F)
56-60	51-55
56-59	50-57
56-55	55-60

According to these values, the temperature of a deep litter seems to be much higher than the temperature of the air at floor level and would indicate that deep litter are giving considerable heat to evaporate the water they contain and to warm up the house. With proper location of air intakes condensation in the litter should never occur.

From the above review of literature it follows that the amount of heat produced in deep litter, although not fully known, is probably very important for the purpose of litter drying and temperature control. Some research is required to determine the amount of heat generated in deep litter under various environmental temperatures and various litter management conditions.

d. Heat transferred from earth through floor. Ashby, Miller et al (4) state: "In a house with floor or walls in contact with the ground the heat from the earth helps to warm the house whenever the air temperature is lower than that of the ground. Ground heat helps especially to avoid extreme drops in house temperature during cold spells." Discussing this factor, Parker (92) states: "Heat from the earth may help in preventing rapid temperature drops during cold spells. As no information is available on this topic, this source is likewise considered as a factor of safety."

Most other ventilation designers have also neglected the heat loss or heat gain through the floor. Strahan (89) states on this subject: "Although recognizing the probable effect of this factor under actual working condition, by eliminating this variable factor for the present, it is possible to get a reasonable comparison between house designs which may differ in other respects." According to Shier (92): "Where barns are built against earth banks, that portion of the wall in contact with the earth need not be included in calculating the insulating figures, as the temperature of the ground is usually close to that desired in the stable during winter months."

Referring to the writer's data on deep litter and soil temperature presented in the previous section, it is seen that the earth temperature under a poultry house litter seems to be much higher than the temperature of the air at floor level, although slightly less than the deep litter

temperature immediately at its surface. This would mean that with deep litter the earth is probably receiving some heat from the litter instead of giving some.

More detailed research covering a greater period (greater range of environmental and outside temperature) and utilizing better instrumentation is required to determine the ground temperature at different depths, and at different distances from the foundation walls, in some typical poultry houses with both earth and concrete floors.

c. Heat stored in the building materials and the equipment. The heat capacity of building materials is a very useful property in that it tends to maintain equilibrium temperature in the house when there is a sudden drop in the outside temperature. This property may be defined as the specific heat of the material per unit volume, that is, the heat necessary to raise the temperature of one cubic foot of this material through one degree Fahrenheit.

Values of specific heat and heat capacity for several building and insulating materials are given in the following table (Table 33) published by Allen, Walker and Jones (1)

Table 33. Specific heat and heat capacity of some building and insulating materials

Material	Specific heat (Btu/lb, °F)	Average density (lb/cu. ft.)	Heat capacity (Btu/cu. ft, °F)
Building materials:			
Brickwork	0.1950	100.0	19.5
Masonry	0.2000	150.0	30.0
Pine	0.4470	35.0	15.65
Oak	0.4700	43.0	20.11
Glass	0.1977	124.0	24.52
Insulating materials:			
Glass wool	0.210	3.0	0.63
Rock wool	0.261	4.0	1.2
Corkboard	0.417	3.0	1.3
85% magnesia	0.276	15.0	4.1
Corrugated asbestos paper	0.245	17.0	4.2
Wood fiberboard	0.241	14.5	3.5

From Allen, Walker and James (1)

f. Latent heat from condensing moisture. When moisture vapor condenses on a surface, some latent heat is given off. However, it is customary to neglect this amount of heat for purpose of design. The reasons for so doing are:

- 1-The amount of condensing water is difficult to estimate
- 2-Much of the latent heat is released on the cold surface where the moisture condenses and is transferred through the wall or window to the outside.

g. Heat from light bulbs and other equipment. Heat from light bulbs and other equipment can be included by the designer if desired. The formula for computing its value is as follow:

$$Q_L \& E = W \times 3.413$$

where $Q_L \& E$ = Heat given off by light bulbs and other equipment (Btu/Hr)

W = Total wattage of the light bulbs and equipment (Watts)

3.413 = Heat equivalent of one electrical watt

h. Heat from personnel. Heat from the personnel can also be accounted for by the designer. The heat given off by the human body varies with the condition. In a room at 70°F, the values are as given in Table 34 (1).

Table 34. Heat given off by the human body under various conditions (in a room temperature at 70°F)*

Condition	Btu/Hr
Adults at rest	440
Adults at work	450-600
Adults at violent exercise	600-1200
Children	240

*From Allen, Walker and Jones (1)

Thompson (101) uses a value of 760 Btu per hour per man for all temperatures, with a latent heat percentage ranging from 12% of this total at 50°F to about 100% of the total at 95°F.

i. Heat from the water, feed and equipment entering the house. This secondary source of heat is considered in experimental work. However, for purpose of design, its value is usually small and negligible. Furthermore, in actual practice, the result is usually a heat loss instead of a heat gain, since these items are usually stored in colder rooms previous to entering the poultry house.

2. Secondary sources of moisture production

Although the birds are the main sources of moisture in poultry houses, there are several secondary sources of moisture production. These are:

- 1-Water spilled, lost (leaks), or evaporated from the waterers
- 2-Water seepage through floors
- 3-Water from entering rain or snow
- 4-Moisture diffused through walls
- 5-Water resulting from the decay of organic matter (unknown)
- 6-Water on egg when laid (negligible)
- 7-Water in the equipment entering the house (negligible)
- 8-Moisture condensed from the entering ventilating air into the litter
- 9-Drips from wall, windows and ceiling where moisture has condensed

Factors 6 and 7 are usually so small to be considered negligible and factor 5 is actually unknown. The five other factors mentioned above will be discussed here.

a. Water spilled, lost (leaks), or evaporated from the waterers. Water spillage has a definite influence on the condition of the litter. According to Charles et al (19) some method to reduce this spillage or to prevent its retention in the litter should be adapted. They therefore recommend the use of water catching devices and the use of a litter which has a low water retention value. These authors measured the percent of water spilled around the waterers to average 2.15% or 0.406 gallons per bird per year, in their tests on poultry house litter moisture.

Experiments by other authors (19) (24) (45) indicate that the amount of water spilled and evaporated from the waterers may amount to about 7% of the water consumed by the birds. If there are some leaks or if some defective valve causes an overflow, this amount may be increased still more.

Dann (27) states: "Waterers must be kept leak-proof and arranged so that the hen stands in a comfortable position to drink. If she must reach too far she will withdraw her head and the drop from her beak and wattles goes on the litter below. The caretaker should also avoid spillage in filling, emptying or cleaning waterers."

Charles et al (19) report that excessive dampness about the waterer can hardly be controlled by litter additions, since these additions merely serve to absorb the spillage and a wet soggy mass results about the waterers.

Poultrymen should pay the greatest attention to the installation and care of their watering system in order to reduce the importance of this factor as a secondary source of moisture.

b. Water seepage through floors. Inadequate drainage around the foundation of the building and under the concrete floors that are on the ground are sometimes a very important cause of secondary source of water. In fact, Dann (27) states on good authority that moisture may be raised by capillary action some seven or eight feet through sandy or gravelly soil, and twice or three times this distance through the heavier soils. He recommends the use of a roofing material between the floor surface and the ground as being essential in practically every poultry house where the floor comes in contact with the ground underneath.

The poultryman should be aware of these facts and if he wants to solve his wet litter problem, he should provide for adequate drainage around and under the building in combination with the use of roofing materials to cut the capillary action. According to Shier (22) this roofing material should not be vapor-proof: "The use of vapor-proof paper in the floor will not prevent damp litter, and may in some cases increase the dampness in the litter by preventing the moisture from moving downward to the colder earth below."

c. Water from entering rain or snow. Rain and snow, where houses are not well constructed, may be an important source of moisture. Leaky roofs should be repaired as soon as noticed and open windows should be protected from entering rain or snow. It is assumed that for good construction, rain, snow and ground water seepage will be negligible.

d. Moisture diffused through walls. Discussing this factor, Parker (12) states: "If vapor barriers are used with insulation as recommended for good construction, the moisture diffused through the walls should be relatively small, especially during the critical moisture removal periods when the outdoor relative humidity is high. Therefore, diffusion of moisture through walls is considered negligible."

It seems to the writer that any moisture diffusion, if appreciable, would result in a moisture loss rather than in a moisture gain, since the greatest amount of diffusion is from the high vapor pressure warm inside air toward the cold low vapor pressure outside air.

e. Moisture condensed from the entering ventilating air. According to Charles, Topper et al. (12), it does not seem probable that moisture is deposited from the outside air directly, since winter outside air is usually lower in temperature and humidity than the poultry house air and therefore, could not add moisture to the poultry pen from its own way. Cold air entering a poultry pen can however, indirectly increase litter moisture since it does, in mixing with warmer, more humid air, chill it below its dew-point and cause the precipitation of some of its moisture. According to these same authors, dry conditions are optimum for the removal of litter moisture from the pen. The reason is that the higher temperature of the poultry house favors evaporation from the litter and the humidity of incoming air being lower, allows this air when warmed to absorb more moisture.

Eller (10) states that dampness of the litter and bedding of poultry and farming houses is often caused by condensation of moisture from the air whenever the floor of the house becomes cold enough to be below the dew-point of the air in the house. He adds: "In Ohio the remedy usually is better insulation to conserve heat and a suitable ventilation system. A small amount of condensation of the litter is not harmful."

Dates (12) concluded from his analysis of some poultry houses: "In the houses studied the litter remained relatively dry under the roost where the birds spent less time and litter tended to be wet in the front of the house where they congregated. This would add to the belief that moisture concentration in litter is not a problem of condensation but one of removing the moisture deposited by the droppings."

f. Drip from wall, windows, and ceiling where moisture has condensed.
Drip from wall, windows, and ceiling where moisture has condensed from the air of the poultry house may be of some importance as a secondary source of moisture. The amount of water from this source may be controlled by adequate insulation and ventilation as mentioned in a following section dealing with "The problem of condensation on walls and ceilings." Therefore, this factor will not be analyzed further here.

D. Design Outside Weather Conditions

The outside weather conditions to consider in the design of adequate and economical insulation value and ventilation rate are:

- 1-Design outside temperature
- 2-Design outside relative humidity
- 3-Design ground temperature
- 4-Design wind conditions
- 5-Design solar heat conditions
- 6-Design air infiltration

1. Design outside temperature

a. Method based on average regional temperature for the winter months.

In 1930, Kelley (58) presented a map of the United States divided into four climatic zones based upon United States Weather Bureau data and involving average temperatures for the months of January and February over a period of thirty years at one hundred selected stations. This map is for the purpose of farm-building design and is shown in Figure 1. The daily mean at 8 a.m. and the monthly mean for each zone are shown here (Table 35):

Table 35. Climatological zones of the United States *

Temperature zone	Daily mean at 8 a.m. (°F)	Mean monthly (°F)
Zone 1	5	11
Zone 2	17	22
Zone 3	27	30
Zone 4	36	above 32

* From data by Kelley (58)

Zones 1, 2 and 3 are the three northern ones and are those in which housing may be said to play an important part in environmental control in the winter time.

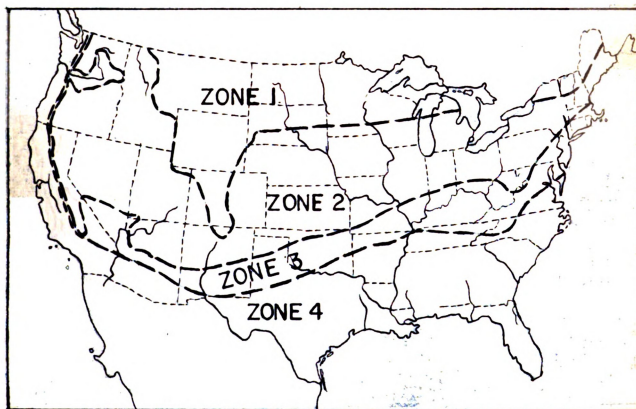
Strahan (28) suggested that the temperatures of these zones be established as standard values to be used in controlling the design of buildings to house dairy cows and poultry, except that the daily mean value at 8 p.m. for Zone 2 be reduced from 17°F to 16°F to maintain an equal temperature difference of 11°F between Zones 1 and 2 and between Zones 2 and 3. He recommends that the ventilation system be designed to function normally when these temperatures are obtained, and so constructed that ventilation may be restricted at lower outside temperatures and increased at higher.

More recently, Ashby, Miller et al (4) presented a variation of Kelley's climatological map (Figure 2). The boundary between Zones 2 and 3 was changed to the 35°F isotherm for average January temperature. The boundary between Zones 3 and 4 was chosen as the 50°F isotherm. The boundary between Zones 1 and 2 was adjusted to take account of the effect of high relative humidity in increasing heat loss by ventilation: "Where the average relative humidity in January is 70%, the zone boundary coincides with the 20°F isotherm. Where it is above 80% the boundary approaches the 25°F isotherm." (4)

Ashby, Miller et al (4) also add concerning these zones: "Minimum temperatures - about 40°F lower than the January average - occur once or twice a year in most of Zones 1 and 2. In Zones 3 and 4 the minimum is usually about 30°F lower than the January average except along the west coast. There are no abrupt changes at the zone lines, and except as influenced by altitude, the changes are noticeable only over considerable distances. The climate within all zones changes with altitude, the average temperature being about one degree lower for each 400-foot rise above the average elevation of the area."

FIGURE 1

CLIMATOLOGICAL ZONES OF THE UNITED STATES
(BASED ON AVERAGE TEMPERATURE FOR THE MONTHS
OF JANUARY AND FEBRUARY)*

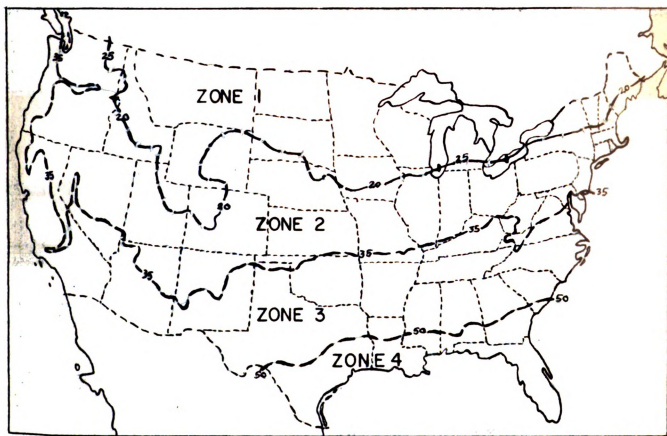


* By Kelley (58)

TEMPERATURE ZONE	DAILY MEAN AT 8 a.m. (°F)	MEAN MONTHLY (°F)
ZONE 1 _____	5° _____	11° _____
ZONE 2 _____	17° _____	22° _____
ZONE 3 _____	27° _____	30° _____
ZONE 4 _____	36° _____	Above 32° _____

FIGURE 2

CLIMATOLOGICAL ZONES OF THE UNITED STATES
(BASED ON AVERAGE TEMPERATURE AND RELATIVE
HUMIDITY FOR THE MONTH OF JANUARY)*



* By Ashby, Miller et al (4)

TEMPERATURE ZONE	AVERAGE JANUARY TEMPERATURE	MINIMUM JANUARY TEMPERATURE	
ZONE 1	0-20 and 0-25°F	40	(Approximate
ZONE 2	20-35 and 25-35°F	40	number of
ZONE 3	35-50°F	30	degrees lower
ZONE 4	50-70°F	30	than the Jan. average)

Kalbfleisch and White (57) presented in 1951 a similar climatological map for Canada (Figure 3). This map is based on the mean January temperature and the range for each temperature zone is as follows (Table 36):

Table 36. Climatological zones of Canada *

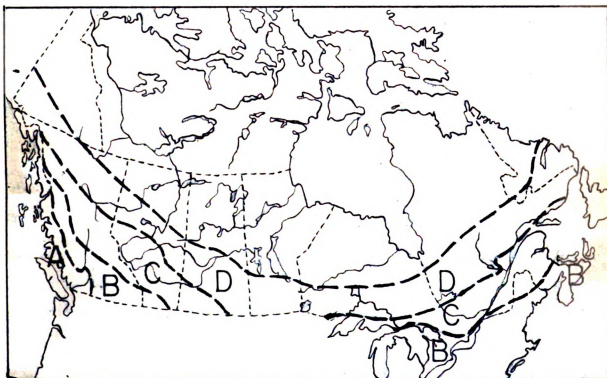
Temperature zone	Mean January temperature
Zone D	-5 to 5°F
Zone C	+5 to +15°F
Zone B	+15 to +25°F
Zone A	+25 to +35°F

* From data by Kalbfleisch and White (57)

Zone D covers the agricultural part of Canada located furthest north and represents the one where housing and ventilation design are the most critical.

FIGURE 3

CLIMATOLOGICAL ZONES OF CANADA
(BASED ON MEAN JANUARY TEMPERATURE)*



* By Kalbfleisch and White (57)

TEMPERATURE ZONE	MEAN JANUARY TEMP RANGE
ZONE A _____	+35 to +25 ° F
ZONE B _____	+25 to +15 ° F
ZONE C _____	+15 to +5 ° F
ZONE D _____	+5 to -5 ° F

b. Method based on average local temperature for the winter months.

One of the disadvantages of the climatological region method is that these regions are too big and cover localities of too widely different conditions. For this reason, the writer believes that this method could be improved somewhat by taking local average temperatures instead of climatological region average temperature. The Insulation Board Institute (49) states in this respect: "To make insulation recommendations for laying houses based upon average conditions for the several climatic zones is not safe because there is such a wide spread between the average and the temperatures which can and do occur at individual localities within any given zone. This does not hold true generally for other types of livestock shelters because larger animals make sufficient heat to cushion the effect of this comparatively high temperature variability."

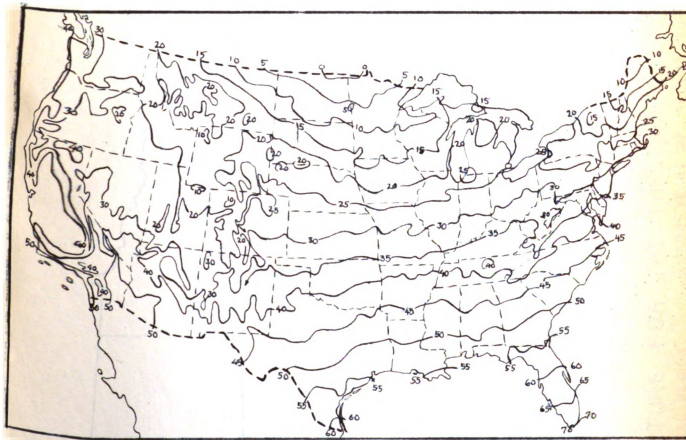
Climatological temperature data of greater local character are presented here (Figures 4 to 7) as an aid to understanding the local climate and determining the design outside temperature for animal shelters. It should be noticed however, that methods c and d are probably more recommended for use.

FIGURE 4

AVERAGE JANUARY TEMPERATURE (°F)

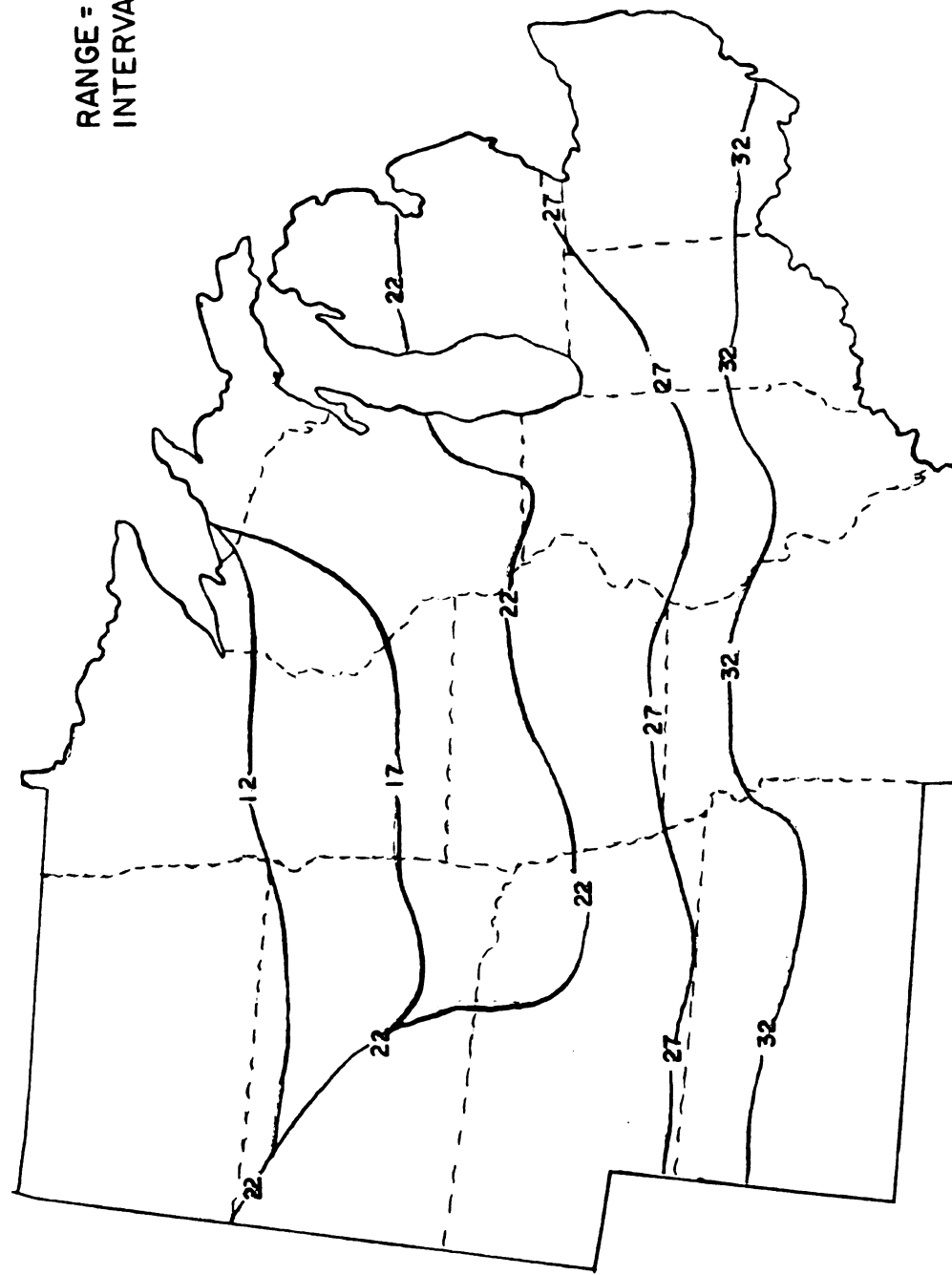
PERIOD 1899-1938 ♦

(BASED ON 200 FIRST ORDER WEATHER BUREAU STATIONS)



*From Climate and Man, U.S. Yearbook of Agriculture, 1941 (21)

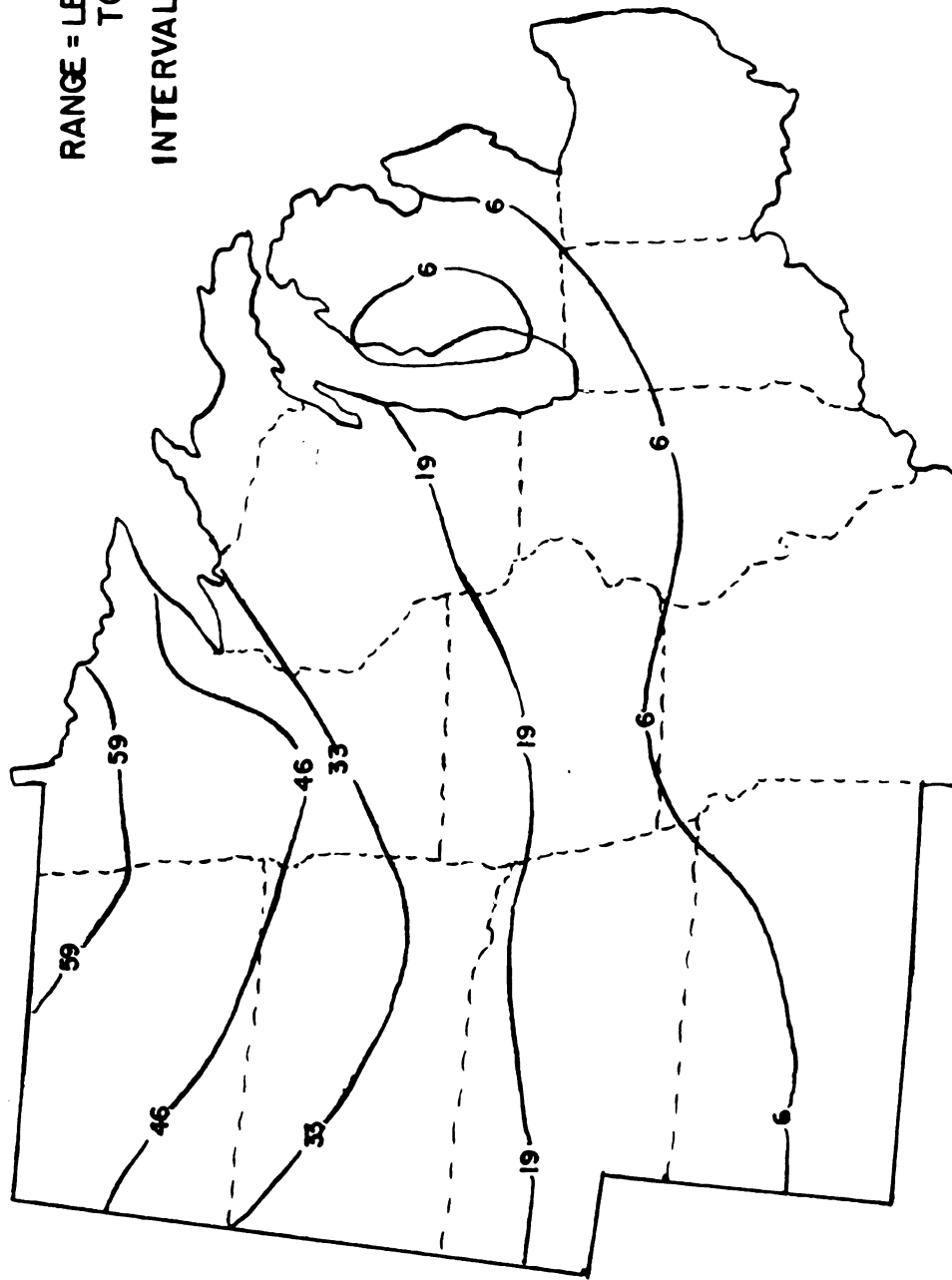
FIGURE 5



RANGE = 7.3 to 38°F
INTERVAL = 5°

MEAN TEMPERATURE FOR THE THREE WINTER MONTHS (DEC., JAN., FEB.)
(FOR YEARS OF RECORD TO 1950) FROM COWLES AND IRWIN (23)

NORTH CENTRAL REGION

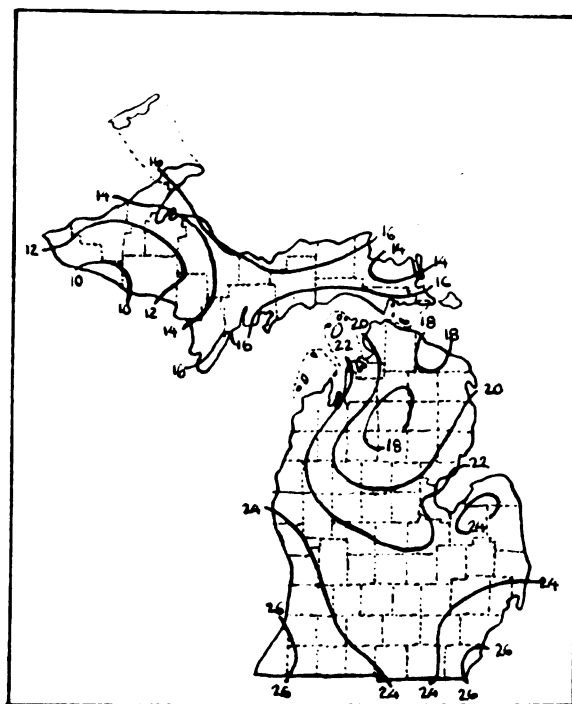


RANGE = LESS THAN 1
TO 61
INTERVAL = 13

FIGURE 6

MEAN NUMBER OF DAYS PER YEAR WITH MINIMUM TEMPERATURE BELOW 0°F
NORTH CENTRAL REGION (FOR YEARS OF RECORD TO 1950) FROM COWLES AND IRWIN(23)

FIGURE 7
AVERAGE JANUARY TEMPERATURE,
MICHIGAN *



* FROM "CLIMATE AND MAN", U.S. YEARBOOK OF AGRICULTURE, 1941 (21)

c. Method based on the lowest local temperature. In heating and air conditioning work, it is customary to base the design temperature on the lowest temperature on record for the given locality. However, this design temperature is taken at least 15°F above this absolute minimum (1). The thermal capacity of the building is assumed to be sufficient to carry over the extreme lows which ordinarily occur in the night hours.

The following table (Table 37) from Allen, Walker and James (1) illustrates the relation between the lowest temperature ever reported and the design temperature usually chosen for a few selected localities.

Table 37. Relationship between local lowest temperatures and the temperatures used for design of heating systems *

City	Average temperature from Oct. 1 to May 1 (°F)	Lowest temperature ever reported (°F)	Design temperature usually assumed (°F)	Difference between lowest design temperature (°F)
Atlanta	51.5	- 8	+ 10	18
Boston	38.1	-18	0	13
Chicago	36.4	-23	-10	14
Cleveland	37.2	-17	- 5	14
Detroit	35.8	-24	-10	14
New York	40.7	-14	0	14
Philadelphia	42.7	-11	0	11
Washington, D.C.	43.4	-15	0	15

* From data by Allen, Walker and James (1)

These authors report that: "since design temperatures are generally specified in 5°F intervals, it will be observed that these values check closely with the above rule."

Close (20) also found that the recommended design temperatures for heating and air conditioning checked closely with the average of the minimum temperatures that have occurred once each winter for a period from 1899 to 1938.

However, Barre and Sarnet (11) state that although the outdoor design temperature for heating is usually assumed as approximately 15°F higher than the extreme recorded lower temperature for the locality, "this guide cannot be applied uniformly".

Parker (62) discussing this method remarks that "the extreme low temperature for Blacksburg, Virginia was -27°F on December 30, 1917. This would give a temperature for design of -12°F if the above recommendation were followed. However, he states that the recommended residential design temperature for the locality is just slightly below 0°F , which makes a difference of approximately 25°F with the absolute minimum. As will be shown in the description of method d, Parker found the design temperature for poultry houses to be 5°F in this locality. This is just slightly more than five degrees higher than the recommended residential heating design temperature. He therefore recommends that: "in the absence of better information it is felt that five degrees above the recommended residential design temperature may be used as the outdoor minimum design temperature for poultry ventilation systems in areas near Blacksburg, Virginia." He adds that: "caution should be taken in applying this general analogy in other areas."

The following figure (Figure 8) published by the National Warm Air Heating and Air Conditioning Association (77) gives the outside design temperatures for heating estimates for many cities of both United States and Canada. It is believed that such a table will be valuable. For animal shelter application, these temperatures should however be raised by something like 5°F or 10°F . Figures 9, 10 and 11 are presented to supplement Figure 8.

FIGURE 8

OUTSIDE AIR TEMPERATURES FOR HEATING ESTIMATES

The following table lists the recommended outdoor winter temperatures to use as the basis for preparing heating estimates. These figures should be used unless the customer specifies the temperature upon which the estimate shall be made.

The figures given in this table are not the extreme minimum temperatures recorded in each locality and it is to be expected that lower temperatures will be encountered. However, very low temperatures usually exist only for short periods. It is thus unnecessary and uneconomical to design the heating system for the extreme minimum temperature. Change to meet local conditions when necessary.

State and City	Design Temp. F.	Janet	10	Hogewar	10	Jonestown	10	TENNESSEE	BR. COLUMBIA	Schrieber	20
Alabama		Florida	10	South Sea	10	Lake Florida	20	Chattanooga	15	St. Louis	10
Anniston	5	Rockford	10	Mario	20	Little Falls	20	Jackson	10	St. Catharines	10
Birmingham	5	Springfield	10	MINNESOTA		Lockport	5	Jackson City	10	Stratford	10
Chickasaw	5	INDIANA		Alexandria	25	New York	5	Knoxville	10	St. Thomas	10
Orlando	0	Elkhart	10	Duluth	20	Norwich	10	Memphis	10	Tulsa	10
Mobile	15	Brananville	0	Minneapolis	20	Ogdenburg	20	Nashville	10	Toronto	10
Montgomery	10	Fort Wayne	10	St. Cloud	25	Overpeck	10	St. Louis	10	Wilmington	10
Tombago	5	Indianapolis	10	St. Paul	20	Peoria	10	St. Louis	10	Windsor	10
ARIZONA		South Bend	10	MISSISSIPPI		Rockford	10	TEXAS		PRINCE EDWARD IS.	
Alhambra	15	Terre Haute	10	Biloxi	15	Savannah	10	Abilene	15	Charlotte	10
Flagstaff	10	IOWA		Columbus	15	Waterbury	10	Austin	10	Summerside	10
Glenn	10	Burlington	10	Greenburg	15	WINSTON	10	El Paso	10	AMUS	30
Hughes	10	Cedar Rapids	20	Jackson	15	NORTH CAROLINA		Galveston	25	Betherville	20
Phoenix	25	Clinton	10	Savannah	15	Asheville	5	Houston	25	Brome	20
Tucson	20	Council Bluffs	10	Natchez	15	Charlotte	10	San Antonio	20	Cowara	20
Yuma	20	Des Moines	15	Vicksburg	15	Greensboro	25	Waco	20	Drummondville	20
ARKANSAS		Dubuque	10	MISSOURI		St. Louis	15	UTAH		QUEBEC	
Fort Smith	5	Port Dodge	20	Columbia	10	Springfield	10	Logan	15	Amos	30
Hot Springs	5	Kearney	10	Nashville	10	WINSTON	10	Ogden	15	Betherville	20
Little Rock	5	Marionville	20	Kansas City	10	WINSTON	10	St. Louis	15	Brum	20
Pine Bluff	10	St. Louis	20	St. Louis	10	WINSTON	10	Braden	25	Cowara	20
Tyranah	10	Waterloo	20	St. Louis	10	WINSTON	10	Braden	25	Drummondville	20
CALIFORNIA		KANSAS		SPRINGFIELD	10	WINSTON	10	Braden	25	Drummondville	20
Bakersfield	10	Atchison	10	MONTANA		WINSTON	10	Braden	25	Drummondville	20
Fresno	25	Concordia	10	Anaconda	15	WINSTON	10	Braden	25	Drummondville	20
Los Angeles	20	Dodge City	10	Billings	25	WINSTON	10	Braden	25	Drummondville	20
Monterey	25	Lawrence	10	Butte	20	WINSTON	10	Braden	25	Drummondville	20
Pasadena	25	Malina	10	Butte	20	WINSTON	10	Braden	25	Drummondville	20
Sacramento	25	Tonka	10	Butte	20	WINSTON	10	Braden	25	Drummondville	20
San Diego	35	Wichita	5	Butte	20	WINSTON	10	Braden	25	Drummondville	20
San Francisco	35	KENTUCKY		Butte	20	WINSTON	10	Braden	25	Drummondville	20
San Jose	20	Bowling Green	0	NEBRASKA		WINSTON	10	Braden	25	Drummondville	20
COLORADO		Franklin	0	Grand Island	15	WINSTON	10	Braden	25	Drummondville	20
Boulder	15	Franklin	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Colorado Springs	25	Lexington	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Denver	15	Louisville	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Fort Collins	15	Owensboro	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Grand Junction	15	LOUISIANA		Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Laurens	25	Alexandria	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Pueblo	25	Baton Rouge	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
CONNECTICUT		New Orleans	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Bridgewater	0	Shreveport	10	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Bridgeport	0	MAINE		Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Bridgeport	0	Bar Harbor	15	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Bridgeport	0	Belfast	10	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Bridgeport	0	Lexington	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Bridgeport	0	Louisville	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Bridgeport	0	Owensboro	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
DELAWARE		LOUISIANA		Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Dover	5	Alexandria	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Milford	5	Baton Rouge	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Wilmington	5	New Orleans	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
DIST. OF COLUMBIA		Shreveport	10	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Washington	0	MAINE		Hastings	15	WINSTON	10	Braden	25	Drummondville	20
FLORIDA		Bar Harbor	15	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Fort Myers	25	Belfast	10	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Gainesville	25	Lexington	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Jacksonville	25	Louisville	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Key West	25	Owensboro	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Miami	45	LOUISIANA		Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Orlando	45	Alexandria	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Pensacola	25	Baton Rouge	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Pompano Beach	25	New Orleans	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Tampa	25	Shreveport	10	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
GEORGIA		MAINE		Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Athens	10	Bar Harbor	15	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Atlanta	10	Belfast	10	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Augusta	15	Lexington	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Columbus	15	Louisville	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Macon	15	Owensboro	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Savannah	15	LOUISIANA		Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Way Cross	20	Alexandria	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
ID. ARO		Baton Rouge	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Merley	0	New Orleans	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Lewiston	0	Shreveport	10	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Pocatello	10	MAINE		Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Twin Falls	10	Bar Harbor	15	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
ILL. INOIS		Belfast	10	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Aurora	10	Lexington	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Bloomington	10	Louisville	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chicago	10	Owensboro	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Springfield	10	LOUISIANA		Hastings	15	WINSTON	10	Braden	25	Drummondville	20
St. Louis	10	Alexandria	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Decatur	10	Baton Rouge	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Elgin	10	New Orleans	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
INDIANA		Shreveport	10	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Alpena	10	MAINE		Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Ann Arbor	10	Bar Harbor	15	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Big Rapids	15	Belfast	10	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chickasaw	15	Lexington	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Elkhart	10	Louisville	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Flint	10	Owensboro	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Grand Rapids	15	LOUISIANA		Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Lansing	15	Alexandria	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Livonia	15	Baton Rouge	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Marquette	15	New Orleans	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Port Huron	15	Shreveport	10	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
NEW JERSEY		MAINE		Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Albany	5	Bar Harbor	15	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Albany Park	5	Belfast	10	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Atlantic City	5	Lexington	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Bayonne	5	Louisville	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Belvidere	10	Owensboro	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Berkeley	10	LOUISIANA		Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Boston	10	Alexandria	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Burr	10	Baton Rouge	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Camden	10	New Orleans	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Canfield	10	Shreveport	10	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	MAINE		Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	Bar Harbor	15	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	Belfast	10	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	Lexington	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	Louisville	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	Owensboro	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	LOUISIANA		Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	Alexandria	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	Baton Rouge	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	New Orleans	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	Shreveport	10	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	MAINE		Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	Bar Harbor	15	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	Belfast	10	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	Lexington	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	Louisville	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	Owensboro	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	LOUISIANA		Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	Alexandria	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	Baton Rouge	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	New Orleans	20	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	Shreveport	10	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	MAINE		Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	Bar Harbor	15	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	Belfast	10	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	Lexington	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	Louisville	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	Owensboro	0	Hastings	15	WINSTON	10	Braden	25	Drummondville	20
Chapel Hill	10	LOUISIANA		Hast							

1

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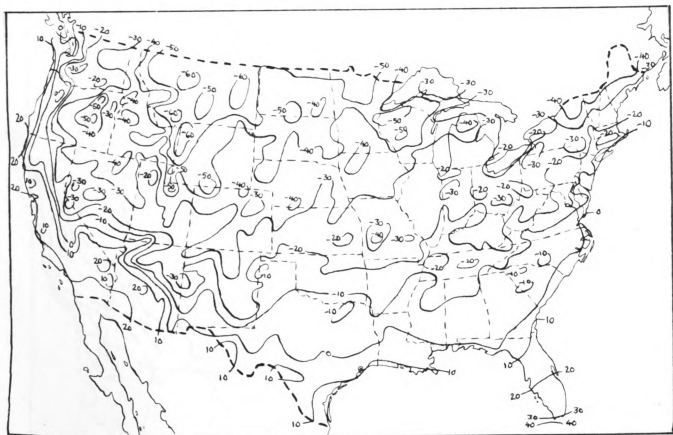
7

FIGURE 9

LOWEST TEMPERATURES EVER OBSERVED (°F)

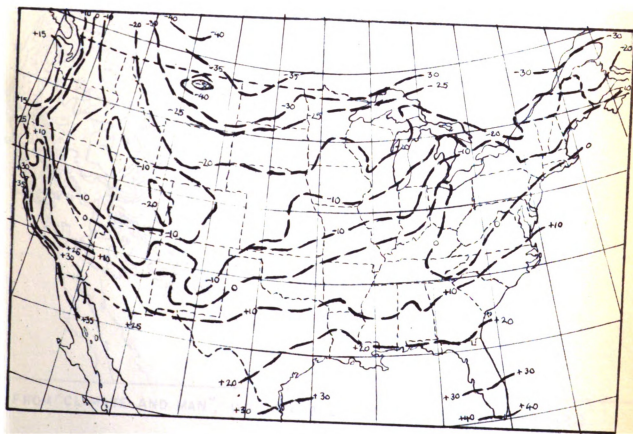
PERIOD 1899-1938 *

(BASED ON 200 FIRST ORDER WEATHER BUREAU STATIONS)



* From Climate and Man, U.S. Yearbook of Agriculture, 1941 (21)

FIGURE 10

ISOTHERMS OF WINTER OUTDOOR
DESIGN TEMPERATURE •

* From Heating Ventilating Air Conditioning Guide 1950 (5)

FIGURE 11

AVERAGE ANNUAL MINIMUM TEMPERATURE (°F)

(BASED ON 200 FIRST ORDER WEATHER BUREAU STATIONS)

PERIOD 1899—1938*



* FROM "CLIMATE AND MAN", U.S. YEARBOOK OF AGRICULTURE, 1941 (21)

d. Method of local "expected failures". In 1952, Parker (82) recommended a new method called "expected failure method". This method consists of assuming a design outside temperature and analyzing the local weather records over a period of as many years as possible, to determine the number of failures that would have occurred during that period. A failure is counted

- (a) When the temperature remains in a range of 2 degrees below the assumed design temperature for 10 hours or more
- (b) When the temperature remains in a range of more than 2 degrees below the assumed design temperature for 3 hours or more.

The same author (82) made an analysis of twelve year records for Blacksburg, Virginia, and transcribed minimum temperatures and the amount of time that the temperatures remained below -5, 0, 5, 10, 15 and 20°F. From the data it appeared that approximately 5°F would be a satisfactory outdoor minimum design temperatures. Applying the above definition of a failure, the results showed eight failures in twelve years or an average of two-thirds failure per year. Since several of the failures were for 2°, 3° and 4°F, these failures are probably not very serious, he concluded that a minimum outdoor design temperature of 5°F is considered satisfactory for Blacksburg, Virginia.

Parker (82) also states: "It was noticed that 5°F is slightly more than five degrees higher than the recommended residential heating design temperature of Blacksburg, Virginia. Therefore, in the absence of better information, it is felt that five degrees above the recommended residential design temperature may be used as the outdoor minimum design temperature for poultry ventilation systems in areas near Blacksburg, Virginia. Caution should be taken in applying this general analogy in other areas."

The writer using Parker's method analyzed winter temperatures for East Lansing for periods from 1949 to 1953. The following table (Table 38) shows the number of failures which have occurred by using design temperature of -10, -5, -2 and 0°F.

Table 38. Computation of the outside design air temperature for East Lansing

Outside design temperature assumed	Winter	Number of failures per winter	Total number of failures for the four winters	Average number of failures per winter
-10°F*	1949-1950	No failure		
	1950-1951	No failure		
	1951-1952	No failure		
	1952-1953	No failure		
			0 failure	0 failure
- 5°F**	1949-1950	No failure		
	1950-1951	2 failures		
	1951-1952	No failure		
	1952-1953	No failure		
			2 failures	1/2 failure
- 2°F***	1949-1950	No failure		
	1950-1951	2 failures		
	1951-1952	No failure		
	1952-1953	No failure		
			2 failures	1/2 failure
0°F****	1949-1950	2 failures		
	1950-1951	4 failures		
	1951-1952	2 failures		
	1952-1953	No failure		
			8 failures	2 failures

* A failure was counted for each 10 hours the temperature remained between -10 and -12 or each 3 hours it remained below -12

** A failure was counted for each 10 hours the temperature remained between -5 and -7 or each 3 hours it remained below -7

*** A failure was counted for each 10 hours the temperature remained between -2 and -4 or each 3 hours it remained below -4

**** A failure was counted for each 10 hours the temperature remained between 0 and -2 or each 3 hours it remained below -2

Consideration of the above table (Table 38) shows that 0°F is probably a good outside design temperature for Lansing. Since Lansing outside design air temperature for heating estimates is -10°F (Figure 8), it seems to the writer that the design outside temperature for animal shelters for most localities can be safely assumed from 5 to 10 degrees above the recommended residential design local temperature for heating as presented in Figure 8.

2. Design outside relative humidity

Strahan (99) recommends the use of an outdoor relative humidity of 100% since "the most difficult conditions for removal of moisture will be when outside air is saturated (100% R.H.) and when under this condition it is necessary to hold the inside relative humidity at 85% to avoid condensation." Parker (82) also recommends the use of an outdoor relative humidity of 100% for design during the critical period "since a relative humidity of 100% exists during rainy or foggy periods and these conditions may last for several days." He believes that this value should be applicable to the humid east.

Barre and Sammet (11) used an outdoor relative humidity of 80% in the drawing of their air flow chart. Oliver (78) also used an outdoor relative humidity of 70% and 80% in his calculations which cover the outdoor temperature range from 0 to 40°F. It looks like each had in mind the estimation of the probable performance under average outdoor weather conditions, rather than computation for the purpose of design under extreme conditions.

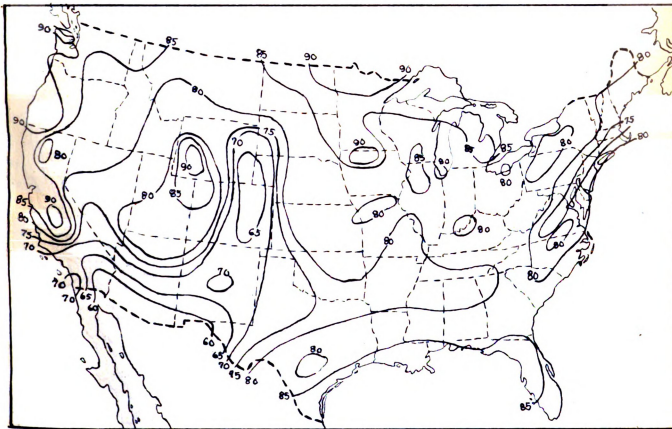
The writer believes that for critical design analysis 100% relative humidity should be used or at least a value of 90% or 95%. For general performance analysis a value of 80% is satisfactory in most cases.

Figures 12 to 15 are presented to give an idea of local January average relative humidity at different times of the day.

FIGURE 12

AVERAGE RELATIVE HUMIDITY~ 8 a.m.(E.S.T.)
JANUARY

(BASED ON 200 FIRST ORDER WEATHER BUREAU STATIONS)
PERIOD 1899-1938 *

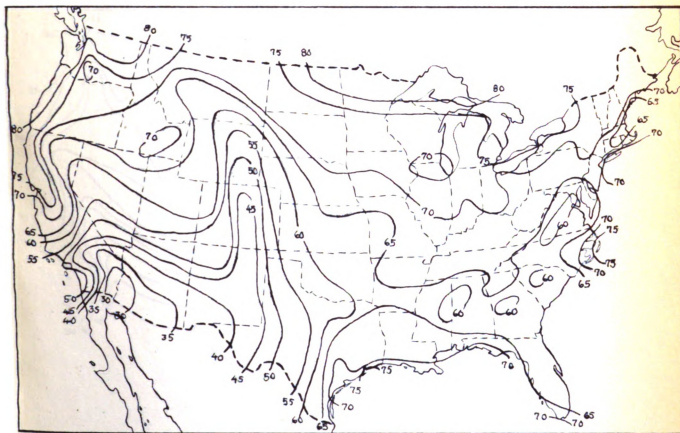


*From Climate and Man, U.S. Yearbook of Agriculture, 1941 (21)

FIGURE 13

AVERAGE RELATIVE HUMIDITY—Local noon
JANUARY

(BASED ON 200 FIRST ORDER WEATHER BUREAU STATIONS)
PERIOD 1899—1938 *

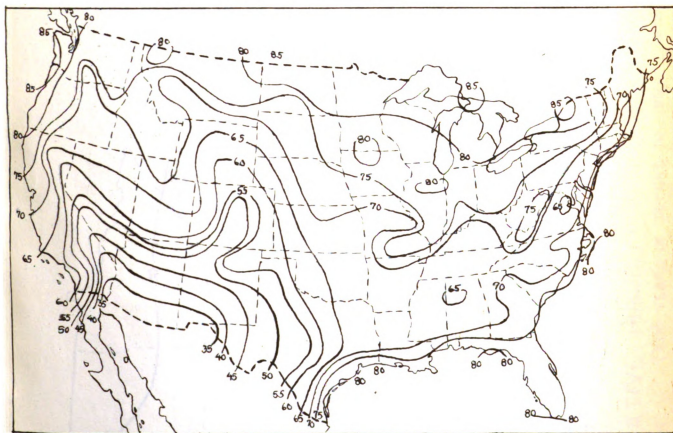


*From 'Climate and Man', U.S. Yearbook of Agriculture, 1941 (21)

FIGURE 14

AVERAGE RELATIVE HUMIDITY — 8 P.M. (E.S.T.)
JANUARY

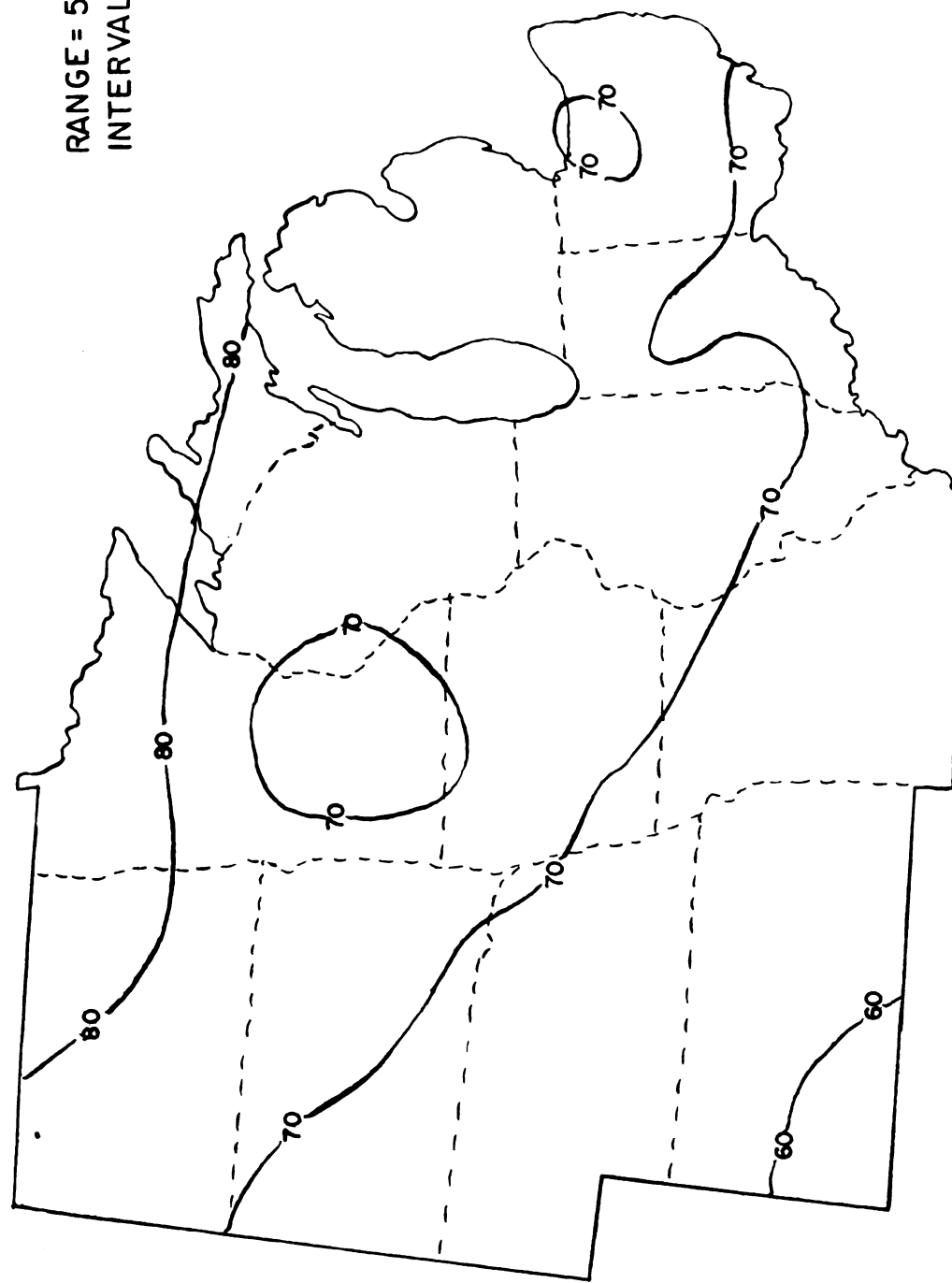
(BASED ON 200 FIRST ORDER WEATHER BUREAU STATIONS)
PERIOD 1899 — 1938*



* FROM "CLIMATE AND MAN", U.S. YEARBOOK OF AGRICULTURE, 1941 (21)

FIGURE 15

RANGE = 57 TO 81%
INTERVAL = 10%



MEAN RELATIVE HUMIDITY FOR JANUARY

NORTH CENTRAL REGION (FOR YEARS OF RECORD TO 1950) FROM COWLES AND IRWIN(23)

3. Design ground temperature

The moisture content, consistency and heat capacity of soil greatly affect the ground temperature at various depths but, according to Allen, Walker and James (1), it can generally be assumed that the ground temperature under a basement floor will average about 50°F in most localities.

If such a temperature is true for poultry house floors, it can be seen that for house temperatures of below 50°F, there would be a heat gain from the soil. Moreover, Bates (12) obtained the following average temperatures from his one-week test on a concrete floor poultry house. These temperatures were taken five feet from the back of the house.

1-Outdoor temperature	25°F
2-Temperature at the surface of the concrete floor	30°F
3-Temperature in litter two inches above the floor	39°F
4-Air temperature immediately above surface of litter	34°F

The earth temperature under the concrete floor therefore seems to be colder than the air temperature and to be a kind of intermediate value between the outdoor temperature and the temperature of the air in the poultry house. This would mean that the soil would be receiving some heat from the litter instead of giving some to the house.

The writer believes that some research is required to find the ground temperature at different depths around the foundations and under the floor or litter of animal shelters. The results of such research would show if there is a loss or a gain of heat and would perhaps make possible the determination of its value for purpose of design or performance analysis.

4. Design wind conditions (vs Infiltration and exterior surface conductance)

The heat losses are affected both by the outside temperature and by the wind velocity. The relative importance of these two factors varies with the design of the building.

Allen, Walker and James (1) state: "If a building were, say, entirely of brick, with no windows, the effect of the wind would be merely to increase the exterior surface conductance, which is a minor factor, and the outside temperature would control the heat loss. On the other hand, a building with a large number of leaky windows would have a much larger heat loss with high wind velocity. Here the outside temperature would also be important for two reasons. It affects the heat loss through the walls and glass, and it affects the heat loss per cubic foot of infiltration."

The authors recommend the choice of a local average wind velocity (December, January and February) for the purpose of calculating the infiltration loss.

Such averages are presented in the following table (Table 39) for a few selected localities (1). Direction of prevailing wind is also given.

Table 39. Average wind velocity and direction of prevailing wind for a few selected localities*

City	Average wind velocity for Dec. Jan. and Feb. (mph)	Direction of prevailing wind for Dec. Jan. and Feb.
Atlanta	11.4	N.W.
Boston	11.2	W
Chicago	12.5	W
Cleveland	15.0	S.W.
Detroit	12.7	S.W.
New York	16.7	N.W.
Philadelphia	11.0	N.W.
Washington, D.C.	7.9	N.W.

* From data by Allen, Walker and James (1)

As can be seen, the average wind velocity is in general above 10 mph. This probably explains why, for determining the surface conductance, it is sufficiently accurate to assume a velocity of 15 mph.

Since there is actually an increased heat loss on the windward side of a building, the question may arise concerning the opportunity of adding an allowance for this factor. Allen, Walker and James (1) answer the question by saying: "From years of none-too-reliable experience there has grown a practice of adding an allowance to the calculated heat loss on the sides of the building exposed to the prevailing winter winds. This exposure allowance, when used, is ordinarily 15%. It is difficult to defend this practice, in as much as the transmission losses and the infiltration are already figured for a 15-mile, or higher, wind velocity. Exposure allowances are arbitrary safety factors, which were more necessary in former years than they are today when methods of calculations are more exact, and the present tendency is away from the practice."

5. Design solar heat conditions

The analysis of solar heat as a secondary source of heat has been presented in a previous section and will not be repeated here.

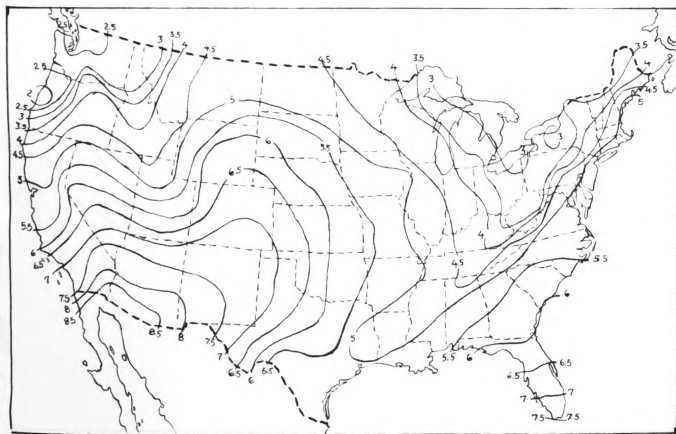
However, the following figures (Figures 16 to 19) are presented in order to give the designer a better understanding of the solar conditions characterizing the local area he is covering.

FIGURE 16

AVERAGE NUMBER OF HOURS OF SUNSHINE, DAILY,
WINTER (DEC. - FEB.)

(BASED ON 200 FIRST ORDER WEATHER BUREAU STATIONS)

PERIOD 1899 - 1938*

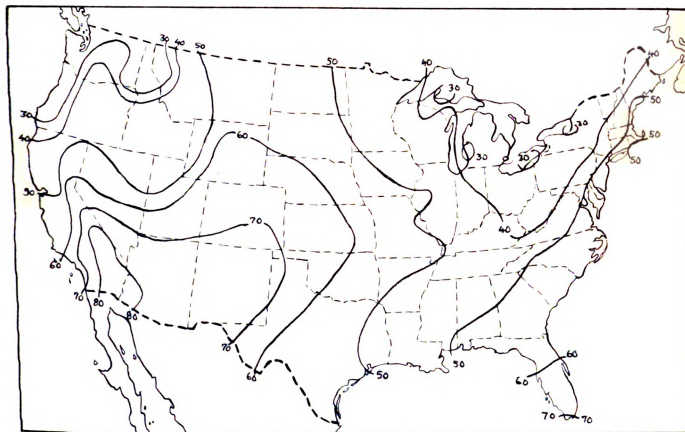


* FROM "CLIMATE AND MAN", U.S. YEARBOOK OF AGRICULTURE, 1941 (21)

FIGURE 17

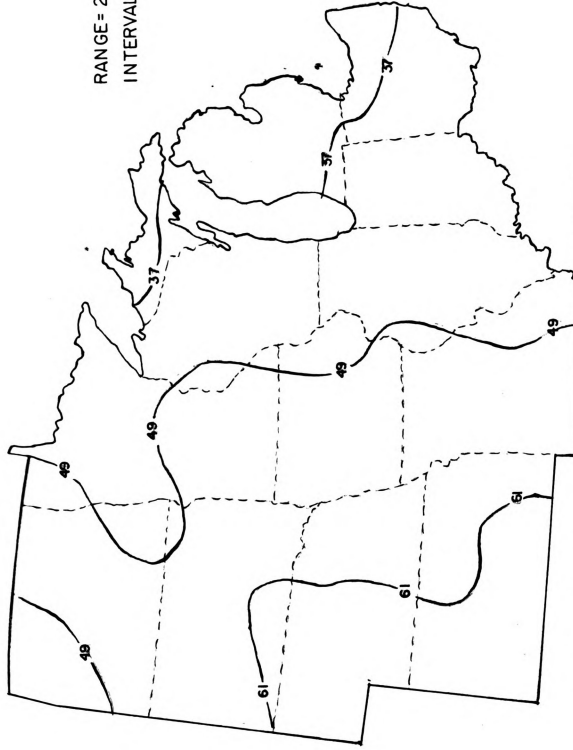
PERCENTAGE OF POSSIBLE SUNSHINE,
WINTER (DEC.-FEB.)

(BASED ON 200 FIRST ORDER WEATHER BUREAU STATIONS)
PERIOD 1899-1938 *



* FROM "CLIMATE AND MAN", U.S. YEARBOOK OF AGRICULTURE, 1941 (21)

FIGURE 18

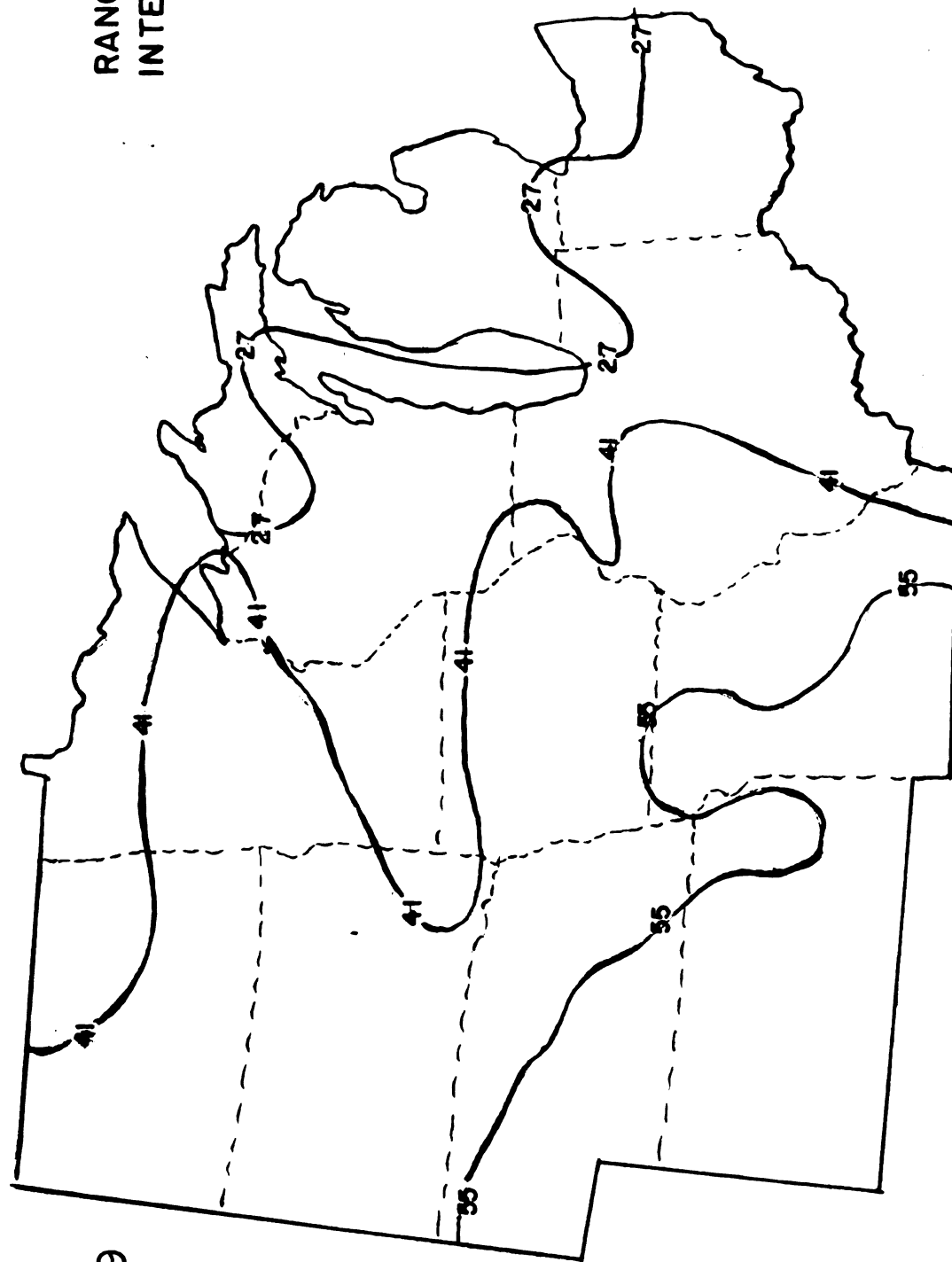


PERCENT POSSIBLE SUNSHINE FOR JANUARY

NORTH CENTRAL REGION (FOR YEARS OF RECORD TO 1950) FROM COWLES AND IRWIN (23)

RANGE = 26 TO 67%
INTERVAL = 12 %

FIGURE 19



DAYS CLEAR DURING THE FIVE COLD MONTHS (NOV. TO MARCH INCL.)

NORTH CENTRAL REGION (FOR YEARS OF RECORD TO 1950) FROM COWLES AND IRWIN(23)

6. Design air infiltration through cracks

Design air infiltration through cracks is affected by wind and for this reason is classified in this section. Two methods are usually used to find the air infiltration value. These methods will be described here:

a. Air change method. This method of computing the amount of infiltration through the cracks around windows and doors and through construction joints consists of assuming that the volume of air in the room or building is renewed a certain number of times per hour. The number of air changes usually assumed are given in the following table (Table 40)

Table 40. Air changes by infiltration and exfiltration in different kinds of rooms*

Description of the room	Changes per hour
1-Rooms with window on one or two sides	1 - $1\frac{1}{2}$
2-Rooms with windows on three or four sides	2

* From data by Allen, Walker and Jones (1)

This simple method is sufficiently accurate in many cases.

b. Length of crack method. A second method which seems preferable (1) is to compute the length of the crack around the windows and to assume an amount of air leakage per foot of crack depending on the window construction, based on tests of window leakage.

The following values (Table 41) abstracted from the A.S.H. & V.E. Guide, 1945 (25) are based on tests by various investigators but are approximately 20% less than the actual test results, to allow for the probable reduction in leakage in an actual building due to the building up of pressure within the building.

Table 41. Air infiltration due to windows of various types *

Type of window	Wind velocity		
	5 mph	15 mph	30 mph
1-Double-hung, wood-sash windows (per ft of crack around sash and meeting)			
1-Window unlocked, without weather strips	6.6	39.3**	103.7
2-With weather strips	4.3	23.6	63.4
2-Double-hung, metal sash windows (per ft of crack around sash and meeting rail)			
1-No weather strips, window locked	20	70	154
2-No weather strips, window unlocked	20	74	170
3-Weather strips, window unlocked	6	32	76
3-Rolled-section steel windows (per ft of crack around ventilating sash)			
1-Industrial type, 1/16" crack	52	176	372
2-Residential type, 1/32" crack	14	52	128
3-Heavy casement type, 1/32" crack	8	38	96

* From data in the A.S.H. & V.E. Guide 1945 (25)

* * According to Goodman (37) this figure also supplies to well fitted doors

Since air is flowing inward through some of the windows and outward through others located on the opposite side of the buildings, the generally accepted method for rooms having three or four exposed walls is to take the amount of cracks in the wall having the greatest number, but in no case less than half of the total cracks.

Allen, Walker and James (1) give the following considerations concerning storm sashes: "Storm sashes are often not very tight-fitting and therefore, may not greatly reduce the infiltration if the original window itself is well fitted or is weather-stripped. But if the window is poorly fitted and the storm sash is tight, the infiltration is much reduced by the latter. It is impracticable to set up any definite standards, but tests indicate that:

1-For a loose window and fastened storm sash, the reduction of infiltration due to the storm sash may be 60%

2-For a fairly tight window, not weather-stripped and a fastened storm sash = the reduction may be 20%."

It should be noted that this method is empirical and not very accurate. Moreover, the conditions are so varied that no exact method could be devised.

It is a good practice to check the amount of infiltration obtained by the present method with the figures from the air change method. According to Allen, Walker and James (1) the infiltration finally assumed should not be less than the equivalent of one-half of an air change per hour.

B. Methods of Standardization of Poultry House Ventilation

Because of the great relation existing between insulation of houses and the efficiency of ventilation systems, certain methods have been proposed as a means of comparison in measuring the performance of ventilation systems in houses of wide difference in design.

1. The AC method

This method was proposed by Strahan in 1940 (99): "If precise observations could be made on performance in houses with AC characteristics ranging between 0.6 and 4.5, and if these observations could be studied and results analyzed, fairly definite relations between the value AC and actual performance should show up that would lead to justifiable recommendations concerning design and management of houses for different climatic conditions. Would it not be possible to develop some sort of cooperative investigation by means of which data from different sections could be correlated in some appropriate central office, and would it not be reasonable to expect that from such a study there would emerge design recommendations that could be generally accepted because they would be rationally based?"

The AC value of a building is obtained by multiplying the overall heat transfer coefficient of the building (C or $U_{aver.}$) by the total exposed area per hen (A). The unit is therefore:

Btu lost from the house/Hr, °F, hen

Its range as mentioned above by Strahan is usually between 0.6 and 4.5 for poultry houses. Considering A as the exposed area per 100 hens would make the AC value vary from approximately 60 to 450.

2. The exposure ratio method

Barre and Sammet (10) introduced in 1949 a modification of the AC value, called the "Exposure ratio." The exposure ratio is obtained by dividing

the AC value by the total net sensible heat available in the house. If A is considered as the exposed area per hen and $U_{aver.}$ is replaced by C, the equation is:

$$\text{Exposure ratio} = \frac{A U_{aver.}}{Q_s} = \frac{AC}{Q_s}$$

The unit is therefore:

Btu lost from the house/"r, of, bird, Btu of sensible heat

Since the AC value is already rather small, being usually in the range of 0.5 to 4.5, dividing it by the total net sensible heat makes it fractional and of an undesirable character for use in computations (0.005 to 0.25 approximately).

Considering A as the exposed area per 100 hens would make the value of the exposure ratio vary from approximately 0.5 to 25 which might be of some advantage as compared to the small range (0.6 to 4.5) or rather wide range (60 to 450) of the AC value.

The AC method is the one which has been mostly used up to this day, probably because of its fixed character for a given house, as compared to the exposure ratio which may take any possible value according to the amount of sensible heat produced in the house.

F. Poultry House Ventilation Formulas

1. Rate of ventilation required for temperature control

a. Strahan formula. In 1940, Strahan (99) presented the first simplified formula for determining the rate of ventilation required to control the temperature of poultry houses

$$V_s = \frac{52}{D} (Q_s - ACD)$$

where V_s = Air flow rate required per bird to control temperature (cfh/bird)

Q_s = Net amount of sensible heat produced in the house (Btu/hr, bird)

$D = \Delta t$ = Difference between inside and outside temperature ($^{\circ}\text{F}$)

$C = U_{\text{aver}}$, = Average heat transfer coefficient of the building (Btu/Hr, sq. ft., $^{\circ}\text{F}$)

A = Total exposed area per bird (Sq. ft/Bird)

$52^* = \frac{v}{0.24} = \frac{12.48}{0.24}$ = Number of cubic feet of air raised 1°F by one Btu at slightly above freezing temperatures(33°F and 80% R.H.)

v = Specific volume of the moist inside air = 12.48 cu.ft., at 33°F and 80% R.H. (cu.ft/lb dry air)

0.24 = Specific heat of air at constant pressure

This formula was derived from the following heat balance equation which neglects the secondary sources of heat losses in poultry houses:

Net sensible heat produced in the poultry house	=	Heat loss due to the building	+	Heat loss due to the air change (Ventilation and Infiltration)
---	---	----------------------------------	---	---

$$Q_s = ACD + \frac{V_s D}{52}$$

* Strahan was using 50 instead of 52, which corresponds to about 17°F .

A check of the units of each term on the right of the last equation gives:

$$ACD = \frac{\text{ft}^2}{\text{hen}} \times \frac{\text{Btu}}{\text{hr } ^\circ\text{F ft}^2} \times ^\circ\text{F} = \text{Btu/hr, hen}$$

$$\frac{V_s D}{52} = \frac{\frac{\text{ft}^3}{\text{hr, hen}} \times ^\circ\text{F}}{\frac{\text{ft}^3 \text{ } ^\circ\text{F}}{\text{Btu}}} = \frac{\text{ft}^3 \text{ } ^\circ\text{F}}{\text{hr, hen}} \times \frac{\text{Btu}}{\text{ft}^3 \text{ } ^\circ\text{F}} = \text{Btu/hr, hen}$$

which is in agreement with the left member Q_s , also Btu/hr, hen.

b. Barre and Sarnet formula. In 1949, Barre and Sarnet (10) presented a different equation than Strahan's formula for the same purpose:

$$V_s = \frac{v}{0.24} \left(\frac{1}{\Delta t} - \frac{AU_{\text{aver.}}}{Q_s} \right) Q_s$$

where V_s = Air flow rate required per bird to control temperature
(cfh/bird)

Q_s = Net amount of sensible heat produced in the house (Btu/hr, bird)

$\Delta t = D$ = Difference between inside and outside air temperature ($^\circ\text{F}$)

$U_{\text{aver}} = C$ = Average heat transfer coefficient of the building
(Btu/Hr, sq. ft., $^\circ\text{F}$)

v = Specific volume of one pound of dry air (Cu. ft/Lb dry air)

0.24 = Specific heat of air at constant pressure (Btu/Lb dry air)

A = Total exposed area per bird (sq.ft/bird)

By analysis and comparison of the two formulas, the writer found this second formula to reduce to Strahan formula if $U_{\text{aver.}}$ is replaced by C ,

Δt by D , and $\frac{v}{0.24}$ by 52 :

$$V_s = 52 \left(\frac{1}{D} - \frac{AC}{Q_s} \right) Q_s = 52 \left(\frac{Q_s}{D} - AC \right)$$

$$V_s = 52 \frac{Q_s - ACD}{D} = \frac{52}{D} (Q_s - ACD) \quad (\text{Strahan formula})$$

It should be noted here that the Barre and Sammet formula is just a complication of Strahan's, the latter being much easier to remember. If one keeps in mind that the value of $\frac{v}{0.24}$ may vary, although usually around 52 for poultry houses, the writer believes that Strahan's formula is preferable. Where greater accuracy is wanted, Strahan's formula can be slightly changed to take the form:

$$V_s = \frac{v}{0.24} (Q_s - ACD)$$

As a numerical check, consider the following example:

Numerical check

Given: Inside temperature = 35°F

Inside relative humidity = 80%

Outside temperature = 0°F (temperature differential = 35°F)

AC value of the house = 0.75

Wanted: The amount of ventilation allowable for temperature control

Solution:

A) Use of Strahan formula:

$$V_s = 52 \frac{Q_s - ACD}{D} = 52 \times \frac{43 - 0.75 \times 35}{35} = 52 \times \frac{43 - 26.25}{35}$$

$$V_s = \frac{52 \times 16.75}{35} = 24.900 \text{ cfh/hen (or 0.422 cfm/hen)}$$

B) Use of the modified Barre and Sammet formula:

$$\begin{aligned} V_s &= 52 \left(\frac{1}{\Delta t} - \frac{AC}{Q_s} \right) Q_s = 52 \left(\frac{1}{35} - \frac{0.75}{43} \right) 43 \\ &= 52 \left(\frac{43}{35} - 0.75 \right) = 52 (1.229 - 0.75) \\ &= 52 \times 0.479 = 24.900 \text{ cfh/hen (or 0.422 fhm/hen)} \end{aligned}$$

3) Use of the unmodified Barre and Sammet formula:

$$\begin{aligned}
 V_s &= \frac{V}{0.24} \left(\frac{1}{\Delta t} - \frac{A_0}{T_s} \right) \cdot s = \frac{12.53}{0.24} \left(\frac{1}{35} - \frac{0.75}{13} \right) \cdot 43 \\
 &= 52.21 \left(\frac{1}{35} - \frac{0.75}{13} \right) \cdot 43 = 52.21 \left(\frac{43}{35} - 0.75 \right) \\
 &= 52.21 (1.229 - 0.75) = 52.21 \times 0.479 = 25.009 \text{ cfh/hen (or 0.415 cfm/hen)}
 \end{aligned}$$

The difference obtained by the use of the unmodified Barre and Sammet formula as compared to Strahan formula is within 4% error (refer to page 180) for the range of inside temperatures from 15 to 50°F. For purpose of design, this simplification is therefore very satisfactory.

2. Rate of ventilation required for moisture removal

a. Standard formula. The standard ventilation formula for moisture removal is

$$V_L = \frac{\text{Moisture produced in the house, per hen}}{v_{iact} \times \text{Moisture carrying capacity of one pound of outside (or incoming air)}} = \text{cfh/hen}$$

If the moisture production is computed in grains per hour per hen and the moisture content of inside and outside air is also considered in grains, the above equation becomes:

$$\begin{aligned}
 V_L &= \frac{W_T (\text{grains/hr, hen})}{\left(\frac{u_i w_{s_i} - u_o w_{s_o}}{v_{iact}} \right)} = \frac{W_T (\text{grains/hr, hen})}{\left(\frac{w_i - w_o}{v_{iact}} \right)} \\
 &= \frac{W_T (\text{grains/hr, hen}) \times v_{iact}}{(w_i - w_o)} \quad \text{Formula (1)}
 \end{aligned}$$

If the moisture production is computed in pounds per hour per one hundred hens and the moisture content of inside and outside air is also considered in pounds, the equation becomes:

$$V_L = \frac{W_T (\text{lbs/hr, 100 hens})}{100 \left(\frac{u_i w_{s_i} - u_o w_{s_o}}{v_{iact}} \right)} = \frac{W_T (\text{lbs/hr, 100 hens})}{\left(\frac{100 u_i w_{s_i} - 100 u_o w_{s_o}}{v_{iact}} \right)}$$

$$= \frac{W_T (\text{lbs/hr, 100 hens}) \times v_{i_{\text{act}}}}{(100 u_i W_{s_i} - 100 u_o W_{s_o})} = \frac{W_T (\text{lbs/hr, 100 hens})}{100 \times \left(\frac{W_i - W_o}{v_{i_{\text{act}}}} \right)}$$

$$= \frac{W_T (\text{lbs/hr, 100 hens})}{\left(\frac{100 W_i - 100 W_o}{v_{i_{\text{act}}}} \right)} = \frac{W_T (\text{lbs/hr, 100 hens}) \times v_{i_{\text{act}}}}{(100 W_i - 100 W_o)} = \text{cfm/hen}$$

Formula (2)

In these formulas, the meaning of the symbols are as follows:

- V_L = Ventilation rate for moisture removal (cfh/hen)
- W_T = Moisture production in the house (grains/hr, hen) or (lbs/hr, 100 hens)
- u_i = Degree of saturation of inside (or outgoing) air (For practical purpose usually may be assumed equal to the relative humidity)
- u_o = Degree of saturation of outside (or incoming) air
- W_{s_i} = Water content of inside (or outgoing) air when saturated (grains/lb dry air) or (lb/lb of dry air), consistent with M unit
- W_{s_o} = Water content of outside (or incoming) air when saturated (grains/lb dry air) or (lb/lb dry air), consistent with M unit
- W_i = Water content of inside (or outgoing) air at the given temperature and relative humidity, consistent with M unit
- W_o = Water content of inside (or incoming) air at the given temperature and relative humidity, consistent with M unit
- $v_{i_{\text{act}}}$ = Specific volume of the moist inside air (cu.ft/lb dry air)

Numerical check

As a numerical check of these formulas let it be required to remove 114 grains of moisture per hour per hen or $\frac{114 \times 100}{7000} = 1.63$ pound per hour per 100 hens which is the total expired and excreted water production of four-pound hens according to Barre and Sarnet (11). Inside and outside conditions of 35 and 0°F and inside and outside relative humidity of 80 and 100% are assumed.

$$\begin{aligned} \text{Formula (1)} \quad V &= \frac{M (\text{grains/hr, hen}) \times v_1}{(W_1 - W_0)} \\ &= \frac{114 \times 12.53}{(23.86 \times 5.5)} = 77.7 \text{ cfh/hen (or 1.295 cfm/hen)} \end{aligned}$$

$$\begin{aligned} \text{Formula (2)} \quad V &= \frac{M (\text{lb/hr, 100 hens}) \times v_1}{(100 W_1 - 100 W_0)} \\ &= \frac{1.63 \times 12.53}{(0.34096 - 0.07852)} = 77.7 \text{ cfh/hen (or 1.295 cfm/hen)} \end{aligned}$$

b. Cropsey formula. In 1951, Cropsey (26) presented a mathematical analysis of the ventilation problem to show that it is theoretically possible to obtain dry litter in poultry houses by choosing the rate of flow in such a way as to obtain the lowest inside relative humidity.

With this idea in mind he derived the following formula for the relative humidity in terms of air changes:

$$R.H. = \frac{14.75 \left(0.0082 + \frac{0.326 C}{HV} \right)}{0.1217 \left(\frac{0.0048 \times 43C}{0.0191 HV + SU} \right) + \left(\frac{0.000095 \times (43C)^2}{(0.0191 HV + SU^2)} \right)}$$

where R.H. = Relative humidity (%)

C = Number of chickens

H = Number of air changes

V = Volume of the house (cu.ft.)

S = Exposed area of the house (sq.ft)

U = $U_{\text{aver.}} = C = \text{Average heat transmission coefficient for the house (Btu/Hr, sq. ft., °F)}$

The assumptions used in the derivation of this formula were: 40°F and 100% relative humidity outside conditions and 17 Btu of latent heat per four pound bird (Barne and Sammet values), assumptions which are debatable.

By further differentiating this equation with respect to N and then solving for N, Cropsey got the following equation:

$$N = 0.0191 \frac{SU}{V}$$

where N = Number of air changes

S = Exposed area of the house (sq.ft)

U = Aver. heat transfer coefficient of the house (Btu/Hr, sq. ft., °F)

V = Volume of the house (sq.ft)

which represents the number of air changes to secure the minimum relative humidity. It should be noted here that his last equation is not an exact differential for positive values of N. However, Cropsey in a letter to the writer states: "Practically the formulas for minimum relative humidity has worked out very well because there is only a small difference in relative humidity between the formula $N = \frac{SU (0.0191)}{V}$ and the true minimum. To illustrate refer to the table below (Table 42)."

Table 42. Number of air changes required to secure the minimum relative humidities and value of these minimums.*

U	$N = \frac{0.0191 SU}{V}$	R.H.	N exact minimum	R.H.
0.05	1.2	64.6%	1.39	64.4%
0.10	2.4	81.9%	3.28	81.1%
0.20	4.7	92.0%	7.66	90.6%
0.40	9.4	96.1%	16.94	95.1%

* From computations by Cropsey (26)

The procedure is first to find the number of air changes to secure the minimum relative humidity and then to substitute it in the first equation to get the value of this minimum relative humidity. Cropsey adds in his letter: "The formula for the exact minimum is long and cumbersome. With the formula above, one can quickly determine with reasonable accuracy the minimum relative humidity obtainable from a poultry house under almost the best conditions of ventilation. Also it will show very quickly what effect additional insulation will have on relative humidity."

Cropsey believes that by choosing the proper degree of ventilation, dry litter could be obtained with the use of a lesser amount of expensive insulation. To verify this statement experimentally he conducted some tests during the winter of 1950-51 and found the litter of the experimental poultry house to remain dry when the proper rate of ventilation as obtained by the use of his formula was applied. According to him a check pen of the same size required five changes of litter in order to maintain the litter below 40% moisture content.

The limitations of the Cropsey formulas are:

- 1-The formula is more difficult to solve than the standard formula
- 2-The assumption of 40°F outside temperature does not suit northern states
- 3-The formula is limited to four pound laying hens sheltered in houses where Barre and Cornet's heat values apply.

3. Poultry House Ventilation Charts

In this section the writer's intention is (a) to present in chronological order a review of all the ventilation charts prepared for simplification of poultry house ventilation design, (b) to analyze their basic assumptions, and (c) to point out their limitations.

1. Strahan ventilation chart

In 1940, Strahan (29) presented a performance chart (Figure 20) for poultry houses containing four pound birds. In the construction of this chart, he used the total heat production for active birds not in production (38.9 Btu/hr) and the percentage of latent heat (7.9%) given by Mitchell and Kelley (73). This percentage of latent heat corresponds to a poultry house temperature of 35°F which the author assumes to be constant for the range of outside temperatures from 5 to 30°F.

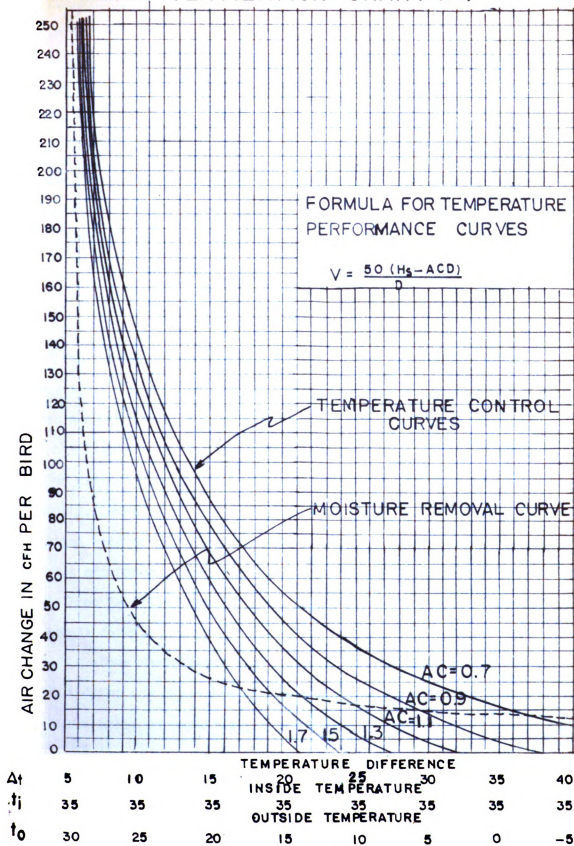
The solid curves represent the air flow variation required to maintain a constant temperature of 35°F. The dotted line shows the air flow requirement to remove the expired moisture. This line was computed on the assumption of 85% relative humidity and 35°F inside conditions and 100% relative humidity and a range from -5 to 30°F outside conditions. In figure 20, it is noted that the dotted line for removing expired moisture crosses each solid line for temperature control at only one point. This demonstrates the fact that for a given combination of temperature, AC value and outside temperature, only one rate of air flow satisfies both the heat balance and the moisture balance equations. This rate is called critical air flow rate. If the rate of air flow used is bigger than this critical air flow rate the relative humidity will decrease below 85%. If the rate of air flow used is smaller than this critical rate, moisture will accumulate in the house and condensation will occur if this condition remains long enough.

The limitations of the Strahan chart are as follows:

- 1-The inside temperature is considered constant at 35°F for the whole range of -5 to 30°F outside temperature, which conditions do not apply to all poultry houses.
- 2-The curve for moisture removal is also based on a 35°F constant inside temperature, although many poultry houses have thermostat settings of 40 and even 50°F.
- 3-The total heat production used is for active birds not in production, and do not include heat from litter decomposition and solar radiation.
- 4-The latent heat figure use is for active birds not in production.
- 5-The chart is limited to four pound hens.

FIGURE 20

STRAHAN VENTILATION CHART (99)



2. Barre and Sarnet ventilation chart

In 1942, Barre and Sarnet (10) presented a different type of performance chart (Figure 21) for poultry houses containing four-pound laying birds. In the construction of their chart they used their own estimates of the total heat (60 Btu/hr) and an average percentage of latent heat value of 10% for the range of inside temperatures from 30 to 60°F. The inside temperature instead of being assumed constant at 35°F, like Strahan did, is arbitrarily assumed to vary from thirty to sixty degrees with changes from -10 to 50°F in the outside temperature:

t_i	30	35	40	45	50	55	60
t_o	-10	0	10	20	30	40	50
Δt	40	35	30	25	20	15	10

The dotted curves represent the air flow variation required to maintain the desired inside temperature. The heavy curved lines crossing the dotted lines shows the air flow requirement to remove both the expired moisture and the liquid water eliminated in the excrement. This line was computed on the assumptions of 80% inside relative humidity and 80% outside relative humidity. Here also it is noted that for each set of conditions there is only one rate of air flow satisfying both the heat balance and the moisture balance equations. The point of intersection of the dotted lines and the heavy line is called critical air flow rate and has the same meaning as in Strahan's chart.

The above description shows the close relationship between Strahan's chart and the one presented by Barre and Sarnet. The difference lies mainly in the basic assumptions. However, one improvement was the addition by Barre and Sarnet of a chart for determining the maximum conductance which can be used without excessive condensation in the house. This chart is shown above the one for ventilation rates determination. The curves for

condensation were computed on the assumption of 30% inside and outside relative humidities and the relation of inside to outside temperature previously described.

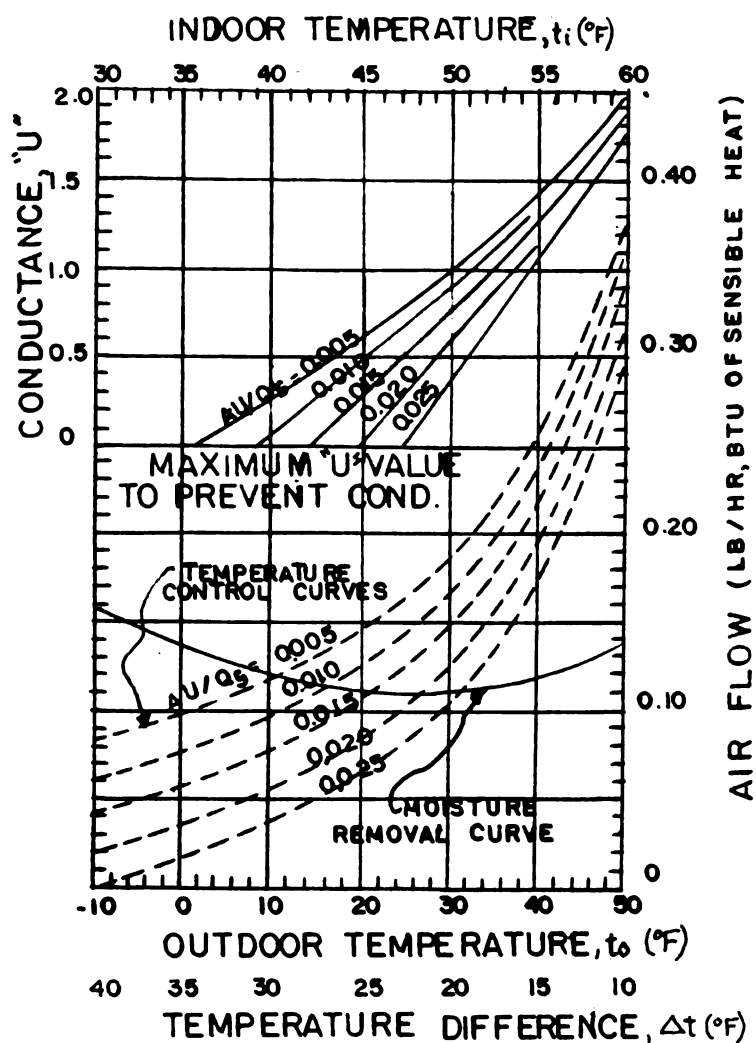
The limitations of the Barre and Garret chart are as follow:

- 1-The relation between inside and outside temperature is arbitrarily chosen and does not apply to all poultry houses.
- 2-The curve for moisture removal is based on 80% inside and outside relative humidities and does not apply to other conditions.
- 3-The air flow rate is given in pound per hour per Btu of sensible heat, and for practical use has to be converted to cfm.
- 4-The total heat production of birds is based on a wrong assumption as shown in a previous section.
- 5-The latent heat figures used include both water expired by the birds and water evaporated from the litter, although it seems that at very low temperature (critical design temperature), a part of this heat will be used for temperature control rather than for evaporation of moisture from the litter.
- 6-The chart is limited to four pound hens.

FIGURE 21

BARRE AND SAMMET

VENTILATION CHART (10)



NB.

AIR FLOW BASED ON ASSUMPTION OF NO CONDENSATION

$Q_s = 43$ BTU/HR, HEN

$Q_L = 17$ " "

MAXIMUM R.H. = 80 %

3. Stapleton and Cox ventilation chart

In 1950, Stapleton and Cox (95) presented a third type of performance chart (Figure 22) for poultry houses containing six pound hens. In the construction of their chart they assumed a total heat production of 52 Btu per six pound bird, as given by Mitchell and Kelley for active non-producing birds, but they included also the effect of solar radiation and litter decomposition as found in their tests. Their chart therefore groups the effect of heat from birds, solar radiation and litter decomposition. They also assumed the total moisture production as being 1.5 pounds per day per one hundred hens. Indoor air was taken at 80% relative humidity and outside air at 25°F and 70% relative humidity. Temperature differential range is from five to twenty-five degrees (Δt or D):

t_o	=	25	25	25	25	25
Δt	=	5	10	15	20	25
t_i	=	30	35	40	45	50

The chart has three sections. The heavy lines in the bottom section represent the air flow variation required to maintain the desired temperature differential. Three curves are shown for values of $AC = 50, 100$ and 200 , where A is the total exposed area per 100 six pound birds. Other curves can be added for different values of AC . This part of the chart can be applied to any outside temperature condition.

The heavy lines in the upper section of the chart show the air flow variation required to remove both the expired moisture and the liquid water eliminated in the excrement. Since outdoor temperature conditions were assumed to be 25°F and 70% in the computation of these curves, this section of the chart applies only for these conditions.

The dotted line in the lower section of the chart represents the maximum "U" value which can be allowed without excessive condensation. Two

curves are shown for values of $\Delta t = 0^\circ$ and 25°F . Other curves can be added for other values of Δt . Since the indoor relative humidity was assumed as being 80% in the computation of these curves, this section of the chart is limited to this condition.

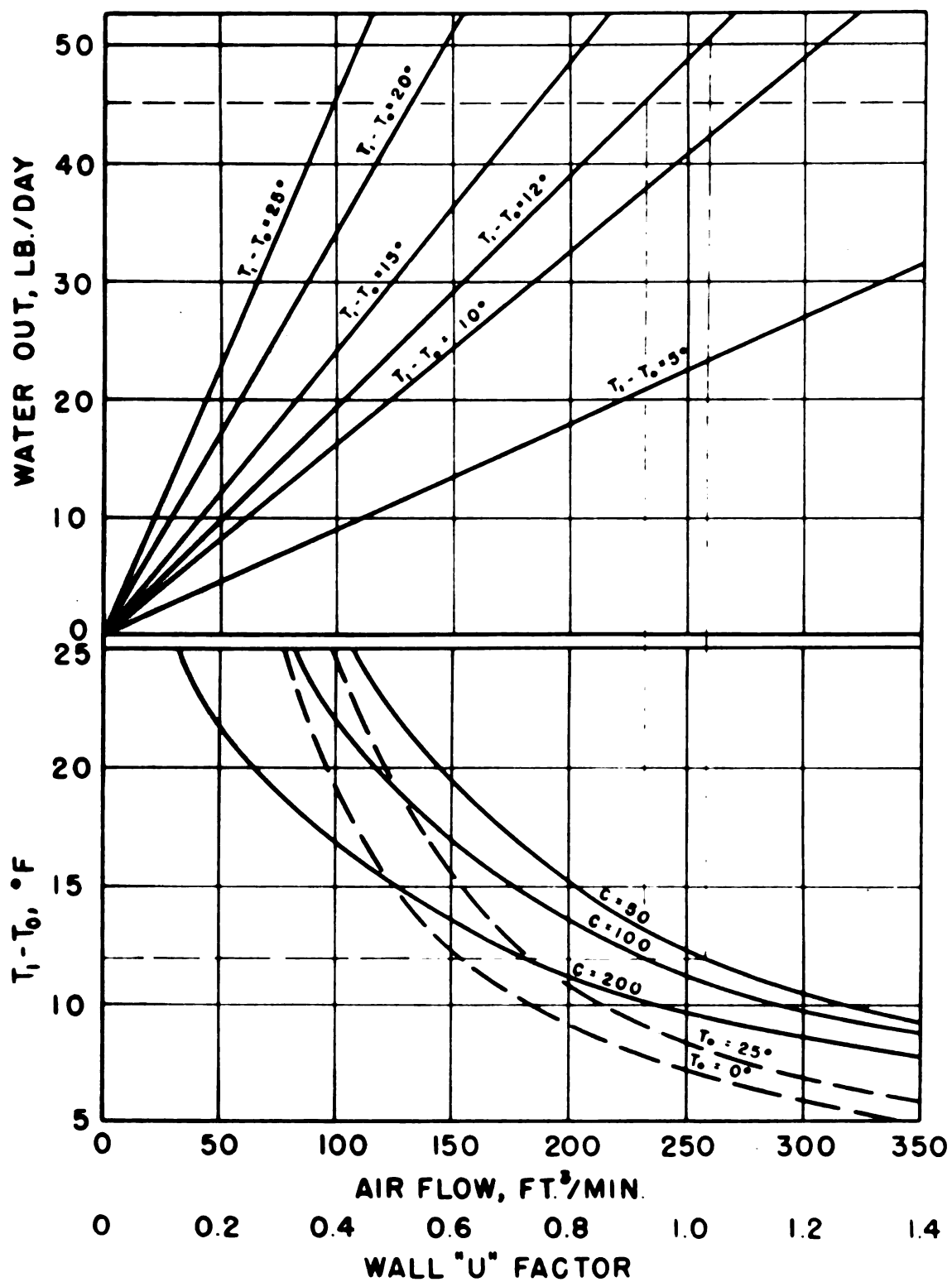
One advantage of this chart is that it is based chiefly on house differential temperature (Δt) which according to the authors is a factor readily measured.

The limitations of the Stapleton and Cox chart are:

- 1-Indoor and outdoor relative humidities are fixed, and the chart does not apply to other conditions.
- 2-The curves for moisture removal are based on 80% inside relative humidity and outside conditions of 25°F and 70% relative humidity, and do not apply to other conditions.
- 3-The curves for maximum "U" value possible without excessive condensation are based on 80% inside relative humidity and outside conditions of (a) 25°F and 70% relative humidity, and (b) 0°F and 70% relative humidity, although it seems that for design conditions an outside relative humidity of 100% is probably more suitable.

FIGURE 22

STAPLETON AND COX VENTILATION CHART (95)



h. Oliver ventilation charts.

Also in 1950, Oliver (78) presented a fourth type of ventilation chart (Figures 23-28) for poultry houses having an AC value equal to one and containing 100 heavy breed hens or 133 light breed hens (four pound hens). The total heat production was taken from Barre and Sammet (10) as being 60 Btu per hour per four-pound hen, which makes a total of 7980 Btu per hour for 133 four-pound hens. The total moisture production (expired and excreted moisture) was taken as 1.5 pounds per day per 133 four-pound hens. Approximately 2080 Btu per hour are required for the vaporization of this water, which left 5,900 Btu available for house losses and for heating the ventilating air. However, the author considered 6,000 Btu instead of 5,900 because of the solar heat (average of three to five hours of sunlight per day) and the heat from the morning and evening lights in the house.

The house losses in Btu per day and the sensible heat available for ventilation in Btu per day are shown here (Table 43) against difference between house temperature and outside temperature:

Table 43. Heat available for heating ventilating air, as affected by environmental temperature*

Differential between house and outside temperature (°F)	Total net sensible heat production (Btu/hr, 133 4-lb hens)	Building losses (Btu/hr)	Heat available for heating ventilating air (Btu/hr)
	Q_s	Q_b	Q_v
0°F	6000	0	6000
5°F	6000	500	5500
10°F	6000	1000	5000
20°F	6000	2000	4000
30°F	6000	3000	3000
40°F	6000	4000	2000

* From computations by Oliver (78)

Figures 23, 24 and 25 show the rate of ventilation required for temperature control. These three charts are based on the following equations:

- (a) Pounds of air required per hour for temperature control:

$$\text{Pounds of air/hr, 133 4-lb hens} = \frac{Q_v}{h_i - h_o}$$

- (b) Cfm of air required for temperature control:

$$\text{Cfm of air/133 4-lb hens} = \frac{(\text{lb air/hr, 133 4-lb hens}) \times (v_{i\text{act}})}{(60 \text{ min.})}$$

where Q_v = Part of the sensible heat available for heating the ventilating air (Btu)

h_i = Enthalpy of inside air at the given temperature and relative humidity (Btu/lb dry air)

h_o = Enthalpy of outside air at the given temperature and relative humidity (Btu/lb dry air)

$v_{i\text{act}}$ = Specific volume of the moist inside air (cu.ft/lb dry air)

Figures 26, 27 and 28 show the amount of moisture removed per day from the house for various conditions of outside temperature and humidity with different levels of ventilation, and also the temperature that will result in the house.

These three charts are based on the following equation:

- (c) Pounds of water removed per day:

$$\text{Pounds of water removed/day} = \frac{\text{lb air/hr, 133 4-lb hen} \times (W_i - W_o) \times 24 \text{ hrs}}{7000 \text{ grains/lb}}$$

where W_i = Grains of moisture per lb of inside air at the given temperature and relative humidity

W_o = Grains of moisture per lb of outside air at the given temperature and relative humidity

The amount of moisture removed should be above 1.5 pounds for the relative humidity of the house to stay below the relative humidity assumed in the charts and for the litter to remain dry. The conditions assumed for each of these charts are:

Chart	Inside conditions		Outside conditions	
	Temperature range	R.H. _i	Temperature range	R.H. _o
No. 1 & No. 4	20° to 60°F	90%	10° to 40°F	70%
No. 2 & No. 5	20° to 60°F	80%	10° to 40°F	80%
No. 3 & No. 6	20° to 60°F	90%	10° to 40°F	80%

Oliver (78) states that Figures 23 and 26 with their 70% average outside relative humidity are more representative of average winter conditions in northeastern states, while Figures 24, 25, 27 and 28 with their 80% outside relative humidity are more representative of the north central states.

Oliver's charts have some very good characteristics as will be seen by a check of them. Consider $t_o = 25^\circ\text{F}$ and $\text{R.H.}_o = 70\%$ while $t_i = 35^\circ\text{F}$ and $\text{R.H.}_i = 80\%$. Reference to a psychrometric chart gives the following:

	Temperature (°F)	Relative humidity (%)	Moisture content of the air (Grains/Lb)	Heat content of the air (Btu/Lb)	Specific volume (Cu. ft/Lb)
Inside air	35°	80%	$W_i = 24$	$h_i = 12.1$	$v_{i\text{act}} = 12.53$
Outside air	25°	70%	$W_o = 14$	$h_o = 8.2$	
	10°		$W_i - W_o = 10$	$h_i - h_o = 3.9$	

Looking in Table 43, for 10°F a value of 5,000 Btu/hr is found for

Q_v. By application of formulas (a) (b) and (c), the results obtained are:

$$(a) \text{ Pounds of air/hr, 133 4-lb hens} = \frac{Q_v}{h_i - h_o} = \frac{5000}{3.9} = 1282 \text{ lb/hr}$$

$$(b) \text{ Cfm of air 133 4-lb hens} = \frac{\text{lb air/hr, 133 4-lb hens} \times V_{\text{act}}}{60 \text{ min}} \\ = \frac{1282 \times 12.53}{60} = 267 \text{ cfm}$$

This value is read horizontally from Figure 24 at the point where 25° outside and 35°F inside temperatures meet.

$$(c) \text{ Pounds of water removed/day} = \frac{\text{lb air/hr, 133 4-lb hens} \times (w_i - w_o) \times 24 \text{ hrs}}{7000 \text{ grains/lb}} \\ = \frac{1282 \times 10 \times 24}{7000} = 44 \text{ lb/day}$$

This value is read horizontally in Figure 27 at the point where 25° outside and 35°F inside temperatures meet. It should be noticed that this point also gives obliquely the amount of ventilation required for temperature control as found from Figure 24 (267 cfm). Since the amount of water removed was found to be 44 pounds, one pound will remain in the house each day. According to Oliver (78): "The litter will have to absorb this or the air flow and moisture removal can be increased, which will produce a slight drop in the house and discharge temperature." The author adds that if the amount of water removed per day is above 45 lb/day: "... the relative humidity will drop until a balance is reached."

The limitations of Oliver's chart are:

- 1-The AC value of the house is considered equal to 1 and the charts do not apply to houses having different AC values.
- 2-The total heat production is taken from Barre and Sammet whose estimate was previously proven to be based on a wrong assumption.
- 3-Only one set of inside and outside relative humidity can be considered for each chart.

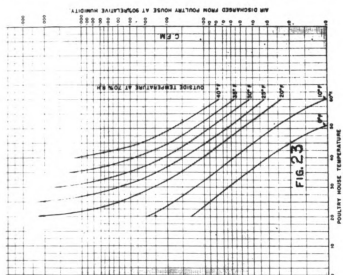
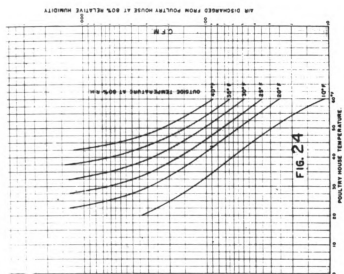
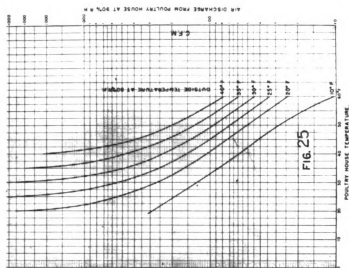
4.—The drawing of each chart involves numerous computations because of the curvature of the lines which require the plotting of many points.

5.—The latent heat figure used includes both water expired by the birds and water evaporated from the litter, although it seems that at a very low temperature (critical design temperature) a part of this heat will be used for temperature control rather than for evaporation of moisture from the litter.

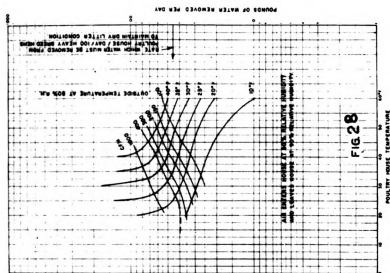
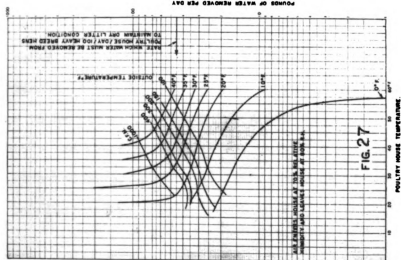
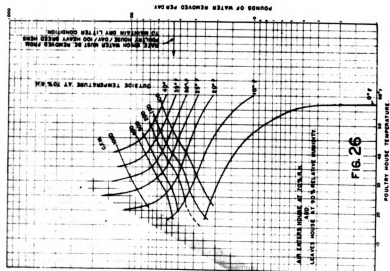
6.—The charts are for 133 four pound birds or 100 heavy breed birds only.

OLIVER VENTILATION CHARTS (95)

150



OLIVER VENTILATION CHARTS (95)



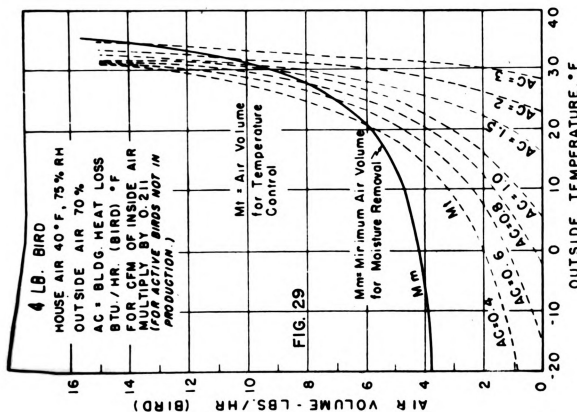
5. Giese and Bond ventilation chart

In 1952, Giese and Bond (35) presented a fifth type of performance chart (Figure 29) developed from the data of Mitchell and Kelley (73). This chart is for poultry houses containing four pound hens. The inside temperature is assumed constant at 40°F.

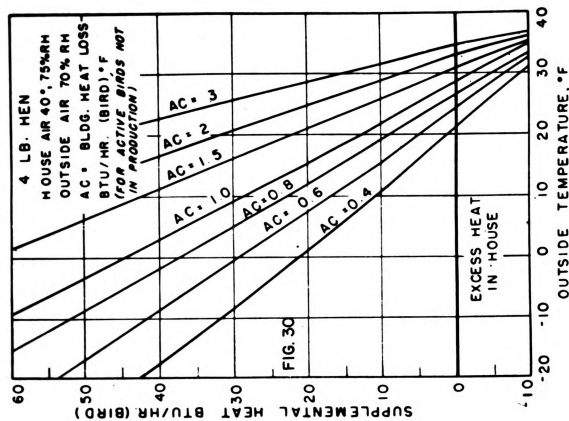
The dotted lines represent the air flow variation required to maintain a constant temperature of 40°F. The solid line shows the air flow requirement to remove the total moisture produced. This line was constructed on the assumption of 75% inside relative humidity and 70% outside relative humidity.

It will be noted that this chart is very similar to that of Strahan, the difference being in the basic assumptions of inside temperature and amount of moisture to be removed.

Its limitations are therefore similar to that of the Strahan chart and will not be repeated here. Figure 30 is for purpose of heat exchanger design and will be discussed in a later section.



GIESE AND BOND VENTILATION CHART (35)

GIESE AND BOND CHART
FOR HEAT EXCHANGER DESIGN (35)

11/11/11

11/11/11

11/11/11

4. Poultry House Ventilation Monographs

1. Rate of ventilation required for moisture removal

The weight of dry air entering a poultry house is considered equal to the weight of dry air leaving the house. However, the volumes of air and vapor entering and leaving the house are not equal. This fact makes it necessary to use two different equations to find the rate of ventilation required for moisture removal, depending on the type of ventilation system used.

$$1) V_L = \frac{W_T}{\left(\frac{W_i - W_o}{v_{oact}} \right)} = \frac{W_T \times v_{oact}}{W_i - W_o}$$

This formula gives the volume of air going in through the blower of a blow-in system or going in through the air intakes of an exhaust system.

$$2) V_L = \frac{W_T}{\left(\frac{W_i - W_o}{v_{iact}} \right)} = \frac{W_T \times v_{iact}}{W_i - W_o}$$

This formula gives the volume of air going out through the fan of an exhaust system or going out through the air exits of a blow-in system.

Since exhaust fans are used most of the time in animal shelters, this last equation is the most widely used. In these two equations, the meaning of the symbols is as follow:

V_L = Ventilation rate required for moisture removal

(Cu. ft. of moist air/hr.)

W_T = Rate of moisture production in the house

(lb of moisture/hr)

W_i = Moisture content of inside air (lb of moisture/lb dry air)

W_o = Moisture content of outside air (lb of moisture/lb dry air)

ΔW = Moisture carrying capacity of outside air (lb of moisture/lb dry air)

$v_{o \text{ act}}$ = Spec. vol. of the moist outside air (Cu. ft/Lb dry air)

$v_{i \text{ act}}$ = Spec. vol. of the moist inside air (Cu. ft/Lb dry air)

a. Parker's nomograph for determining the ventilation rate required for moisture removal. In 1952, Parker (32) presented a nomograph for determining the ventilation rate required for moisture removal. The theory on which this nomograph was constructed has been studied by the writer and is presented here.

This nomograph may be considered as being a combination of two equations each containing three variables:

$$1) \Delta w = \frac{w_i - w_o}{v_i} = \left(\frac{w_i}{v_i} \right) - \left(\frac{w_o}{v_i} \right) = K_1 - K_2$$

$$2) V_L = \frac{W_T}{60 \Delta w}$$

where Δw = Moisture carrying capacity of outside air (Lb of moisture/Cu. ft of moist inside air)

w_i = Moisture content of inside air (Lb of moisture/Lb of dry air)

w_o = Moisture content of outside air (Lb of moisture/Lb of dry air)

v_i = Specific volume of the moist inside air (Cu.ft/Lb of dry air)

$K_1 = \frac{w_i}{v_i}$ = Moisture content of inside air (Lb of moisture/Cu. ft. of moist air)

$K_2 = \frac{w_o}{v_i}$ = Has no practical meaning; is useful only for the mathematical solution.

W_T = Rate of moisture production in the house (Lb of moisture/Hr)

60 = To reduce the rate of ventilation from one hour
to one minute.

V_L = Ventilation rate required for moisture removal
(Cu.ft. of moist air/min.)

The first of the two equations is really a four variable equation and is not easily presented in a nomograph. It would result in a more complicated nomograph involving the use of three index lines instead of two. In view of simplifying this nomograph, Parker reduced this equation to a three variable one ($\Delta W, K_1, K_2$) by assuming fixed indoor-outdoor temperature relations:

Outdoor temperature	(t_o)	15	20	25	30	35	40	45	50	55
Assumed indoor temperature	(t_i)	30	35	40	45	45	50	55	60	60
Temperature difference	(Δt)	15	15	15	15	10	10	10	10	5

Such assumptions result in an error when these conditions do not hold true. However, according to Parker (32): "If the inside temperature deviates from the assumed value, the error would be rather small and may be neglected in practical applications."

The first equation may be changed to take the form $K_1 = K_2 + \Delta W$ where K_1 is the result of an addition. By reference to any textbook on nomographs (16), it can be seen that such an equation can be solved by a chart having three parallel scales. The two variables which are added are presented to the left and to the right and the result is read on the center scale (K_1). The modulus of each of these two scales determines the modulus of the center scale and the spacing of the scales.

In the second equation V_L is the result of a division. Reference to any textbook on nomographs (16) shows that such an equation can be solved by an N-chart composed of two parallel scales and one intermediary oblique scale for the result of the division (V_L).

Combining these two equations, the result is a five variable equation giving the ventilation rate required for moisture removal in a house of given design temperature and relative humidity conditions and of given water production.

$$V_L = \frac{W_T}{60 \Delta W} = \frac{W_T}{60 \left(\frac{W_1 - W_0}{v_1} \right)} = \text{cu. ft./Min.}$$

Values of W_0 and W_1 were taken from Zimmerman and Lavine, 1945 (110).

One application is presented on the nomograph and is checked here:

Application

Given: Number of hens = 500

Average weight of hens = 6 lb

Amount of moisture respired daily by the total number of birds =

43 lb (From Barott and Pringle) (or 8.6 lb/Day, 100 hens or

0.3533 lb/hr, 100 hens)

Outdoor design temperature = 5°F minimum

Outdoor design relative humidity = 100% (critical periods)

Indoor design temperature = 30°F minimum

Indoor design relative humidity = 80% (critical periods)

Wanted: The minimum rate of ventilation required to remove respired moisture at all times

True mathematical solution: (Zimmerman and Lavine, 1945)

$$W_{1\text{sat}} (30^\circ\text{F}) = 34.76 \times 10^{-4} \text{ lb/Lb dry air}$$

$$W_{180\%} (30^\circ\text{F}) = 34.76 \times 0.80 = 27.808 \times 10^{-4} \text{ lb/Lb dry air}$$

$$W_{0\text{sat}} (0^\circ\text{F}) = 9.322 \times 10^{-4} \text{ lb/Lb dry air}$$

$$\Delta W = 18.486 \times 10^{-4} \text{ lb/Lb dry air}$$

$$v_{i\text{act}} (30^\circ\text{F}, 80\%) = 12.39 \quad (\text{By use of slide rule No 5})$$

$$V_L = \frac{W_T \times v_{iact}}{6000 \Delta W} = \frac{0.3585 \times 12.39}{6000 \times 18.486 \times 10^{-4}} = \frac{0.3585 \times 12.39}{0.6 \times 18.486} = 0.1494 \text{ cfm/Hen}$$

Solution:

- 1-Find the intersection point for $t_o = 50^\circ\text{F}$ and R.H._o = 100% and project conditions horizontally to right edge of chart A to obtain W_o
- 2-Find the intersection point for $t_i = 30^\circ\text{F}$ and R.H._i = 80% and project conditions horizontally to right edge of chart B to obtain W_i
- 3-Draw an index line through W_o and W_i and find ΔW at the point where this index line intercepts the ΔW scale (Pivot line)*
- 4-Draw an index line through the value of ΔW found above and $W_T = 43 \text{ lb}$ and read the answer on the V_L scale = 230 cfm (or 0.46 cfm/Hen)

Percentage of error:

$$\% \text{ error} = \left(\frac{0.46 - 0.1494}{0.1494} \right) 100 = + \frac{0.0104 \times 100}{0.1494} = +2.3\%$$

In order to check more thoroughly the accuracy of this nomograph, the writer worked out 18 additional applications: ($W_T = 2.0 \text{ lb./Hr.}$, 100 hens or 1.8 lb./Day , 100 hens)

(*Parker did not put graduations on his W_i , W_o and ΔW scale, although he could have done so.)

Check	t_1	R.H. ₁	t_0	R.H. ₀	W_0	Number of hens	V_L (Nomograph) Total Per hen	V_L (Math. solut.)	Diff.	% error
No. 1	75	70	60	70	-	-	-	-	-	-
No. 2	65	80	50	50	288	600	400 0.665	0.6615	+0.0035	+0.5%
No. 3	60	90	50	90	240	500	760 1.52	1.44	+0.08	+1.55%
No. 4	50	80	40	60	288	600	862 1.45	1.442	+0.008	+0.56%
No. 5	50	90	40	100	144	300	775 2.58	2.58	0	0%
No. 6	40	80	30	70	144	300	720 2.40	2.43	-0.03	-1.25%
No. 7	40	70	30	90	48	100	745 7.45	8.14	-0.69	-8.50%
No. 8	40	70	20	70	192	400	825 2.06	2.06	0	0
No. 9	35	90	30	40	240	500	840 1.68	1.703	-0.023	-1.35%
No. 10	35	80	20	50	240	500	910 1.82	1.83	-0.01	-0.55%
No. 11	30	90	20	100	96	200	275 4.88	4.90	-0.02	-0.41%
No. 12	30	80	10	90	144	300	860 2.86	2.84	+0.02	+0.70%
No. 13	30	90	10	100	144	300	750 2.50	2.50	0	0
No. 14	30	80	0	60	144	300	568 1.89	1.86	+0.03	+1.61%
No. 15	25	90	10	70	144	300	820 2.73	2.70	+0.03	+1.1%
No. 16	25	80	-20	80	144	300	620 2.07	2.055	+0.015	+0.73%
No. 17	20	80	0	8	96	200	750 3.75	3.72	+0.03	+0.81%
No. 18	20	80	-20	90	144	300	810 2.70	2.66	+0.04	+1.5%

To conclude from the above application and checks, it can be said that the accuracy of this nomograph is very good in general, since of the 18 applications chosen at random, only two were in error by more than 5%.

NOMOGRAPH FOR DETERMINING MOISTURE REMOVED

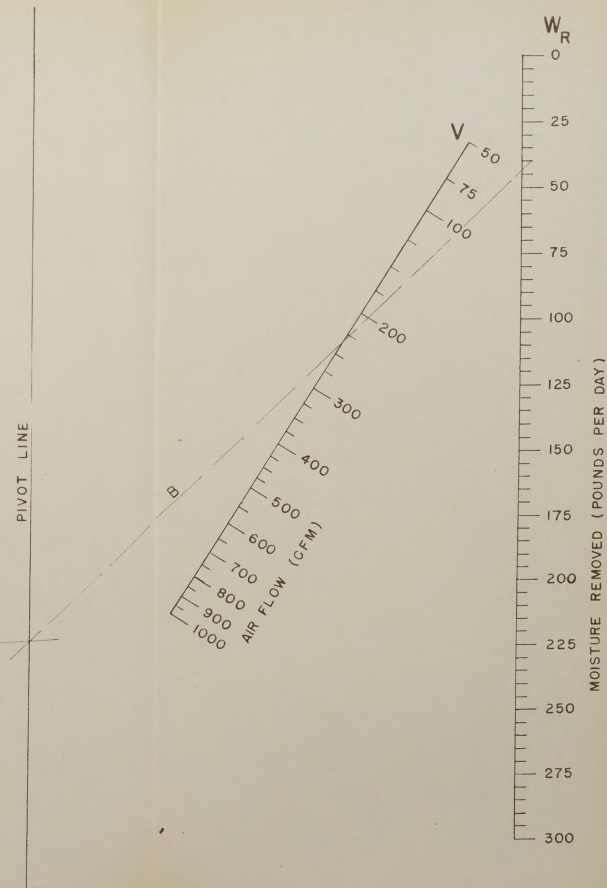
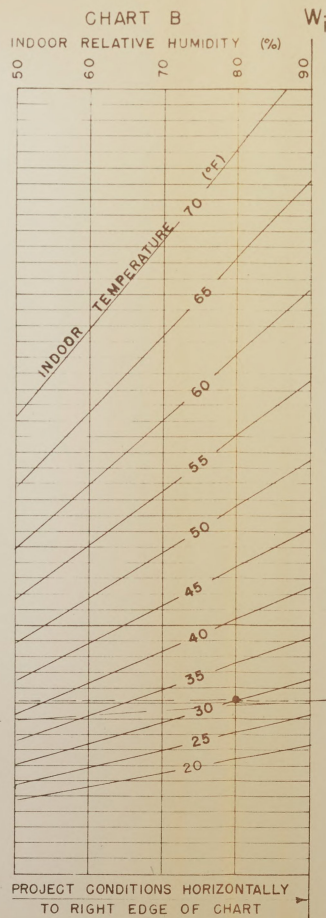
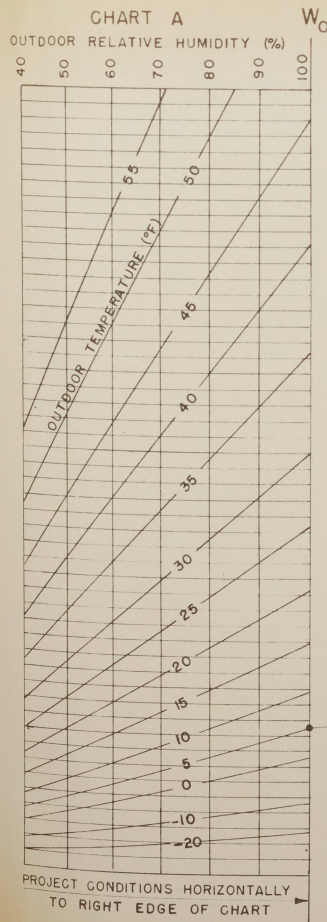


Figure I

b. Choinière nomograph for determining the ventilation rate required for moisture removal. The writer is presenting here a second type of nomograph to solve the general equations for the rate of ventilation required for moisture removal.

This nomograph is a combination of the same equations as the ones used by Parker (82), except that the meaning of the symbols is changed somewhat as shown here:

$$1) \Delta W = \frac{W_1 - W_0}{v_1} = \left(\frac{W_1}{v_1} \right) - \left(\frac{W_0}{v_1} \right) = K_1 - K_2$$

$$2) V_L = \frac{W_T}{60 \times 100 \times \Delta W} = \frac{W_T}{6000 \Delta W}$$

where ΔW = Moisture carrying capacity of outside air

(Lb. /cu. ft. of moist inside air)

W_1 = Moisture content of inside air (Lb./Lb. dry air)

W_0 = Moisture content of outside air (Lb./Lb. dry air)

v_1 = Specific volume of the moist inside air

(Cu. ft./Lb. dry air)

$K_1 = \frac{W_1}{v_1}$ = Moisture content of inside air

(Lb./Cu. ft. of moist air)

$K_2 = \frac{W_0}{v_1}$ = Moisture content of the outside air per cubic

foot of moist air, when considered as having the

temperature and the specific volume of the moist

inside air. This term has a mathematical meaning.

However, for convenience it will be called "Mois-

ture content of the outside air" (Lb./Cu. ft. of

moist air)

W_T = Rate of moisture production in the house
(lb/Hr, 100 hens) and (lb/Day, 100 hens)

60 = To reduce the rate of ventilation from one
hour to one minute

100 = To reduce the rate of ventilation from 100
hens to one hen

V_L = Ventilation rate required for moisture
removal (cfm/Hen)

Equation 1 is really a four variable equation and is not easily presented in a nomograph (it would result in a more complicated nomograph). In view of reducing the first equation to a three variables form, three methods can be used:

- 1-The volume of entering air may be considered equal to the volume of outgoing air (i.e. we may consider that for each cubic foot of inside air going out, there is a cubic foot of outside air going in).
- 2-A constant v_i value may be assumed
- 3-A variable v_i value may be assumed, (i.e. we may consider fixed indoor-outdoor temperature relations).

The writer computed some typical percentage of expected error which would be obtained by assuming the volume of entering air as equal to the volume of outgoing air. The results are shown here: (W_T = 2 lb/Hr, 100 hens, or 4.8 lb/Day, 100 hens). Values of W_i and W_o for these computations were taken from Heating, Ventilating, Air Conditioning Guide, 1951 (6), and values of v_i were taken from Table 44.

t_i	R.H. _i	t_o	R.H. _o	Δt	True math. solut.	V_L (cfm/hour) Assumption of equal volumes	Difference	Expected error (%)
(°F)	(%)	(°F)	(%)	(°F)				
60	70	55	80	5	11.81	11.72	+ 2.88	+24.3%
		50	80	10	2.71	2.95	+ 0.24	+ 8.9
		45	80	15	1.64	1.748	+0.108	+ 6.6
		40	80	20	1.23	1.298	+0.068	+ 5.5
50	80	45	90	5	10.80	11.75	+0.95	+8.8%
		35	90	15	1.395	2.013	+0.118	+6.7
		25	90	25	1.178	1.223	+0.045	+6.2
		15	90	35	0.937	0.963	+0.026	+2.77
		0	90	50	0.797	0.809	+0.012	+1.51
40	80	35	90	5	13.07	—	—	—
		25	100	15	2.938	3.135	+0.197	+6.3%
		15	100	25	1.699	1.765	+0.066	+3.88
		5	100	35	1.34	1.375	+0.035	+2.65
		0	100	40	1.247	1.275	+0.028	+2.24
30	80	25	100	5	127.1	—	—	—
		15	100	15	3.84	4.04	+0.200	+5.2%
		5	100	25	2.363	2.448	+0.085	+3.6
		0	100	30	2.088	2.145	+0.057	+2.73
		-5	100	35	1.912	1.954	+0.042	+2.19
20	90	15	100	5	116.8	—	—	—
		5	100	15	5.77	6.05	+0.28	+4.8%
		0	100	20	4.325	4.49	+0.165	+3.82
		-5	100	25	3.62	3.728	+0.108	+2.78
		-15	100	35	2.943	3.00	+0.057	+1.935

Since expected errors above 4 or 5% could easily be encountered, the above assumption was therefore put aside.

The writer computed next the percentage of expected error which would be obtained by using constant v_i values of 12.53 (35°F, 80% R.H.) and 12.67 (40°F, 80% R.H.). The results are shown here:

t_i (°F)	v_{iact} (80%)	Difference		Expected error (%)	
		1-Difference from 12.53	2-Difference from 12.67	1-Use of 12.53	2-Use of 12.67
75	13.75	+1.22	+1.08	+9.2%	8.6%
70	13.61	+1.08	+0.94	+8.6	7.4
65	13.44	+0.91	+0.77	+6.7	6.1
60	13.28	+0.75	+0.61	+5.6	4.8
55	13.12	+0.59	+0.55	+4.5	4.3
50	12.97	+0.44	+0.30	+3.4	2.4
45	12.82	+0.29	+0.15	+2.2	1.2
40	12.67	+0.14	0	+1.1	0
35	12.53	0	-0.14	0	1.1
32	12.45	-0.08	-0.22	-0.6	1.7
30	12.39	-0.14	-0.28	-1.1	2.2
25	12.25	-0.28	-0.42	-2.2	3.3
20	12.11	-0.42	-0.56	-3.3	4.4

< 4%
error

< 4%
error

As shown in the table, nomographs based on this second assumption would be in error by as much as 4 to 10% for computations involving the use of temperatures from 50 to 75°F. This type of simplification was therefore put aside.

The third method of simplification consists in assuming a variable v_i value or in other words in assuming fixed indoor-outdoor temperature relations, as used previously by Parker (82). Since Parker's nomograph seemed to have a rather satisfactory accuracy, the writer choose this method of simplification as the most reliable one. The relations used were as follow:

t_o	65	60	55	50	45	40	35	30	25	20
t_i	70	65	60	60	55	50	45	45	40	35
Δt	5	5	5	10	10	10	10	15	15	15
v_i (80%)	13.61	13.44	13.28	13.28	12.97	12.97	12.82	12.82	12.67	12.53

It should be noted that this method involves an error but as mentioned previously by Parker (82) the error will generally be rather small and may be neglected in practical applications.

Modified psychrometric charts for $\frac{W_1}{V_1}$ and $\frac{W_o}{V_o}$ were computed by the writer, using data from the Heating, Ventilating, Air Conditioning Guide, 1951(6). These charts are presented as curved lines, on the nomograph, for purpose of resembling the standard psychrometric charts.

Two variations of the same nomograph were made. The first one is a single nomograph requiring two index lines meeting on the ΔW scale as a pivot line. This type of nomograph is especially suitable when one is interested only in finding the minimum rate of ventilation required for moisture removal. The second one is made of two separate nomographs requiring an index line for each. The advantage of this type is to give a perpendicular V scale with uniform graduation which make it possible to make a complete heat and moisture balance nomograph as will be done later in Part III.

In order to illustrate the use of these nomographs, consider the following applications:

Application No. 1

Given: Rate of water production in the house = 45 lb/Day, 100 hens
(or 1.875 lb/Hr, 100 hens)

Indoor temperature = 40°F.

Indoor relative humidity = 80%

Outdoor temperature = 0°F.

Outdoor relative humidity = 100%

Wanted: The minimum rate of ventilation required under the above condition to remove all the moisture produced.

True mathematical solution: (Heating, Ventilating, Air Conditioning Guide, 1951)

$$W_{1\text{sat}} (40^\circ\text{F}) = 52.13 \times 10^{-4} \text{ lb/Lb dry air}$$

$$W_{180\%} (40^\circ\text{F}) = 52.13 \times 0.80 = 41.704 \times 10^{-4} \text{ lb/Lb dry air}$$

$$W_{0sat} (0^{\circ}F) = 7.852 \times 10^{-4} \text{ lb/lb dry air}$$

$$\Delta W = 41.704 - 7.852 = 33.852 \text{ lb/lb dry air}$$

$$v_{iact} (40^{\circ}F, 80\%) = 12.67 \text{ (From Table 14)}$$

$$V_L = \frac{W_T \times v_{iact}}{6000 \Delta W} = \frac{1.875 \times 12.67}{6000 \times 33.852 \times 10^{-4}} = \frac{1.875 \times 12.67}{0.6 \times 33.852} = 1.1696 \text{ cfm/Hen}$$

Solution by use of the nomographs:

- 1-Find the intersection point for $t_o = 0^{\circ}F$ and R.H._o = 100% and project conditions horizontally to right edge of chart A to obtain $W_o = 6.4$
- 2-Find the intersection point for $t_i = 40^{\circ}F$ and R.H._i = 80% and project conditions horizontally to right edge of chart A to obtain $W_i = 33.0$
- 3-Draw an index line through W_o and W_i and find $\Delta W = 26.6$ at the point where this index line intercepts the ΔW scale
- 4-Draw an index line through the value of $\Delta W = 26.6$ found above and $W_T = 45$ lb and read the answer on the V_L scale = 1.175 cfm/Hen.

Percentage of error:

$$\% \text{ error} = \frac{1.175 - 1.1696}{1.1696} \times 100 = \frac{+0.0054 \times 100}{1.1696} = +0.46\%$$

Application No. 2

Given:

Rate of water production in the house = 45 lb/Day, 100 hens
(or 1.875 lb/Hr, 100 hens)

Indoor temperature = 50°F

Indoor relative humidity = 80%

Outdoor temperature = 20°F

Outdoor relative humidity = 100%

Wanted:

The minimum rate of ventilation required under the above conditions to remove all the moisture produced.

True mathematical solution: (Heating, Ventilating, Air Conditioning Guide, 1951)

$$W_{i\text{sat}} (50^{\circ}\text{F}) = 76.58 \times 10^{-4} \text{ lb/Lb dry air}$$

$$W_{i80\%} (50^{\circ}\text{F}) = 76.58 \times 0.80 = 61.264 \times 10^{-4} \text{ lb/Lb dry air}$$

$$W_{o\text{sat}} (20^{\circ}\text{F}) = 21.52 \times 10^{-4} \text{ lb/Lb dry air}$$

$$W = 39.744 \times 10^{-4} \text{ lb/Lb dry air}$$

$$v_{i\text{act}} (50^{\circ}\text{F}, 80\%) = 12.97 \text{ (From Table 144)}$$

$$V_L = \frac{W_T \times v_{i\text{act}}}{6000 \times W} = \frac{1.875 \times 12.97}{6000 \times 39.744 \times 10^{-4}} - \frac{1.875 \times 12.97}{0.6 \times 39.744} = 1.0198 \text{ cfm/Hen}$$

Solution by use of the nomographs:

- 1-Find the intersection point for $t_o = 20^{\circ}\text{F}$ and $\text{R.H.}_o = 100\%$ and project conditions horizontally to right edge of chart A to obtain $W_o = 17.3$
- 2-Find the intersection point for $t_i = 50^{\circ}\text{F}$ and $\text{R.H.}_i = 80\%$ and project conditions horizontally to right edge of chart B to obtain $W_i = 17.3$
- 3-Draw an index line through W_o and W_i and find $\Delta W = 30$ at the point where this index line intercepts the ΔW scale.
- 4-Draw an index line through the values of $\Delta W = 30$ and $W_T = 145$ pounds and read the answer on the V_L scale = 1.04 cfm/Hen.

Percentage of error:

$$\% \text{ error} = \left(\frac{1.04 - 1.0198}{1.0198} \right) 100 = \frac{+0.0202 \times 100}{1.0198} = +1.98\%$$

In order to check more thoroughly the accuracy of these nomographs, the writer worked out 18 additional applications: ($W_T = 2.0 \text{ lb/Hr}$, 100 hens, or 48 lb/Day, 100 hens)

Check	t_i	R.H. _i	$W_{i,act}$	t_o	R.H. _o	$W_{o,act}$	ΔW	W_T	V_L	Diff.	Error	
	(°F)	(%)	*	(°F)	(%)	*	*	**	Use of the nom- ograph	True math. solut.	(%)	
No. 1	75	70	94.0	60	70	57.1	36.9	2.0	0.860	0.845	0.023	0.72
No. 2	65	80	79	50	50	39.0	46.0	2.0	0.76	0.761	0.001	0.15
No. 3	60	90	75.1	50	20	51.9	23.2	2.0	1.13	1.140	0.01	0.70
No. 4	50	80	47.2	40	60	24.0	23.2	2.0	1.13	1.111	0.011	0.77
No. 5	50	90	53.2	40	100	40.2	13.0	2.0	2.55	2.512	0.17	0.66
No. 6	40	80	32.8	30	70	19.0	13.0	2.0	2.40	2.385	0.005	0.21
No. 7	40	70	29.0	30	90	24.2	4.7	2.0	7.50	7.80	0.30	3.9
No. 8	40	70	29.0	20	70	13.1	15.9	2.0	1.07	1.965	0.005	0.25
No. 9	35	90	31.0	30	40	10.8	20.2	2.0	1.65	1.695	0.045	2.66
No. 10	35	80	27.4	20	50	8.4	18.6	2.0	1.60	1.78	0.002	1.13
No. 11	30	90	25.2	20	100	17.2	8.0	2.0	4.20	4.32	0.12	2.78
No. 12	30	80	22.4	10	90	9.4	12.8	2.0	2.63	2.71	0.02	0.77
No. 13	30	90	25.2	10	100	10.5	14.8	2.0	7.27	2.30	0.03	1.3
No. 14	30	80	22.4	0	60	2.7	13.8	2.0	1.78	1.805	0.025	1.35
No. 15	25	90	20.2	10	70	7.5	12.8	2.0	2.63	2.65	0.02	0.76
No. 16	25	80	18.0	-20	100	2.0	16.0	2.0	2.10	2.123	0.023	1.03
No. 17	20	80	14.0	0	80	5.0	8.8	2.0	3.80	3.69	0.11	2.98
No. 18	20	80	14.0	-20	100	2.0	11.8	2.0	2.82	2.767	0.053	1.92

* LB/CU.FT. OF MOIST AIR * 10-4

** LB/HR, 100 HENS

From the above applications and checks, it is seen that the accuracy of this nomograph is in general within 3 or 4% for the whole range of inside temperature from 20 to 75°F.

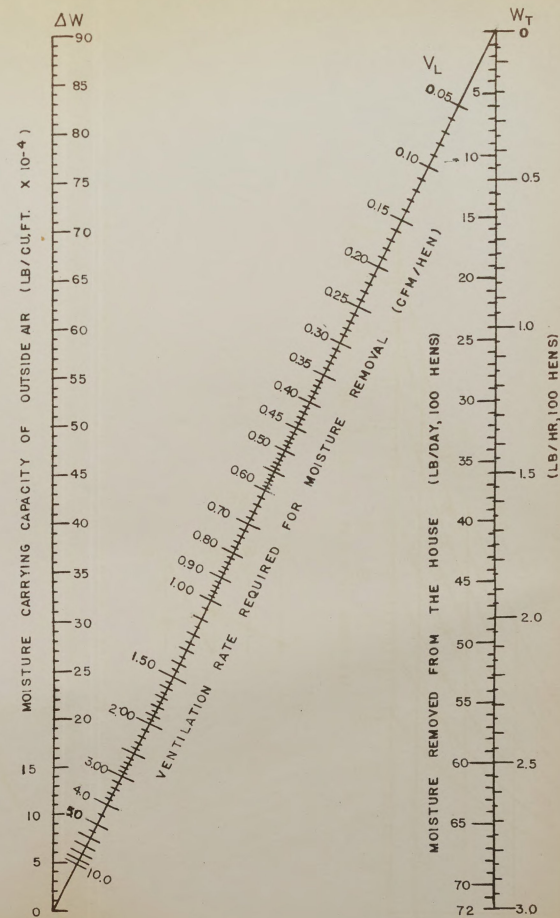
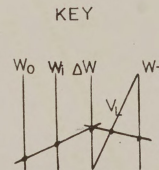
Some of the advantages of this nomograph over Parker's are:

- 1-The ventilation rate is given in cfm per hen which makes it very suitable for purpose of comparison and analysis (To find the fan capacity for a given house, this value must be multiplied by the number of birds in the house, which is a simple operation)
- 2-The water production is given both in pounds per hour and in pounds per day per 100 hens
- 3-The V scale of the second nomograph presented by the writer is vertical, which makes it possible to make a heat and moisture balance nomograph, as will be done later.

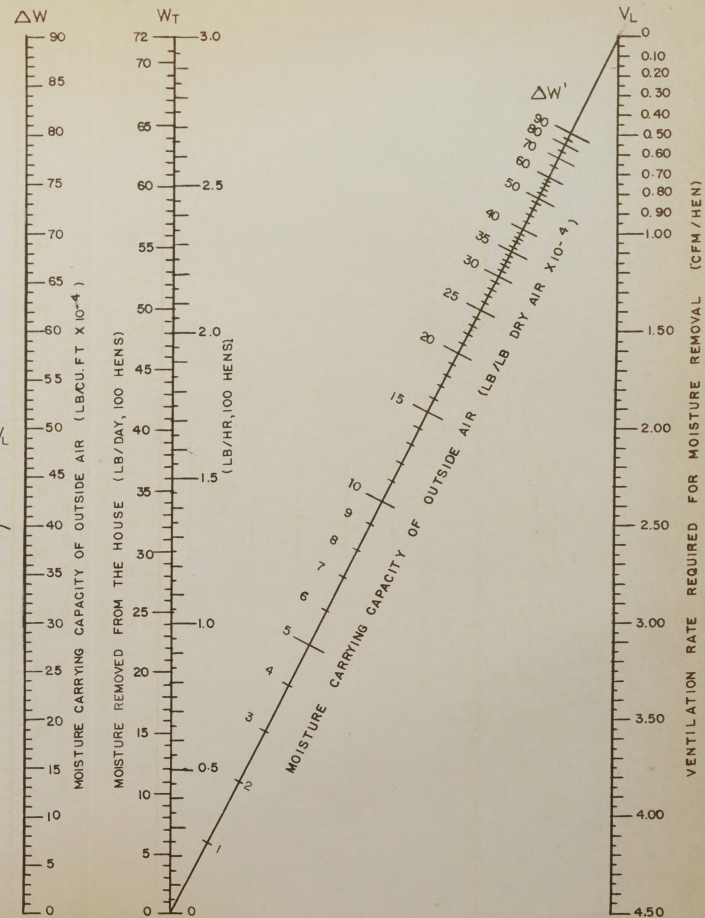
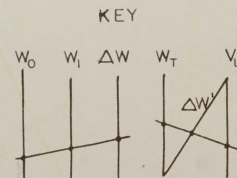
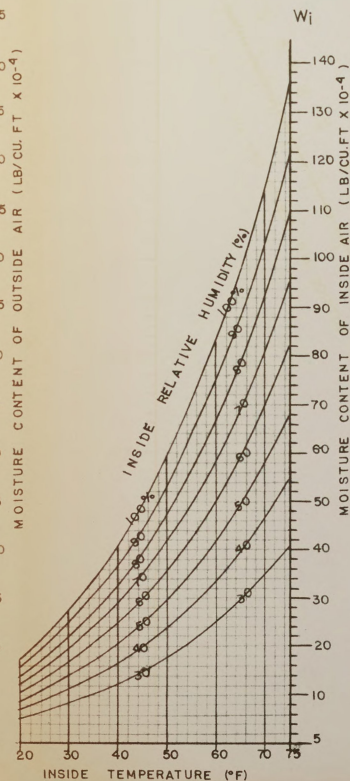
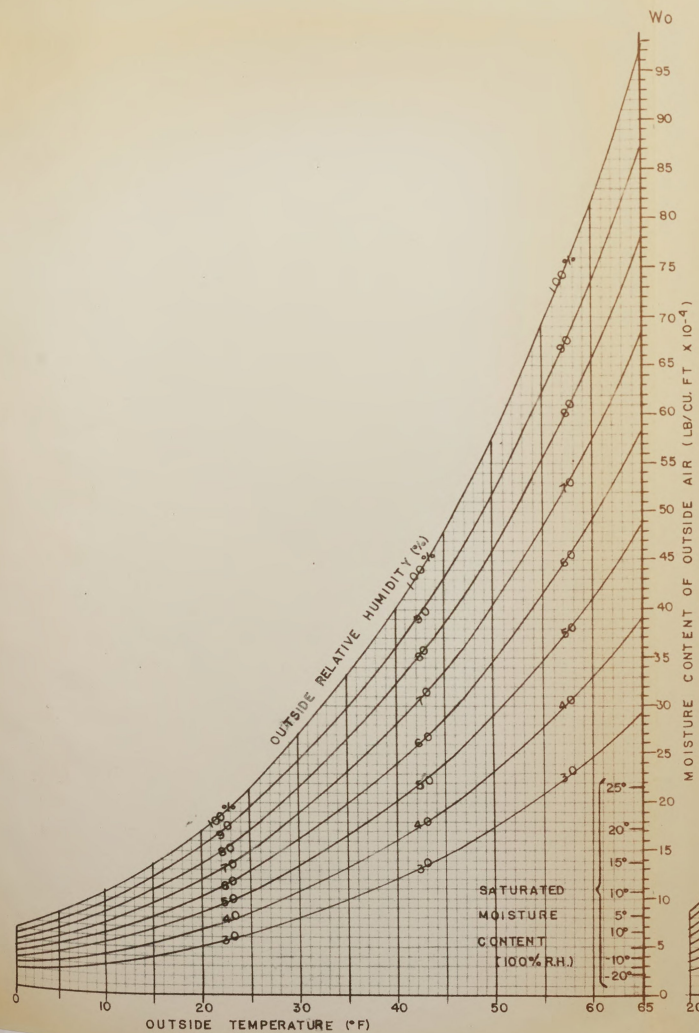
To conclude from Parker's and the writer's nomographs for moisture removal, it can be said that these nomographs are only approximate because of the assumption of fixed indoor-outdoor temperature relations on which they are based. However, as shown by the numerous checks done for each of them, they are accurate enough for most design problems.



NOMOGRAPH FOR DETERMINING THE RATE OF VENTILATION REQUIRED FOR MOISTURE REMOVAL



NOMOGRAPH FOR DETERMINING THE RATE OF VENTILATION REQUIRED FOR MOISTURE REMOVAL



2. Rate of ventilation required for temperature control

a. Parker's nomograph for determining the ventilation rate required for temperature control. In addition to his nomograph for the moisture removal formula, Parker (82) presented a second nomograph for determining the ventilation rate required for temperature control. The theory by which this nomograph was constructed is shown here.

This nomograph may be considered as being a combination of two equations containing three and four variable respectively

$$1) Q_s = N \times q_s$$

where Q_s = Total sensible heat production in the house (Btu/Hr)

N = Number of birds in the house

q_s = Sensible heat production per hen (Btu/Hr, hen)

$$2) Q_s = "AU" \Delta t + 60 V_s c \Delta t$$

where Q_s = Total sensible heat production in the house (Btu/Hr)

A = Total exposed area of the house (Sq.ft)

U = Overall heat transfer coefficient of the house
(Btu/Hr, sq.ft., °F)

"AU" = Heat loss characteristic of the house (Btu/Hr, °F)

"AU" is considered here as a whole i.e. as a simple variable

Δt = Temperature differential between inside and outside air (°F)

60 = Number of minutes in an hour

V_s = Ventilation rate required for temperature control
(Lb of dry air/minute)

c = Specific heat of air at constant pressure = 0.24

The first equation may be changed to take the form $N = \frac{Q_s}{q_s}$ where N is the result of a division. This equation can therefore be solved by an H-chart composed of two parallel scales and one intermediary oblique scale for the result of the division (N). As to the second equation, it must first be separated into two equal parts each containing two variables and made equal to a third variable (dummy variable), for instance K .

$$2a) K = \frac{Q_s}{\Delta t}$$

$$2b) K = "AU" + 60 V_s \times c$$

$$K = "AU" + 60 V_s \times 0.24$$

$$K = "AU" + 12.4 V_s$$

Equation 2a can be changed to take the form $\Delta t = \frac{Q_s}{K}$ where Δt is the result of a division. This equation can therefore be solved by an H-chart with Δt as the oblique scale and K as the scale to the outer right. In equation 2b K is the result of an addition. This equation can therefore be solved by a chart having three parallel scales, K being the center scale.

Solving equation 2) for V_s the result is:

$$V_s = \frac{Q_s - "AU" \Delta t}{60 c \Delta t} = \text{lb of dry air required for temperature control/Min.}$$

To convert this weight of ventilating air to cfm of air, it would be necessary to multiply the result by $v_{i,act}$ (Specific volume of one pound of moist inside air at the given temperature and relative humidity). Since this value is variable, its introduction would complicate the nomograph and would involve the use of four index lines instead of three.

In view of simplifying his nomograph, Parker (82) assumed the value of $v_{i,act}$ to be constant at 12.72 (dry air at 45°F) which corresponds to $Z = \frac{v_1}{0.24} = \frac{12.72}{0.24} = 53$. Such an approximate conversion involves an error. However, Parker states that: "The small discrepancy involved at temperatures other than 45°F may be neglected in general application work."

In order to illustrate the use of this second nomograph, consider the following applications concerning the same poultry house as the one analyzed previously for moisture removal.

Application No. 1

Given:

Number of hens = 500

Average weight of hens = 6 lb

Amount of sensible heat produced hourly by one bird = 57.5

(From Mitchell & Kelley total heat production and Barott and Pringle latent heat production)

Outdoor design temperature = 5°F minimum

Indoor design temperature = 30°F minimum

"AU" value of the house = 11.96 (Where A = Total exposed area of the house)

Wanted:

The maximum rate of ventilation required to maintain the above temperature differential ($\Delta T = 30 - 5 = 25^\circ$)

True mathematical solution:

v_1 (30°F, 80%) = 12.39 (By use of slide rule No 5)

$$AC = \frac{AU}{N} = \frac{11.96}{500} = 2.992 \text{ Btu/Hr, } ^\circ\text{F, Hen}$$

$$v_s = \frac{v_1}{0.24} \left(\frac{Q_s}{D} - AC \right) \frac{1}{60}$$

$$= \frac{12.39}{0.24} \left(\frac{57.5}{25} - 2.992 \right) \frac{1}{60}$$

$$= \frac{12.39}{0.24} (2.3 - 2.992) \frac{1}{60}$$

$$= \frac{12.39 \times 0.692}{0.24 \times 60} = -0.5954 \text{ cfm/Hen (or -297.7 cfm/500 hens)}$$

(The negative sign indicates that it is impossible to maintain a temperature difference of 25°F)

Solution by use of the nomograph:

- 1-Draw an index line through $W = 6$ lb (or $q_s = 57.5$ Btu) and $N = 500$ and find Q_s at the point where the index line intercepts the Q_s scale.
- 2-Draw an index line through the value of Q_s found above and $\Delta t = 25^\circ\text{F}$ and find K (dummy variable) at the point where the index line intercepts the pivot line.
- 3-Draw an index line through the value of K found above and $AU = 11496$ to find the answer. Since the V_s scale indicates less than 0 cfm (or -290 , by extending the scale below 0). This shows that all the heat produced by the chickens is lost by conduction through exposed building surfaces and no heat is available for heating the ventilating air.

Percentage of error:

$$\% \text{ error} = \left(\frac{290 - 297.7}{297.7} \right) 100 = \frac{7.7 \times 100}{297.7} = 2.6\%$$

Application No. 2

Given: Same data as for application No. 1, except that the "AU" value of the house is given as 900.

Wanted: The minimum rate of ventilation required to maintain a temperature differential of 25° (Δt)

True mathematical solution:

$$v_1 (30^\circ\text{F}, 80\%) = 12.39 \text{ (By use of slide rule No 5)}$$

$$AC = \frac{AU}{N} = \frac{900}{500} = 1.8 \text{ Btu/Hr, } ^\circ\text{F, Hen}$$

$$\begin{aligned}
 V_S &= \frac{v_1}{0.24} \left(\frac{Q_s - AC}{D} \right) \frac{1}{60} \\
 &= \frac{12.39}{0.24} \left(\frac{57.5}{25} - 1.8 \right) \frac{1}{60} \\
 &= \frac{12.39}{0.24} (2.3 - 1.8) \frac{1}{60} = \frac{12.39 \times 0.5}{0.24 \times 60} = 0.43 \text{ cfm/hen} \\
 &\quad (\text{or } 215 \text{ cfm/500 hens})
 \end{aligned}$$

Solution by use of the nomograph:

1 and 2 = Same as for Application No. 1

3 = Draw an index line through the value of K found above and

$\Delta U = 900$ and read the answer on the V_S scale = 225 cfm

per 500 six-pound hens (or 0.46 cfm per hen)

Percentage of error:

$$\% \text{ error} = \frac{225 - 215}{215} \times 100 = \frac{10 \times 100}{215} = +4.65\%$$

Application No. 3

Given:

Number of hens = 500

Average weight of hens = 6 lb

Amount of sensible heat produced hourly by one bird = 57.5 Btu

(From total heat production by Mitchell and Kelley and latent heat production by Barott and Pringle)

Outdoor design temperature = 5°F minimum

Indoor design temperature = 30°F minimum

$V_S = 230$ cfm per 500 six-pound hens (or 0.46 cfm per hen)

Wanted:

The "AU" value of the house required to maintain the above temperature differential = $30 - 5 = 25^\circ (\Delta t)$

True mathematical solution:

$$v_1 (30^\circ\text{F}, 80\%) = 12.39 \text{ (By use of slide rule No 5)}$$

$$V_S = \frac{230}{500} = 0.46 \text{ cfm/Hen}$$

$$\begin{aligned} AC &= \frac{Q_s}{D} - \frac{60 V_S}{\left(\frac{v_i}{0.24}\right)} \\ &= \frac{57.5}{25} - \frac{60 \times 0.46}{\left(\frac{12.39}{0.24}\right)} \\ &= \frac{57.5}{25} - \frac{60 \times 0.46 \times 0.24}{12.39} = 2.3 - 0.5346 = 1.7654 \end{aligned}$$

Solution by use of the nomograph:

- 1-Draw an index line through $W = 6 \text{ lb}$ (or $Q_s = 57.5 \text{ Btu}$) and $N = 500$ and find Q_s at the point where the index line intercepts the Q_s scale.
- 2-Draw an index line through the value of Q_s found above and $\Delta t = 25^\circ\text{F}$ and find K (dummy variable) at the point where the index line intercepts the pivot line.
- 3-Draw an index line through the value of K found above and $V_S = 230 \text{ cfm}$ and find the answer on the "AU" scale = 900 (or $AC = \frac{900}{500} = 1.8$)

Percentage of error:

$$\% \text{ error} = \frac{1.8 - 1.7654}{1.7654} \times 100 + \frac{0.0346 \times 100}{1.7654} = +1.96\%$$

From the above applications, it can be said that Parker's nomograph seems to be accurate enough for most design problems. The degree of expected error due to the assumption of a constant v_i value equal to $\frac{12.39}{0.24}$ will be presented in the discussion of the writer's nomograph.

NOMOGRAPH FOR DETERMINING WINTER VENTILATING RATES

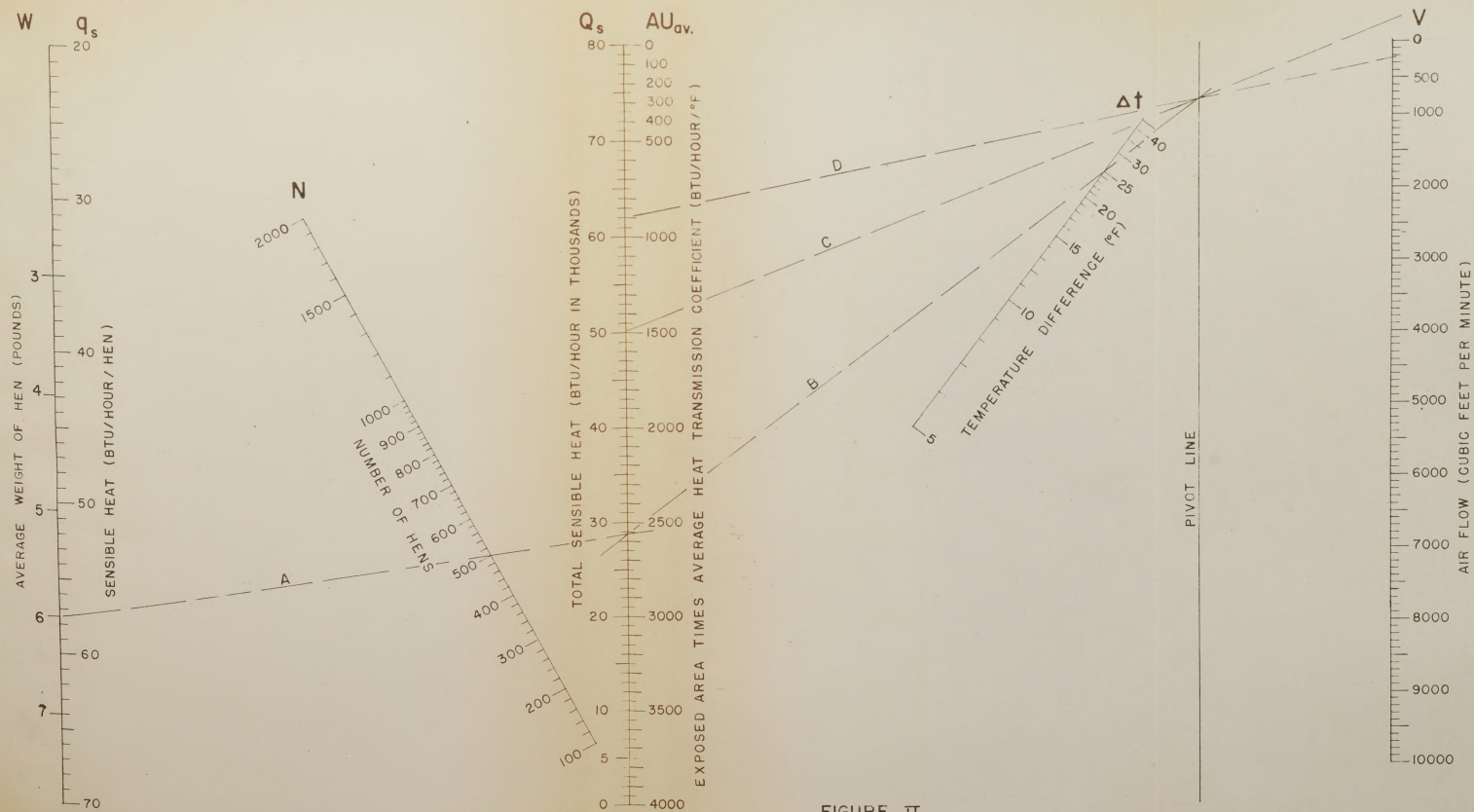


FIGURE II

b. Choinière nomograph for determining the ventilation rate required for temperature control. After analyzing Parker's nomograph for temperature control, the writer thought of some ways of improving this nomograph. Since it was desired to obtain the rate of ventilation per hen, the first equation used in Parker's nomograph ($Q_s = H \times q_s$) was not considered. The terms of the second equation were modified as shown here:

$$Q_s = \frac{60 V_s D}{52} + \text{"AC"} D \text{ or } 60 V_s = \frac{52 Q_s}{D} - 52 \text{"AC"}$$

where Q_s = Total sensible heat production per hen
(Btu/Hr, hen)

60 = Used to reduce the ventilation rate from one
hour to one minute

V_s = Ventilation rate required for temperature
control (cfm/Hen)

D = Temperature difference ($^{\circ}\text{F}$)

$52 = \frac{v}{0.24} = \frac{12.48}{0.24}$ (where v is the specific volume of
one pound of moist air at approximately 33°F and
80% R.H., and 0.24 is the specific heat of air
at constant pressure) = Number of cubic feet
of air raised one $^{\circ}\text{F}$ by one Btu.

"AC" = Heat loss characteristic of the building
(Btu/Hr, $^{\circ}\text{F}$, hen)

This equation was separated into two equal parts each containing two variables and made equal to a third variable (dummy variable), for instance
K:

$$1) K = \frac{52 Q_s}{D}$$

$$2) 60 V_s = K - 52 \text{"AC"}$$

The theory of the arrangement of the different scales and their spacing will be found in any textbook on nomographs (46).

The writer computed the percentage of expected error caused by the assumption of a constant v_1 value. Computations are shown here for both $v_1 = 52$ (as used in this nomograph) and $v_1 = 53$ (as used by Parker and many other designers).

t_1	$v_1(80\%)$	$\frac{v_1(80\%)}{0.24}$	Difference		Expected error (%)	
			1-Diff. from 52	2-Diff. from 53	1-Use of 52 (Writer's nomograph)	2-Use of 53 (Parker's nomograph)
80	13.98	58.25	+6.25	+5.25	-12.0	-10.5
75	13.78	57.42	+5.42	+4.42	-10.4	-8.5
70	13.61	56.71	+4.71	+3.71	-9.0	-7.1
65	13.44	56.00	+4.00	+3.00	-7.7	-5.8
60	13.28	55.33	+3.33	+2.33	-6.4	-4.5
55	13.12	54.66	+2.66	+1.66	-5.1	-3.19
50	12.97	54.04	+2.04	+1.04	-3.92	-2.0
45	12.82	53.42	+1.42	+0.42	-2.73	-0.81
40	12.67	52.82	+0.82	-0.38	-1.19	+0.73
35	12.53	52.21	+0.21	-0.79	-0.40	+1.52
32	12.45	51.87	-0.13	-1.13	+0.25	+2.17
30	12.39	51.62	-0.38	-1.38	+0.73	+2.65
25	12.25	51.04	-0.96	-1.96	+1.84	+3.77
20	12.12	50.50	-1.50	-2.50	+2.88	+4.8
15	11.990	49.96	-2.04	-3.04	+3.92	+5.8
10	11.855	49.37	-2.63	-3.63	+5.0	+7.0
5	11.722	48.83	-3.17	-4.17	+6.1	+8.0
0	11.59	48.28	-3.72	-4.72	+7.15	+8.9

< 4% error

As shown in the table, such a nomograph would give an error as much as 4 to 12% for computations involving the use of inside temperatures from 50 to 80°F. However, for the range of temperatures from 15 to 50°F, the amount of expected error is below 4% and the nomograph can be used safely for this range of inside temperature, when a greater accuracy is not required, as is the case with most design problems.

To illustrate the use of this nomograph, consider the following applications:

Application No. 1Given: Sensible heat production per hen = 45 Btu/Hr

Desired indoor temperature = 40°F

Design outdoor temperature = 0°F

AC value of the house = 1.0

Wanted: The maximum rate of ventilation required to maintain the above temperature difference (40 - 0 = 40°F)True mathematical solution: v_i (40°F, 80%) = 12.67 (From Table 44)

$$\begin{aligned}
 V_s &= \frac{v_i}{0.24} \left(\frac{Q_s}{\Delta t} - AC \right) \frac{1}{60} \\
 &= \frac{12.67}{0.24} \left(\frac{45}{40} - 1.0 \right) \frac{1}{60} \\
 &= \frac{12.67}{0.24} (1.125 - 1.0) \frac{1}{60} = \frac{12.67 \times 0.125}{0.24 \times 60} = 0.11 \text{ cfm/Hen}
 \end{aligned}$$

Solution:1-Using a straight edge, join $Q_s = 45$ to $\Delta t = 40$ and find the pivoting point on the K scale2-Join $AC = 1.0$ and the pivoting point found above and find $V = 0.11$ cfm/Hen on the V scale.Percentage of error:

$$\% \text{ error} = \left(\frac{0.11 - 0.110}{0.110} \right) 100 = 0\%$$

Application No. 2Given: Sensible heat production per hen = 45 Btu/Hr

Desired indoor temperature = 40°F

Desired outdoor temperature = 0°F

Average air infiltration rate = 0.20 cfm/Hen

Wanted: The "AC" value required for the house to maintain this temperature differential, with no ventilation other than the infiltration

Total mathematical solution:

$$v_1 \text{ (100°F, 80\%)} = 12.67 \text{ (From Table 141)}$$

$$AC = \frac{Q_s}{D} - \frac{60 V_s}{\left(\frac{v_1}{0.24}\right)}$$

$$= \frac{45}{40} - \frac{60 \times 0.20}{\left(\frac{12.67}{0.24}\right)}$$

$$= 1.125 - \frac{60 \times 0.20 \times 0.24}{12.67}$$

$$= 1.1250 - 0.2273$$

$$= 0.8977$$

Solution:

- 1-Using a straight edge, join $Q_s = 45$ to $\Delta t = 40$ and find the pivoting point on the K scale
- 2-Join $V = 0.20$ and the pivoting point found above and find $AC = 0.90$ on the "AC" scale.

Percentage of error:

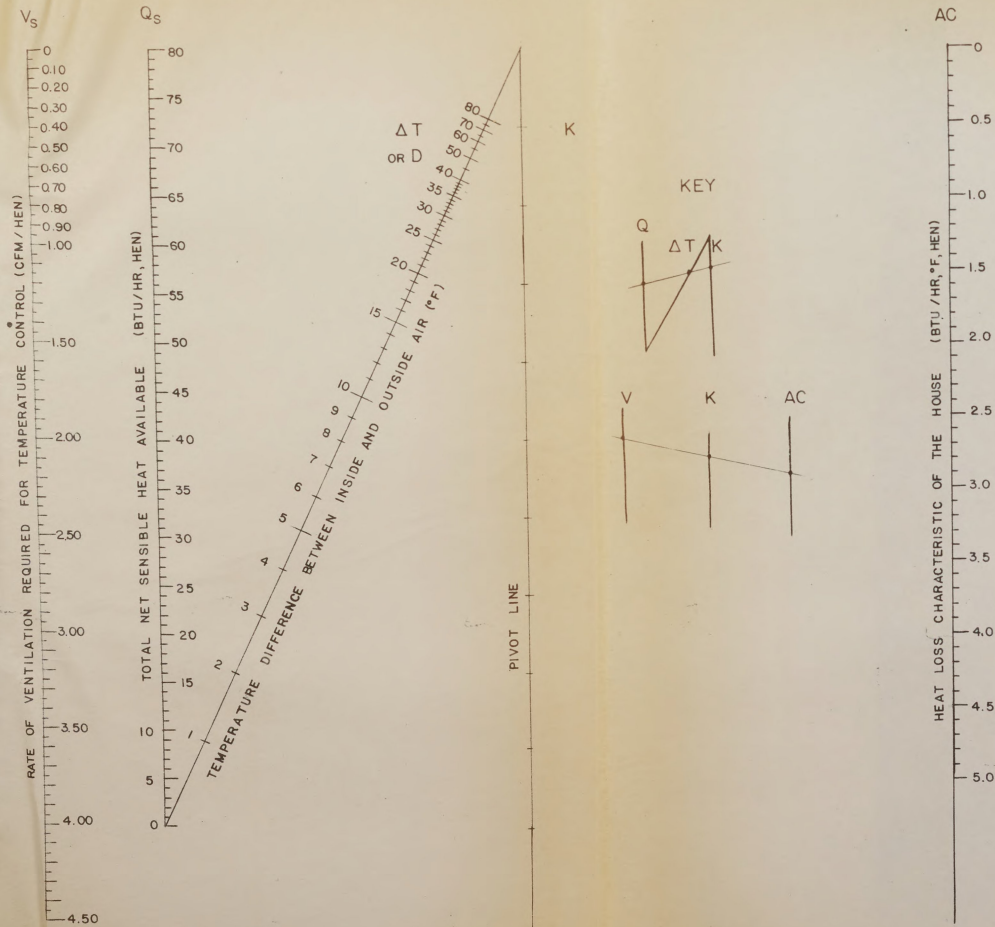
$$\% \text{ error} = \left(\frac{0.90 - 0.8977}{0.8977} \right) 100 = \frac{0.0023 \times 100}{0.8977} = +0.25\%$$

Some of the advantages of this nomograph over Parker's nomograph are:

- 1-The AC value of the house per hen is used instead of the total $AU_{\text{aver.}}$ value, which makes it easier to compare houses of the same capacity but of different insulating value.
- 2-The ventilation rate is given in cfm per hen which makes it very suitable for purposes of comparison and analysis. (To find the fan capacity for a given house, this value must be multiplied by the number of birds in the house, which is a simple operation.)
- 3-The slope of the Δt scale is not so great, making the reading easier.
- 4-The sensible heat from litter decomposition and from solar radiation may be added to the sensible heat from the birds.

From Parker's and the writer's nomographs for temperature control, it can be said that these nomographs are only approximate and because of the assumption of constant v_i value on which they are based, the range of expected error will be above 4% for inside temperatures outside the range of 15 to 50°F and 25 to 55°F respectively.

NOMOGRAPH FOR DETERMINING THE RATE OF VENTILATION REQUIRED FOR TEMPERATURE CONTROL



I. Poultry House Ventilation Composite Charts

Composite charts are another way of presenting the solution of a given formula. Their main advantage is that they usually require a smaller space than nomographs. They are similar to nomographs in that an equation is solved by drawing lines from one axis to another; however, their appearance is much different and for this reason the writer is presenting them apart from simple charts and from nomographs.

A composite chart is made by choosing two variables in the equation to be solve and placing them on a chart, one as the abscissa scale and the other as a graph line. The combination of these two independent variables yields a third dependent variable which is read on the ordinate scale. This dependent variable is in turn used as the ordinate scale on an adjoining chart in which another of the variables in the formula is presented as a graph line. This new combination yields a second dependent variable. The process is continued until all the variables in the equation have been used.

If two variables are to be added or subtracted, the graph lines will be straight and parallel. If they are to be divided or multiplied, the graph lines will be straight divergent lines or curved lines.

1. Rate of ventilation required for moisture removal

a. Goldrich composite chart for determining the ventilation rate required for moisture removal. Goldrich (36) presented in May 1953 a composite chart to solve the equation for moisture removal. This composite chart may be considered as being a combination of two equations each containing three variables:

$$1) \Delta W = W_i - W_o$$

where ΔW = Moisture carrying capacity of outside air

(Lb of moisture/Lb of dry air)

W_i = Moisture content of inside air (Lb of moisture/Lb dry air)

W_o = Moisture content of outside air (Lb of moisture/Lb dry air)

$$2) V_L = \frac{W_T}{60 \Delta W \times 100} = \frac{W_T}{6000 \Delta W}$$

where V_L = Ventilation rate required for moisture removal
(Lb of dry air/min, hen)

W_T = Rate of moisture production in the house
(Lb of moisture/Hr, 100 hens)

ΔW = Moisture carrying capacity of outside air
(Lb of moisture/Lb dry air)

60 = Used to reduce the ventilation rate from one hour to one minute

100 = Used to reduce the ventilation rate from one hundred birds to one bird

The combination of these two formulas gives: $V_L = \frac{W_T}{6000 (W_i - W_o)}$ which

is the standard equation for moisture removal. Since this formula includes four variables and may be divided into two equations involving three variables each (as shown above), two graphs are required to solve it for V_L . However, in order to find the values of W_i and W_o corresponding to the given inside and outside temperature and relative humidity conditions, a third graph is necessary which is a psychrometric chart with W as the ordinate, t as the abscissa and relative humidity as the graph line.

In order to simplify his composite chart, Goldrich (36) assumed that the volume of discharged air equal the volume of entering air. The above assumption bring the following changes in some of the symbols presented above:

V_L = Ventilation rate required for moisture removal

(cu.ft. of moist air/min, hen)

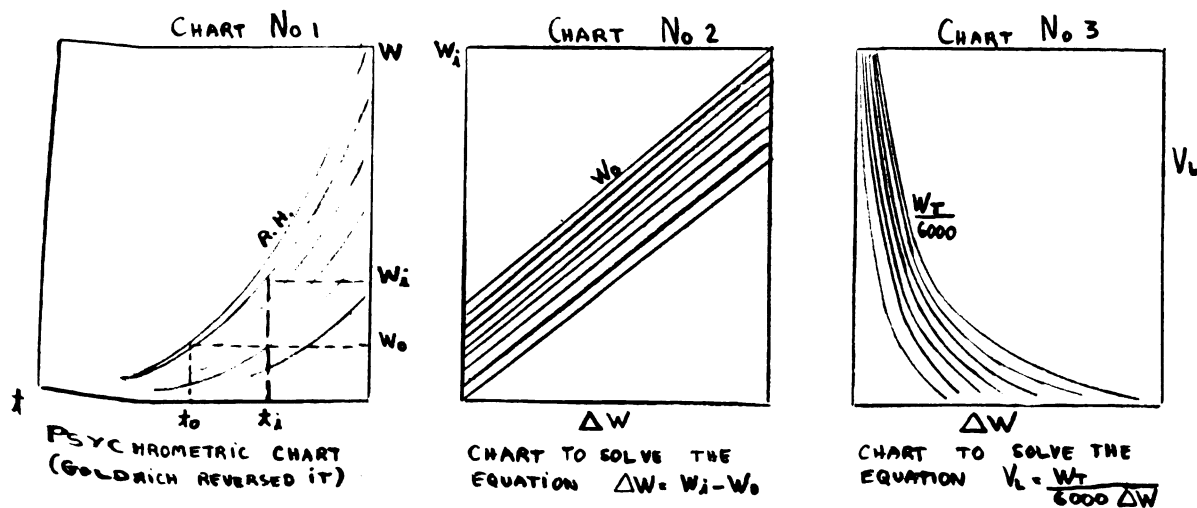
W_i = Moisture content of inside air (Lb of moisture/Cu.ft. of moist air)

W_o = Moisture content of outside air (Lb of moisture/Cu.ft. of moist air)

ΔW = Moisture carrying capacity of outside air (Lb of moisture/Cu.ft. of moist air)

He therefore presented a modified psychrometric chart giving W in pound per cubic foot of air instead of being in pound per pound of dry air as in the standard psychrometric charts

A sketch of the three graphs to be combined is shown here:



Since charts Nos. 2 and 3 have the same abscissa, they must be superimposed. Chart 1 is to the left and charts 2 and 3 are presented to the right.

An application is presented on the chart and is checked here:

Application (Refer to the chart)

Given: Inside temperature = 65°F
 Inside relative humidity = 87%
 Outside temperature = 25°F
 Outside relative humidity = 85%
 Rate of moisture production in the house = 1.25 lb/Hr, 50 birds
 (or 60 lb/Day, 100 birds) (or 2.5 lb/Hr, 100 hens)

Wanted: The rate of ventilation required for the complete removal of
 the moisture as soon as it is produced

True mathematical solution: (Heating, Ventilating, Air Conditioning Guide, 1951)

$$\begin{aligned}
 W_{i\text{sat}} (65^\circ\text{F}) &= 132.6 \times 10^{-4} \text{ lb/Lb dry air} \\
 W_i (65^\circ\text{F}, 87\%) &= 132.6 \times 87\% = 115.362 \times 10^{-4} \text{ lb/Lb dry air} \\
 W_{o\text{sat}} (25^\circ\text{F}) &= 27.33 \times 10^{-4} \text{ lb/Lb dry air} \\
 W_o (25^\circ\text{F}, 85\%) &= 27.33 \times 85 = 23.2305 \times 10^{-4} \text{ lb/Lb dry air} \\
 \Delta W &= 115.3620 - 23.2305 = 92.1315 \times 10^{-4} \text{ lb/Lb dry air} \\
 v_{i\text{act}} (65^\circ\text{F}, 87\%) &= 13.46 \text{ (Use of slide rule No 5)}
 \end{aligned}$$

$$\begin{aligned}
 \text{(a) } V_L &= \frac{W_T \times v_{i\text{act}}}{6000 \Delta W} = \frac{2.5 \times 13.46}{6000 \times 92.1315 \times 10^{-4}} = \frac{2.5 \times 13.46}{0.6 \times 92.1315} = 0.6087 \\
 &\text{cfm/Hen (or 1826.1 cfm/50 hens)}
 \end{aligned}$$

True mathematical solution: (Zimmerman and Lavine, 1945)

$$\begin{aligned}
 W_i (65^\circ\text{F}, 87\%) &= 115.362 \times 10^{-4} \text{ lb/Lb dry air} \\
 W_{o\text{sat}} (25^\circ\text{F}) &= 28.26 \times 10^{-4} \text{ lb/Lb dry air} \\
 W_o (25^\circ\text{F}, 85\%) &= 28.26 \times 85 = 24.021 \times 10^{-4} \text{ lb/Lb dry air} \\
 \Delta W &= 115.362 - 28.26 = 91.341 \times 10^{-4} \text{ lb/Lb dry air} \\
 v_{i\text{act}} (65^\circ\text{F}, 87\%) &= 13.46 \text{ (Use of slide rule No. 5)}
 \end{aligned}$$

$$(b) \quad V_L = \frac{W_T \times V_{\text{inlet}}}{6000 \Delta W} = \frac{2.5 \times 13.46}{6000 \times 91.341 \times 10^{-4}} = \frac{2.5 \times 13.46}{0.6 \times 91.341} = 0.614$$

cfm/Hen (or 1342 cfm/50 hens)

Solution by use of the composite chart:

Goldrich in his illustration (refer to the chart) found $V_S =$
1820 cfm/50 birds (or 0.6066 cfm/Hen)

Percentage of error:

$$(a) \quad \% \text{ error} = \left(\frac{0.6066 - 0.6087}{0.6087} \right) 100 = \frac{-0.00204 \times 100}{0.6087} = -0.336\%$$

$$(b) \quad \% \text{ error} = \left(\frac{0.6066 - 0.614}{0.614} \right) 100 = \frac{-0.0074 \times 100}{0.614} = -1.2\%$$

Based on this unique application, the accuracy of Goldrich's chart would be very good. However, it should be remembered that because of the assumption on which it is based, this chart is subject to an error. In the discussion of his nomograph, the writer has computed the degree of error to be expected. These values are summarized in the following table: ($W_m = 2$ lb/Hr, 100 hens or 48 lb/Day, 100 hens)

From this table it may be concluded:

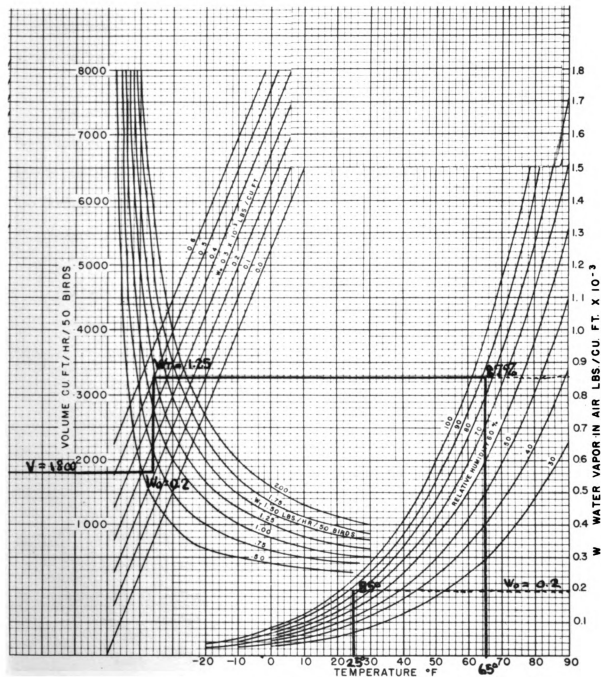
- 1-That the percentage of error for temperature differences smaller than 10° varies from about 10% up to infinity
- 2-That the percentage of error for temperature differences from 10 to 35° decreases from 10% to about 2% with an increase of the temperature difference
- 3-That the percentage of error for a temperature difference from 35 to 50° is around 2%.

Since Goldrich's application is for a temperature difference of 40°F , this probably explained the low percentage of error found. It is probable also that Goldrich (36) has used Zimmerman and Levine psychrometric table (1945) although he did not mention the source of his data.

t_i	R.H. _i	t_o	R.H. _o	Δt	Expected error (%)
60	70	55	80	5	+24.3%
		50	80	10	+ 8.9
		45	80	15	+ 6.6
		40	80	20	+ 5.5
50	80	45	90	5	+ 8.8
		35	90	15	+ 6.7
		25	90	25	+ 6.2
		15	90	35	+ 2.77
40	80	0	90	50	+ 1.51
		35	90	5	—
		25	100	15	+ 6.3%
		15	100	25	+ 3.88
30	80	5	100	35	+ 2.65
		0	100	40	+ 2.24
		25	100	5	—
		15	100	15	5.2%
20	90	5	100	25	3.6
		0	100	30	2.73
		-5	100	35	2.19
		15	100	5	—
		5	100	15	4.8%
		0	100	20	3.82
		-5	100	25	2.78
		-15	100	35	1.935

To conclude, it should be said that the above composite chart is only approximate; it gives its greater accuracy when large temperature differences (35 to 50°F) are involved. It should probably never be used for small temperature differences (10° and less).

GOLDRICH COMPOSITE CHART FOR DETERMINING THE VENTILATION RATE REQUIRED FOR MOISTURE REMOVAL (36)



255

255

b. Choiinidre composite chart for moisture removal. The writer prepared a second type of composite chart which is based on the same formulas than the ones used by Goldrich (36) and is using the modified psychrometric chart prepared by Goldrich (36).

One advantage of this new composite chart over Goldrich's is that the ventilation rate is read in cfm per bird, a value easier to use for comparative analysis. However, because it is based, like Goldrich's chart, on the assumption that the volume of entering air is equal to the volume of discharged air, it is subject to the same errors discussed for the previous chart.

Two applications are presented on the chart and are checked here:

Application No. 1 (Refer to the chart)

Given: Inside temperature = 40°F
 Inside relative humidity = 80%
 Outside temperature = 0°F
 Outside relative humidity = 100%
 Rate of water production in the house = 45 lb/Day, 100 hens
 (or 1.875 lb/Hr, 100 hens)

Wanted: The rate of ventilation required to remove all the moisture as soon as it is produced.

True mathematical solution: (Heating, Ventilating, Air Conditioning Guide, 1951)

$$W_{i\text{sat}} (40^\circ\text{F}) = 52.13 \times 10^{-4} \text{ lb/Lb dry air}$$

$$W_{i\text{act}} (40^\circ\text{F}, 80\%) = 52.13 \times 80\% = 41.704 \times 10^{-4} \text{ lb/Lb dry air}$$

$$W_{o\text{sat}} (0^\circ\text{F}) = 7.872 \times 10^{-4} \text{ lb/Lb dry air}$$

$$\Delta W = 41.704 - 7.872 = 33.832 \times 10^{-4} \text{ lb/Lb dry air}$$

$$v_{i\text{act}} (40^\circ\text{F}, 80\%) = 12.67 \text{ (From Table 44)}$$

$$(a) \quad V_L = \frac{W_T \times v_{i\text{act}}}{6000 \Delta W} = \frac{1.875 \times 12.67}{6000 \times 33.832 \times 10^{-4}} = \frac{1.875 \times 12.67}{0.6 \times 33.832} = 1.115 \text{ cfm/Hen}$$

True mathematical solution: (Zimmerman and Lavine, 1945)

$$W_{i_{act}} (100^\circ\text{F}, 80\%) = 11.704 \times 10^{-4} \text{ lb/Lb dry air}$$

$$W_{o_{sat}} (^\circ\text{F}) = 9.322 \times 10^{-4} \text{ lb/Lb dry air}$$

$$\Delta W = 11.704 - 9.322 = 32.382 \times 10^{-4} \text{ lb/Lb dry air}$$

$$v_{i_{act}} (100^\circ\text{F}, 80\%) = 12.67 \text{ (From Table 44)}$$

$$(b) \quad V_L = \frac{W_T \times v_{i_{act}}}{6000 \Delta W} = \frac{1.875 \times 12.67}{6000 \times 32.382 \times 10^{-4}} = \frac{1.875 \times 12.67}{0.6 \times 32.382} = 1.167 \text{ cfm/hen}$$

Solution by use of the composite chart:

By referring to the composite chart, the answer is found to be

1.20 cfm/hen. (N.B. = The writer had first read it 1.25 but

later revised it and found it to be 1.20 which shows how difficult

it is to accurately read a composite chart)

Percentage of error:

$$(a) \quad \% \text{ error} = \left(\frac{1.20 - 1.115}{1.115} \right) 100 = \frac{0.085 \times 100}{1.115} = +7.6\%$$

$$(b) \quad \% \text{ error} = \left(\frac{1.20 - 1.167}{1.115} \right) 100 = \frac{0.033 \times 100}{1.115} = +2.95\%$$

Since the true mathematical solution using Zimmerman and Lavine data (1945) gives a percentage of error closer to the expected error, the writer believes that Goldrich probably used Zimmerman and Lavine psychrometric table although he does not mention the source of his data.

Application No. 2 (Refer to the chart)

Given: Inside temperature = 50°F

Inside relative humidity = 80%

Outside temperature = 20°F

Outside relative humidity = 100%

Rate of water production in the house = 45 lb/Day, 100 hens

(or 1.875 lb/Hr, 100 hens)

Wanted: The rate of ventilation required to remove all the moisture as soon as it is produced.

True mathematical solution: (Sinneman and Lavine, 1945)

$$W_{i\text{sat}} (50^\circ\text{F}) = 76.55 \times 10^{-4} \text{ lb/Lb dry air}$$

$$W_{i\text{act}} (50^\circ\text{F}, 80\%) = 76.55 \times 80\% = 61.24 \text{ lb/Lb dry air}$$

$$W_{o\text{sat}} (20^\circ\text{F}) = 22.87 \times 10^{-4} \text{ lb/Lb dry air}$$

$$\Delta W = 61.24 - 22.87 = 38.37 \text{ lb/Lb dry air}$$

$$v_{i\text{act}} (50^\circ\text{F}, 80\%) = 12.97 \text{ (From Table L4)}$$

$$V_L = \frac{W_T \times v_{i\text{act}}}{6000 \Delta W} = \frac{1.875 \times 12.97}{6000 \times 38.37 \times 10^{-4}} = \frac{1.875 \times 12.97}{0.6 \times 38.37} = 1.065 \text{ cfm/Man}$$

Solution by use of the composite chart:

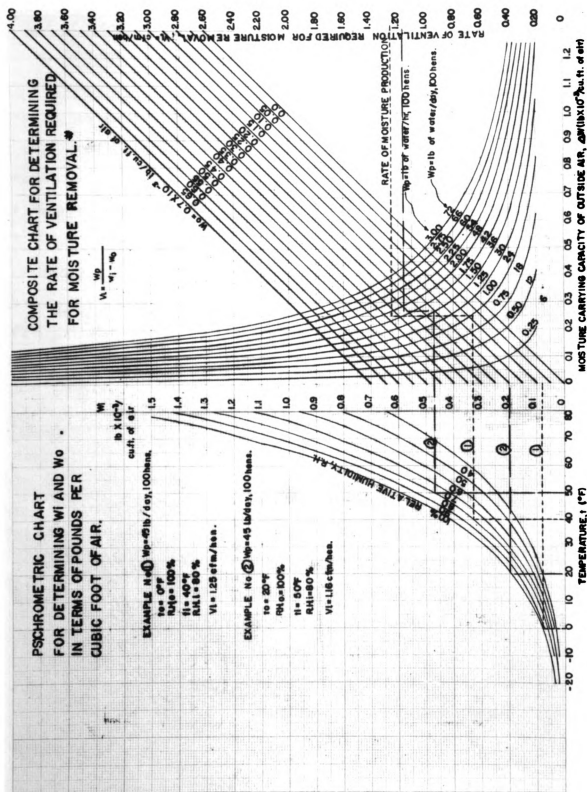
By referring to the chart, the answer is found to be 1.10 cfm/Man

(N.B. = The writer had first read it 1.16 but later revised it and found it to be 1.10, which shows again that composite charts are very difficult to read exactly).

Percentage of error:

$$\% \text{ error} = \left(\frac{1.10 - 1.065}{1.065} \right) 100 = \frac{0.035 \times 100}{1.065} = +3.28\%$$

To conclude from Goldrich's composite chart for moisture removal and the similar chart presented by the writer, it can be said that they are only approximate and great care should be taken in their use. The writer believes that nomographs and slide rules should be preferred since they are easier to read accurately and are therefore more reliable.



* Prepared by the writer

21-11-13

2. Rate of ventilation required for temperature control

a. Goldrich composite chart for determining the ventilation rate required for temperature control. Goldrich (36) also prepared a composite chart to solve the equation for temperature control. This composite chart may be considered as being a combination of six equations each containing three variables:

- 1) $H_{Lat.} = H_{AT} \times L$
- 2) $H_S(B) = H_{AT} - H_{Lat.}$
- 3) $H_S(BL) = H_S(B) \pm H_L$
- 4) $H_S(BLS) = H_S(BL) \pm H_S$
- 5) $K = \frac{53 H_S(BLS)}{\Delta t} = \frac{53 Q_S \text{ Total}}{\Delta t}$
- 6) $V_S = K - 53 AC$

where $H_{Lat.}$ = Latent heat produced by the bird
(Btu/Hr, 50 birds)

H_{AT} = Total Heat given off by the respiratory system
and body of the bird (Mitchell and Kelley's
curve) (Btu/Hr, 50 birds)

L = Percentage of latent heat produced by the birds
(Mitchell and Kelley's curve) (%)

$H_S(B)$ = Sensible heat produced by the birds
(Btu/Hr, 50 birds)

$H_S(BL)$ = Sensible heat from the birds + Net sensible
heat from litter decomposition
(Btu/Hr, 50 birds)

$H_S(BLS) = Q_S \text{ Total} =$ Sensible heat from the birds + Net
sensible heat from litter decomposition + Heat
from the sun, lights, etc. = Total sensible
heat available for temperature control
(Btu/Hr, 50 birds)

H_L = Net sensible heat from litter decomposition

(This is the portion of heat from litter decomposition which is in excess of that used to vaporize the moisture in the litter).

(Btu/Hr, hen)

H_S = Heat from the sun, lights, etc. (Btu/Hr, hen)

K = Dummy variable

$53 = \frac{v}{0.24}$ = Cubic feet of air raised one °F by one Btu.

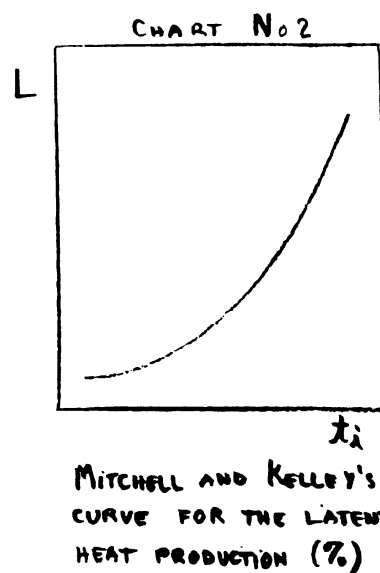
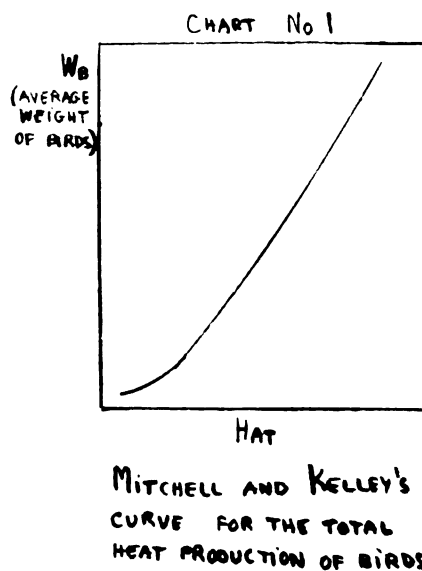
$t = t_i - t_o$ = Temperature difference between inside and outside air (°F)

V_S = Ventilation rate required for temperature control (cfh/50 birds)

AC = Heat loss characteristic of the house
(Btu/Hr, °F, 50 birds)

50 = To convert the rate of ventilation from one to 50 birds

The combination of these formulas gives $V_S = \frac{53 Q_S \text{ Total}}{\Delta t} - 53 AC$ which is the standard equation for temperature control. A sketch of the seven graphs to be combined is shown here:



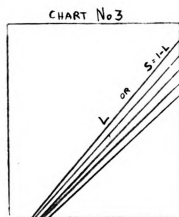


CHART TO SOLVE THE
EQUATION $H_{s(B)} = H_{AT} \times S$

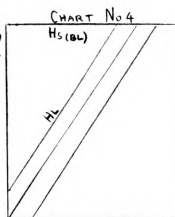


CHART TO SOLVE THE
EQUATION $H_{s(B)} = H_{s(B)} + H_L$

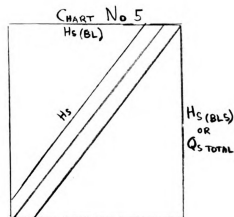


CHART TO SOLVE THE
EQUATION $H_{s(BL)} = H_{s(B)} + H_s$

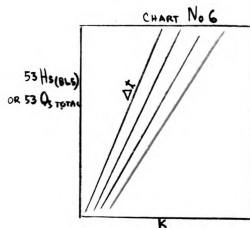


CHART TO SOLVE THE
EQUATION $K = \frac{53 Q_{s, TOTAL}}{\Delta T}$

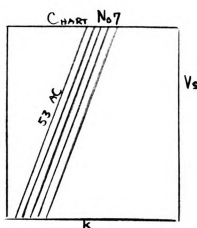


CHART TO SOLVE THE
EQUATION $V_s = K - 53 AC$

An application is presented on the chart and is checked here:

Application (Refer to the chart)

- Given:
- Inside temperature = 65°F
 - Outside temperature = 25°F
 - Inside relative humidity = 87%
 - Weight of birds = 5 lb
 - Total heat production (H_{AT}) = According to Mitchell and Kelley's curve.
 - Percentage of latent heat (L) = According to Mitchell and Kelley's curve.

Net sensible heat from litter decomposition (H_L) = 5 Btu/Hr, hen

Heat from the sun, lights, etc (H_S) = 5 Btu/Hr, hen

Heat loss characteristic of the house (AC) = 30 Btu/Hr, °F, 50 hens

Wanted: The ventilation rate required to maintain the above temperature difference ($65^\circ - 25^\circ = 40^\circ$)

True mathematical solution:

$L = 24\%$ (From Mitchell's curve)

$S = 100 - 24 = 76\%$

$H_{AT} = 2315$ Btu/Hr, 50 birds (From Mitchell's curve)

$H_S(B) = 2315 \times 76\% = 1759.4$

$H_L = 5$ Btu/Hr, bird or 250 Btu/Hr, 50 birds

$H_S = 5$ Btu/Hr, bird or 250 Btu/Hr, 50 birds

$H_{S(Total)} = 1759.4 + 250 + 250 = 1759.4 + 500 = 2259.4$ Btu/Hr, 50 birds

$v_{i_{act}} = 13.45$ cu.ft/Lb of dry air (By use of sliderule No 5)

$$\begin{aligned} V_S &= \frac{v_i}{0.24} \left(\frac{Q_s \text{ Total}}{\Delta t} - AC \right) \frac{1}{60} \\ &= \frac{13.45}{0.24} \left(\frac{2259.4}{40} - 30 \right) \frac{1}{60} \\ &= 56.04 (56.485 - 30) \frac{1}{60} = \frac{56.04 \times 26.485}{60} = 24.737 \text{ cfm/50 birds} \\ &\quad (\text{or } 0.49454 \text{ cfm/bird}) (\text{or } 1184.2 \text{ cfm/ 50 birds}) \end{aligned}$$

Solution by use of the composite chart:

Goldrich in his illustration (refer to the chart) found

$V_S = 1820$ cfm/50 birds or 30.33 cfm/50 birds

(or 0.6066 cfm/bird)

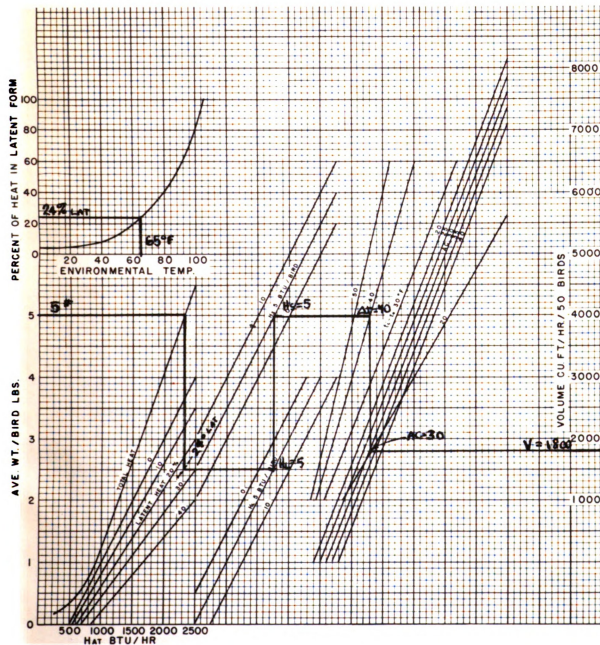
Percentage of error:

$$\% \text{ error} = \left(\frac{0.60666 - 0.49454}{0.49454} \right) 100 = \frac{11.212 \times 100}{0.49454} = 22.7\%$$

After checking this application and finding the big percentage of error involved, the writer wrote to Goldrich concerning the big difference. Goldrich answered that: "the temperature control chart needs correction but will not be done as Mitchell and Kelley's work (percent graph and Btu vs weight) has been made obsolete by some improved data which was completed by the U.S.P.A. at Beltsville by Ota. The chart should be revised using Ota's data but I do not have the time to do so immediately."



GOLDRICH COMPOSITE CHART FOR DETERMINING THE VENTILATION RATE REQUIRED FOR TEMPERATURE CONTROL (36)



5263

1000 1000

b. Thoinière composite chart for determining the ventilation rate required for temperature control. Since the Goldrich composite chart for temperature control seems to give considerable error, the writer prepared a new chart. This new chart may be considered as being a combination of two equations each containing three variables:

$$1) K = \frac{52 Q_s}{\Delta t}$$

where K = Dummy variable

Q_s = Total sensible heat available (Btu/Hr, hen)

Δt = Temperature differential ($^{\circ}\text{F}$)

$52 = \frac{v}{0.24} = \frac{12.48}{0.24}$ (where v is the specific volume of one pound of moist air at approximately 33°F and 80% R.H. and 0.24 is the specific heat of air at constant pressure) = Cubic feet of air raised 1°F by 1 Btu.

$$2) 60 V_s = K - 52 \text{ "AC"}$$

where V_s = Rate of air flow required for temperature control (Cfm/hen)

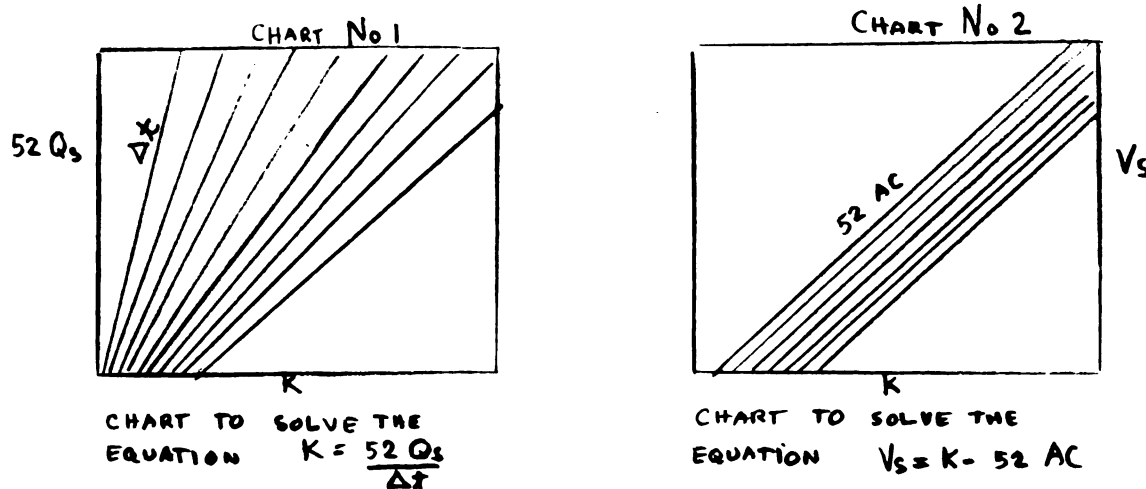
K = Dummy variable

$52 = \frac{v}{0.24} = \frac{12.48}{0.24}$ = Cubic ft of air raised 1°F by 1 Btu.

"AC" = Heat loss characteristic of the building (Btu/Hr, $^{\circ}\text{F}$, hen)

60 = Number of minutes in an hour

The combination of these two formulas gives $60 V_S = \frac{52 Q_s}{\Delta t} - 52 AC$ which is the standard equation for temperature control. Sketches of the two graphs to be combined are shown here:



Since charts Nos. 1 and 2 have the same abscissa, they must be superimposed.

Two applications are presented on the chart and are checked here.

Application No. 1 (Refer to the chart)

Given: Total sensible heat available for temperature control =
40 Btu/Hr, hen

Inside temperature = 40°F

Inside relative humidity = 80%

Outside temperature = 10°F

Heat loss characteristic of the house (AC) = 1.0

Wanted: The rate of ventilation required to maintain the above temperature differential (40 - 10 = 30°)

True mathematical solution:

$$v_{i_{act}} (40^\circ\text{F}, 80\%) = 12.67 \quad (\text{From Table 44})$$

$$\begin{aligned}
 V_S &= \frac{v_{iact}}{0.24} \left(\frac{Q_s - AC}{D} \right) \frac{1}{60} \\
 &= \frac{12.67}{0.24} \left(\frac{40 - 1}{30} \right) \frac{1}{60} = \frac{12.67}{0.24} (1.333 - 1) \frac{1}{60} \\
 &= \frac{12.67 \times 0.333}{0.24 \times 60} = 0.292 \text{ cfm/Hen}
 \end{aligned}$$

Solution by use of the composite chart:

Referring to the composite chart, the answer is found to be
0.30 cfm/Hen

Percentage of error:

$$\% \text{ error} = \left(\frac{0.30 - 0.292}{0.292} \right) 100 = + \frac{0.008 \times 100}{0.292} = +2.7\%$$

Application No. 2 (Refer to the chart)

Given: Total sensible heat available for temperature control =

40 Btu/Hr, hen

Inside temperature = 40°F

Inside relative humidity = 80%

Outside temperature = 30°F

Heat loss characteristic of the house (AC) = 1.0

Wanted: The rate of ventilation required to maintain the above temperature differential (40 - 30 = 10°)

True mathematical solution:

v_{iact} (40°F, 80%) = 12.67 (From Table 44)

$$\begin{aligned}
 V_S &= \frac{v_{iact}}{0.24} \left(\frac{Q_s}{D} \right)^{-AC} \frac{1}{60} \\
 &= \frac{12.67}{0.24} \left(\frac{40}{10} - 1 \right) \frac{1}{60} = \frac{12.67}{0.24} (4 - 1) \frac{1}{60} \\
 &= \frac{12.67 \times 3}{0.24 \times 60} = 2.64 \text{ cfm/Hen}
 \end{aligned}$$

Solution by use of the composite chart:

By referring to the composite chart, the answer is found to be

2.6 cfm/bird

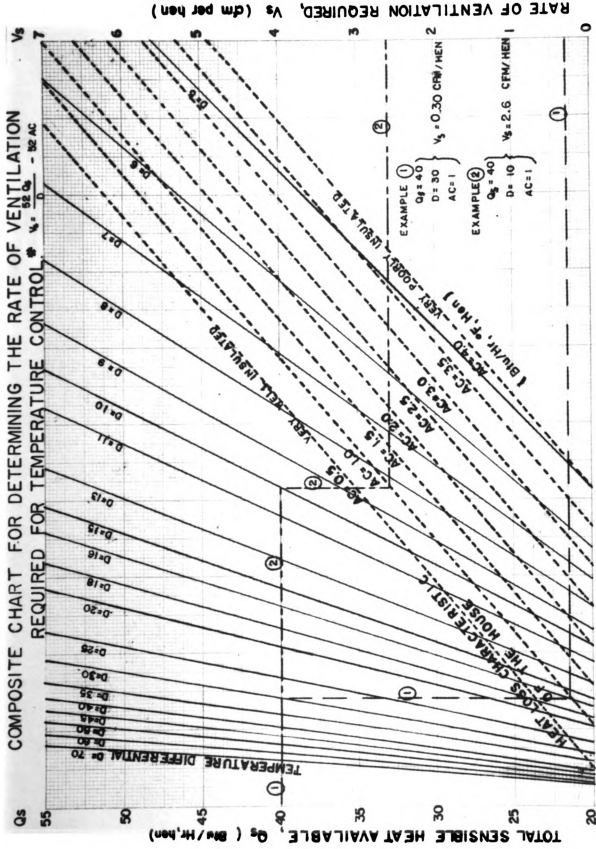
Percentage of error:

$$\% \text{ error} = \left(\frac{2.6 - 2.64}{2.64} \right) 100 = \frac{-0.04 \times 100}{2.64} = -1.52\%$$

One advantage of this composite chart over Goldrich's is that the ventilation rate is read in cfm per bird which is a value easier to use for comparative analysis. Another advantage is that this chart is applicable to any condition and the sensible heat from litter decomposition and from the sun may be added to the sensible heat from the birds.

To conclude from Goldrich's composite chart for temperature control and the similar one presented by the writer, the writer found out that they are somewhat more complicated to read than nomographs because of the intersecting lines. For this reason, the writer's conclusion is that nomographs and slide rules should be preferred.





J. Poultry House Ventilation Slide Rules

Composite charts and nomograms have some advantages but they also have some disadvantages: they usually require very large charts and a straight edge for the solution. For this reason the writer prepared some slide rules to solve the moisture removal formula and the temperature control formula.

1. Slide rules for determining the ventilation rate required for moisture removal

a. Approximate design slide rule, arbitrarily connected. This slide rule (slide rule No 4) is based on an outside relative humidity of 100 % and is made of the combination of three equations:

$$1) W_i = W_{i\text{sat.}} \times \frac{R.H._i}{100}$$

where W_i = Moisture content of inside air, at given relative humidity (Lb. of moisture/Cu.ft. of moist air)

$W_{i\text{sat.}}$ = Moisture content of inside air, at saturation (100% R.H.) (Lb. of moisture/Cu. ft. of moist air)

R.H. = Percentage of relative humidity inside the poultry house

100 = Factor used to bring R.H._i to a decimal value

$$2) \Delta W = W_i - W_{o\text{sat.}}$$

where ΔW = Moisture carrying capacity of outside air (Lb. of moisture/Cu.ft. of moist air)

W_i = Moisture content of inside air, at given relative humidity (Lb. of moisture/Cu.ft. of moist air)

$W_{o\text{sat.}}$ = Moisture content of outside air at design
relative humidity (100% R.H.) (lb. of
moisture/Cu.ft. of moist air)

$$3) V_L = \frac{W_T}{100 \times 60 \times \Delta W} = \frac{W_T}{6000 \Delta W}$$

where V_L = Ventilation rate required for moisture removal
(Cu.ft. of moist air/min, hen)

W_T = Rate of moisture production in the house
(lb. of moisture/hr, 100 hens)

100 = Used to reduce the ventilation rate from 100 to
one hen

60 = Used to reduce the ventilation rate from one hour
to one minute

ΔW = Moisture carrying capacity of outside air
(lb. of moisture/Cu.ft. of moist air)

Equation 1 was changed to a logarithmic form:

$$\begin{aligned} \log W_i &= \log W_{i\text{sat.}} + \log \text{R.H.} \\ + (\log W_{i\text{sat.}}) - (0) + (\log \text{R.H.}) - (\log W_i) &= (0) \end{aligned}$$

The log of $W_{i\text{sat.}}$ was plotted and the values of t_i were substituted for them.

A middle support was required on the slide rule for conversion of log
 W_i values to W_i values. Equation 2 was arranged in the following order

$$+ (W_i) - (0) + (-W_o) - (\Delta W) = (0)$$

The values of W_o were plotted and the values of t_o were substituted for them.

A second middle support was required for conversion of W values to log W
values. Equation 3 was changed to the following logarithmic form:

$$\log V_L - \log W_R - (\log 6000 \Delta W) \\ + (\log 6000 \Delta W) - (0) + (-\log W_R) - (-\log V_L) = (0)$$

The location of the third arrow was slightly lowered in order to partially correct the assumption made by taking the volume of entering air as equal to the volume of outgoing air.

Reference to any textbook on nomographs and slide rules (46) will explain how the slide rule was prepared from the above equations.

In order to illustrate the use of slide rule No 4, consider the following applications for inside temperatures of 20, 30, 40 and 50°F.

Application No. 1 ($t_i = 20^\circ\text{F}$)

Given: Rate of water production in the house = 1.875 lb/hr, 100 hens
(or 45 lb/Day, 100 hens)

Indoor temperature = 20°F

Indoor relative humidity = 85%

Outdoor temperature = -20°F

Outdoor relative humidity = 100%

Wanted: The minimum rate of ventilation required under the above condition to remove all the moisture produced.

True mathematical solution: (Heating, Ventilating, Air Conditioning Guide, 1945)

$$W_{\text{sat.}} (20^\circ\text{F}) = 21.44 \times 10^{-4} \text{ lb/Lb dry air}$$

$$W_i 85\% (20^\circ\text{F}) = 21.44 \times 0.85 = 18.214 \times 10^{-4} \text{ lb/Lb dry air}$$

$$W_{\text{osat.}} (-20^\circ\text{F}) = 2.625 \times 10^{-4} \text{ lb/Lb dry air}$$

$$\Delta W = 18.214 - 2.625 = 15.589 \text{ lb/Lb dry air}$$

$$v_{\text{iact}} (20^\circ\text{F}, 85\%) = 12.12 \text{ (Use of slide rule No 5)}$$

$$V_L = \frac{W_T \times v_{\text{iact}}}{6000 \Delta W} = \frac{1.875 \times 12.12}{6000 \times 15.589 \times 10^{-4}} = \frac{1.875 \times 12.12}{0.6 \times 15.589} = 2.429 \text{ cfm/Hen}$$

Solution by use of the corrected slide rule: (From left to right)

1-Put the first arrow opposite $t_i = 20^\circ\text{F}$ and read $W_i = 15$

opposite R.H. $= 85\%$

2-Put the second arrow opposite $W_i = 15$ and read $\Delta W = 12.75$

opposite $t_o = -20^\circ\text{F}$

3-Put the third arrow opposite $\Delta W = 12.75$ and read $V_L = 2.42$ cfm/Hen

(2.485 non-corrected) opposite $W_T = 1.875$ (or 45)

Percentage of error:

$$\% \text{ error} = \left(\frac{2.42 - 2.429}{2.429} \right) 100 = \frac{-0.009 \times 100}{2.429} = -0.37\%$$

as compared to +2.3 % for the non-corrected slide rule.

Application No. 2 ($t_i = 30^\circ\text{F}$)

Given: Rate of water production in the house = 1.875 lb/Hr, 100 hens
(or 45 lb/day, 100 hens)

Indoor temperature = 30°F

Indoor relative humidity = 90%

Outdoor temperature = 10°F

Outdoor relative humidity = 100%

Wanted: The minimum rate of ventilation required under the above condition to remove all the moisture produced.

True mathematical solution: (Heating, Ventilating, Air Conditioning Guide, 1945)

$$W_{i\text{sat.}} (30^\circ\text{F}) = 34.39 \times 10^{-4} \text{ lb/Lb dry air}$$

$$W_i 90\% (30^\circ\text{F}) = 34.39 \times 0.90 = 30.951 \times 10^{-4} \text{ lb/Lb dry air}$$

$$W_{o\text{sat.}} (10^\circ\text{F}) = 13.11 \times 10^{-4} \text{ lb/Lb dry air}$$

$$\Delta W = 30.951 - 13.11 = 17.84 \times 10^{-4} \text{ lb/Lb dry air}$$

$$V_{i\text{act.}} (30^\circ\text{F}, 80\% \text{ R.H.}) = 12.39 \text{ (Use of slide rule No 5)}$$

$$V_L = \frac{W_i \times V_{i,sat}}{4000} = \frac{1.875 \times 12.39}{4000 \times 17.84 \times 10^{-4}} = \frac{1.875 \times 12.39}{0.6 \times 17.84} = 2.1703 \text{ cfm/hen}$$

Solution by use of the corrected slide rule: (From left to right)

1-Put the first arrow opposite $t_i = 30^\circ\text{F}$ and read $W_i = 24.8$

opposite R.H._i = 80%

2-Put the second arrow opposite $W_i = 24.8$ and read $W = 13.85$

opposite $t_o = 10^\circ\text{F}$

3-Put the third arrow opposite $\Delta W = 13.85$ and read $V_L = 2.22$

(2.285 non-corrected) opposite $W_T = 1.875$ (or 15)

Percentage of error:

$$\% \text{ error} = \left(\frac{2.22 - 2.1703}{2.1703} \right) 100 = \frac{0.0497 \times 100}{2.17} = +2.31\%$$

as compared to +5.2% for the non-corrected slide rule.

Application No. 3 ($t_i = 40^\circ\text{F}$)

Given: Rate of water production in the house = 1.875 lb/hr, 100 hens
(or 15 lb/day, 100 hens)

Indoor temperature = 40°F

Indoor relative humidity = 80%

Outdoor temperature = 0°F

Outdoor relative humidity = 100%

Wanted: The minimum rate of ventilation required under the above condition to remove all the moisture produced.

True mathematical solution: (Heating, Ventilating, Air Conditioning Guide, 1945)

$$W_{i,sat.} (40^\circ\text{F}) = 51.94 \times 10^{-4} \text{ lb/lb dry air}$$

$$W_i \text{ 80\% } (40^\circ\text{F}) = 51.94 \times 0.80 = 41.552 \times 10^{-4} \text{ lb/lb dry air}$$

$$W_{o,sat.} (0^\circ\text{F}) = 7.852 \times 10^{-5} \text{ lb/lb dry air}$$

$$\Delta W = 41.552 - 7.852 = 33.7 \times 10^{-4} \text{ lb/lb dry air}$$

$$v_{i,act} (40^\circ F, 80\% R.H.) = 12.67 \text{ (From Table 44)}$$

$$V_L = \frac{W_T \times v_{i,act}}{6000 \Delta W} = \frac{1.875 \times 12.67}{6000 \times 33.7 \times 10^{-4}} = \frac{1.875 \times 12.67}{0.6 \times 33.7} = 1.1719 \text{ cfm/hen}$$

Solution by use of the corrected slide rule: (From left to right)

1-Put the first arrow opposite $t_i = 40^\circ F$ and read $W_i = 32.75$

opposite R.H._i = 80%

2-Put the second arrow opposite $W_i = 32.75$ and read $\Delta W = 24.2$

opposite $t_o = 0^\circ F$

3-Put the third arrow opposite $\Delta W = 24.2$ and read $V_L = 1.17$

(1.20 non-corrected) opposite $W_T = 1.875$ (or 45)

Percentage of error:

$$\% \text{ error} = \left(\frac{1.17 - 1.1719}{1.1719} \right) \times 100 = \frac{-0.0019 \times 100}{1.1719} = -0.16\%$$

as compared to +2.14% for the non-corrected slide rule.

Application No. 4 ($t_i = 50^\circ F$)

Given: Rate of water production in the house = 1.875 lb/hr, 100 hens
(or 45 lb/day, 100 hens)

Indoor temperature = $50^\circ F$

Indoor relative humidity = 80%

Outdoor temperature = $20^\circ F$

Outdoor relative humidity = 100%

Wanted: The minimum rate of ventilation required under the above conditions to remove all the moisture produced.

True mathematical solution: (Heating, Ventilating, Air Conditioning Guide, 1945)

$$W_{i, sat.} (50^\circ F) = 76.26 \times 10^{-4} \text{ lb/lb dry air}$$

$$W_i 80\% (50^\circ F) = 76.26 \times 0.80 = 61.008 \text{ lb/lb dry air}$$

$$W_{\text{osat.}} (20^{\circ}\text{F}) = 21.44 \times 10^{-4} \text{ lb/lb dry air}$$

$$\Delta W = 41.008 - 21.44 = 39.568 \times 10^{-4} \text{ lb/lb dry air}$$

$$v_{\text{act}} (50^{\circ}\text{F}, 80\%) = 12.97 \text{ (from Table 44)}$$

$$V_L = \frac{W_T \times v_{\text{act}}}{6000 \Delta W} = \frac{1.875 \times 12.97}{6000 \times 39.568 \times 10^{-4}} = \frac{1.875 \times 12.97}{0.6 \times 39.568} = 1.0243 \text{ cfm/Hen}$$

Solution by use of the corrected slide rule: (From left to right)

1-Put the first arrow opposite $t_i = 50^{\circ}\text{F}$ and read $W_i = 47.1$

opposite R.H._i = 80%

2-Put the second arrow opposite $W_i = 47.1$ and read $\Delta W = 29.7$

opposite $t_o = 20^{\circ}\text{F}$.

3-Put the third arrow opposite $\Delta W = 29.7$ and read $V_L = 1.043$

(1.07 non-corrected) opposite $W_T = 1.875$ (or 45)

Percentage of error:

$$\% \text{ error} = \left(\frac{1.043 - 1.0243}{1.0243} \right) 100 = \frac{0.0187 \times 100}{1.0243} = +1.83\%$$

(as compared to +1.46% for the non-corrected slide rule)

In order to check more thoroughly the accuracy of this slide rule

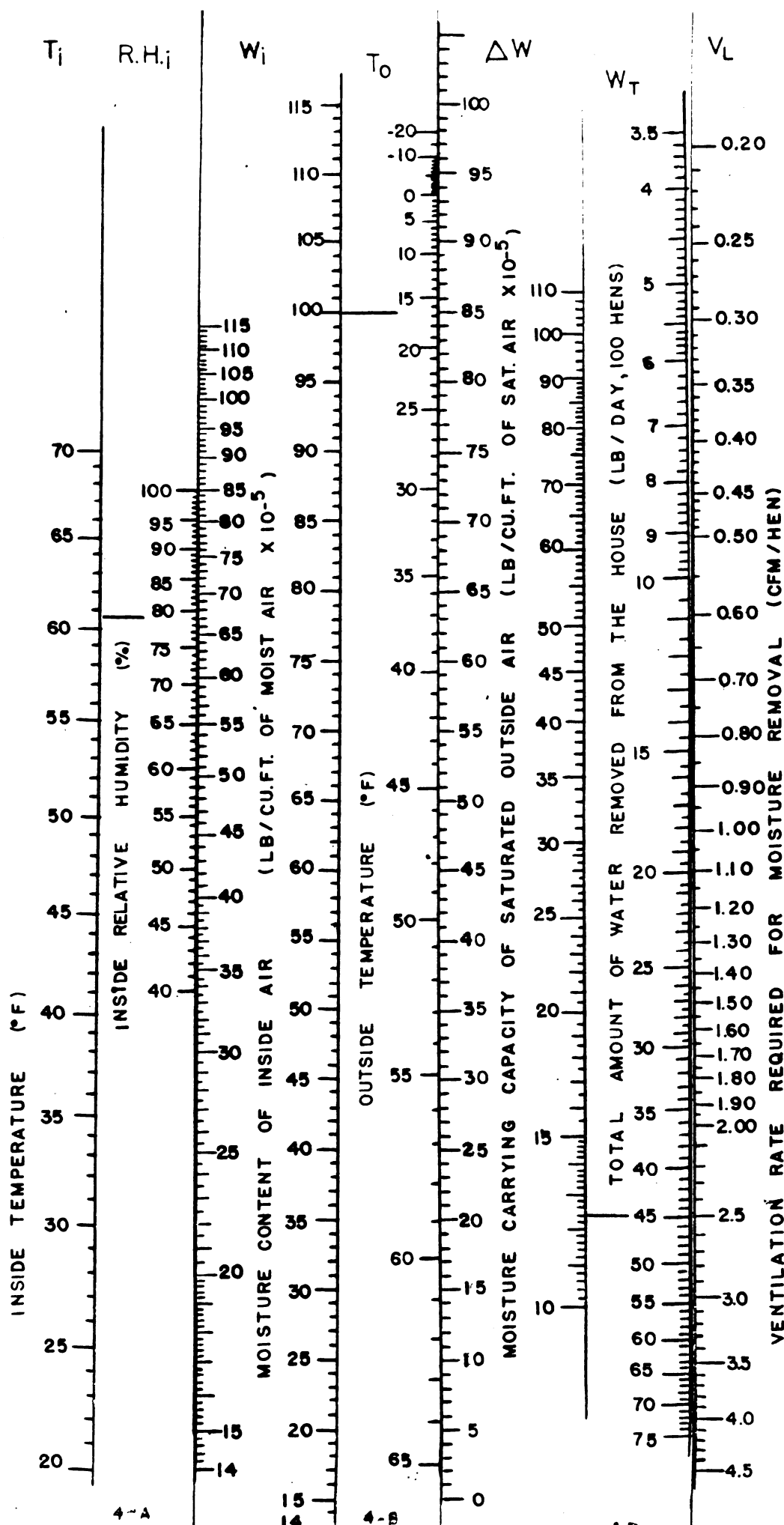
the writer computed 10 additional applications: ($W_T = 2.0 \text{ lb/Hr}$, 100 hens
or 1.8 lb/Day , 100 hens)

Check	t_i	R.H. _i	W_{act}	t_o	R.H. _o	ΔW	W_T	V_L	Diff.	% Error
								Use of slide rule	True math. solut.	
No. 1	50	80	47.1	20	100	29.7	2.0	1.118	1.087	+0.031 +2.85%
No. 2	50	80	47.1	10	100	36.3	2.0	0.91	0.897	+0.013 +1.45
No. 3	50	80	47.1	0	100	40.75	2.0	0.815	0.81	+0.005 +0.62
No. 4	50	80	47.1	-10	100	43.4	2.0	0.765	0.762	+0.003 +0.39
No. 5	40	80	32.75	10	100	22	2.0	1.50	1.478	+0.022 +1.49
No. 6	40	80	32.75	0	100	26.2	2.0	1.25	1.251	-0.001 -0.08
No. 7	40	80	32.75	-10	100	28.85	2.0	1.145	1.14	+0.005 +0.44
No. 8	30	80	22.25	10	100	11.6	2.0	2.84	2.78	+0.06 +2.16
No. 9	30	80	22.25	0	100	15.8	2.0	2.085	2.05	+0.035 +1.71
No. 10	30	80	22.25	-10	100	18.5	2.0	1.785	1.763	+0.022 +1.25

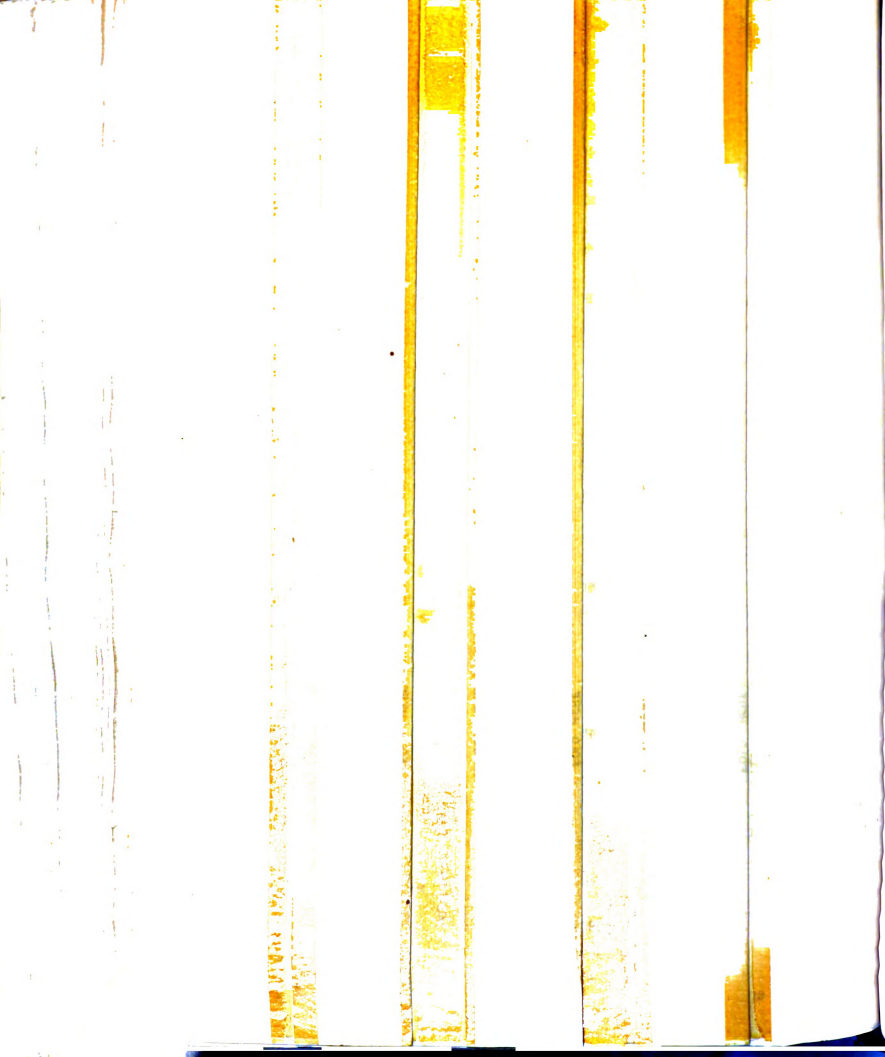
From the above applications and checks, it can be said that slide rule No. 4 seems to maintain an accuracy within 2.5% for the whole range of inside temperature from 20° to 50°F, and for this reason may be considered as given a very satisfactory approximation.

A few of the advantages of slide rule No. 4 are:

- 1-Little space is needed.
- 2-No straight edge is required.
- 3-No psychrometric chart of any kind is needed.
- 4-The ventilation rate is given in cfm per hen which makes it very suitable for purpose of comparison.
- 5-The accuracy seems to be within 2.5% for the whole range of inside temperature from 20° to 50°F.



SLIDE RULE FOR DETERMINING THE RATE OF VENTILATION REQUIRED FOR MOISTURE REMOVAL



b. Exact design slide rule. In all the composite charts, nomographs, and slide rules heretofore presented for solving the moisture removal formula, the results were only approximate because of the use of some kind of modification of the true formula for moisture removal. These modifications were made for purpose of simplification. It can be said in general that in most cases the results given by the above tools are not too different from the true mathematical answers but the errors do not always have the same magnitude and cannot be predicted, as has been shown in the applications illustrating their use.

Of all the tools mentioned above, the corrected slide rule just presented (slide rule No 4) is probably the most accurate and the most satisfactory. However, in an effort to reduce still more the differences from the true mathematical solution, the writer prepared a new slide rule (slide rule No 6) based on the true mathematical formula (unmodified). This formula is:

$$V_L = \left(\frac{W_T}{60 \times 100} \right) \div \left(\frac{W_i - W_o}{v_{i \text{ act}}} \right) = \frac{W_T \times v_{i \text{ act}}}{6000 (W_i - W_o)} = \frac{W_T \times v_{i \text{ act}}}{6000 \Delta W}$$

where V_L = Rate of ventilation required for moisture removal (cfm/Hen)

W_T = Water to be removed from the house (Lb/Hr, 100 hens)

W_i = Moisture content of the inside air at the given temperature and relative humidity (Lb/Lb dry air)

W_o = Moisture content of the outside air at the given temperature and relative humidity (Lb/Lb dry air)

ΔW = Moisture carrying capacity of the outside air (Lb/Lb dry air)

v_{iact} = Specific volume of the mixture of one pound of dry air and the actual weight of moisture held by it at the house temperature and relative humidity (Cu.ft/Lb dry air) (From Table 44 or the writer's slide rule for the specific volume of moist air)

60 = To reduce the water production to one minute

100 = To reduce the water production to one hen

Since v_{iact} is affected by both the temperature and the relative humidity inside the poultry houses, its value must be found in appropriate tables, if available (the writer found no table giving them readily for any combination of temperature and relative humidity), or else may be computed by the use of a special slide rule prepared by the writer (slide rule No 5). Since v_{iact} must be known as a preliminary step in the use of the exact slide rule for moisture removal, slide rule No 5 will be explained first.

Slide rule No 5 (for determining the specific volume of a mixture of dry air and water vapor) can be considered as being a combination of the two following formulas:

$$1) \Delta v_{i_{sat}} \times R.H._1 = \Delta v_{iact}$$

$$\text{and } 2) \Delta v_{iact} + v_{i_{dry}} = v_{iact}$$

where $\Delta v_{i_{sat}}$ = Volume occupied by the saturated moisture held by one pound of dry air at the house temperature (Cu.ft/Lb dry air)

$R.H._1$ = Percentage of relative humidity inside of the house (5)

- $\Delta v_{i\text{act}}$ = Volume occupied by the actual (unsaturated) moisture held by one pound of dry air at the house temperature (Cu.ft/Lb dry air)
- $v_{i\text{dry}}$ = Specific volume of one pound of dry air at the house temperature (Cu.ft/Lb dry air)
- $v_{i\text{act}}$ = Specific volume of the mixture of one pound of dry air and the actual weight of moisture held by it at the house temperature and relative humidity (Cu.ft/Lb dry air)

Equation 1 was changed to a logarithmic form:

$$+(\log v_{i\text{sat}}) - (0) + (\log \text{R.H.}_i) - (\log v_{i\text{act}}) = (0)$$

The values of $\log \Delta v_{i\text{sat}}$ were plotted from Heating, Ventilating, Air Conditioning Guide, 1945, and the values of t_i were substituted for them.

A middle support was required on the slide rule for conversion of $\log \Delta v_{i\text{act}}$ values to $\Delta v_{i\text{act}}$ values. Equation 2 was arranged in the following order:

$$+(\Delta v_{i\text{act}}) - (0) + (v_{i\text{dry}}) - (v_{i\text{act}}) = (0)$$

The values of $v_{i\text{dry}}$ were plotted from Heating, Ventilating, Air Conditioning Guide, 1945, and the values of t_i were substituted for them.

Reference to any textbook on nomographs and slide rules (16) will explain how the slide rule was prepared from the above equations.

In order to make possible a comparison of the results given by slide rule No 5 with the true mathematical results, the writer computed the following table (Table 44) by the use of the standard thermodynamic formulas for the psychrometric volume occupied by a mixture of one pound of dry air and various amounts of water vapor. Dalton's Law of Partial Pressures which states

that: "the volume of a mixture of gases is the same as the volume occupied by each gas at its partial pressure" was used to make these computations. An illustration of the type of computation required is presented here.

Application of the standard thermodynamic formulas
for the psychrometric volume of moist air

Given: Air temperature = 50°F
 Percentage of relative humidity = 70%
 Barometer pressure = 14.696 psia

Wanted: The specific volume of the mixture (dry air and water vapor)
 per pound of dry air

Solution:

a) Use of the partial pressure of water vapor

The formula to use here is:

$$v_g = \frac{W \times R_w \times T}{p_g''}$$

where v_g = Specific volume of water vapor at partial
 pressure p_g'' (Cu.ft/Lb)

W = Specific humidity of air at the given relative
 humidity (Lb/Lb of dry air)

R_w = Gas constant of water vapor = 85.7
 (ft-lb per lb per deg R)

T = Absolute temperature $460 + ^\circ\text{F} = (^\circ\text{R})$

p_g'' = Partial pressure of water vapor (psi)

Step 1: Find the partial pressure of the saturated vapor

Refer to a Steam table. For $t = 50^\circ\text{F}$, $p_g'' = 0.17811$ psi

Step 2: Compute the partial pressure at the given relative humidity

$0.17811 \times 70\% = 0.124677$ psi

Step 3: Find the specific humidity of saturated air

Refer to a Psychrometric table. For $t = 50^{\circ}\text{F}$,

$$W_{\text{sat}} = 0.007626 \text{ lb of water vapor/Lb of dry air}$$

Step 4: Find the specific humidity at the given relative humidity

$$0.007626 \times 70\% = 0.0053382 \text{ lb of water vapor/Lb of dry air}$$

Step 5: Apply the formula

$$\begin{aligned} v_g &= \frac{W \times R_w \times T}{p_g''} \\ &= \frac{0.0053382 \times 85.7 \times (460 + 50)}{14.4 \times 0.124677} \\ &= 12.95 \end{aligned}$$

b) Use of the partial pressure of dry air

The formula to use here is:

$$v_a = \frac{w R_a T}{p_a}$$

where v_a = Specific volume of dry air at partial pressure

p_a (Cu.ft/Lb)

w = Unit weight of dry air = 1 lb (lb)

R_a = Gas constant for air = 53.3

T = Absolute temperature = $460 + ^{\circ}\text{F} = (^{\circ}\text{R})$

p_a = Partial pressure of dry air (psi)

Step 1: Find the partial pressure of the saturated vapor

$$p_g'' = 0.17811 \text{ psi (as found above)}$$

Step 2: Compute the partial pressure at the given relative humidity

$$0.17811 \times 70\% = 0.124677 \text{ psi}$$

Step 3: Compute the partial pressure of dry air

$$14.696 - 0.124677 = 14.5713 \text{ psi}$$

Step 4: Apply the formula

$$v_a = \frac{w R_a T}{p_a} = \frac{1 \times 53.3 \times (460 + 50)}{14.4 \times 14.5713} = 12.95$$

Table 44. Specific volume of wet air (cu.ft./lb dry air)*

R.H.	Temperature (°F)										
	32°	35°	40°	45°	50°	55°	60°	65°	70°	75°	80°
100%	12.46	12.55	12.69	12.84	13.00	13.16	13.32	13.50	13.68	13.86	14.08
95	12.46	12.54	12.68	12.84	12.99	13.15	13.31	13.48	13.66	13.85	14.06
90	12.45	12.54	12.68	12.83	12.98	13.14	13.30	13.47	13.65	13.83	14.03
85	12.45	12.54	12.68	12.82	12.97	13.13	13.29	13.46	13.63	13.81	14.01
80	12.45	12.53	12.67	12.82	12.97	13.12	13.28	13.44	13.61	13.78	13.98
75	12.44	12.53	12.67	12.81	12.96	13.11	13.27	13.43	13.60	13.77	13.96
70	12.44	12.52	12.66	12.80	12.95	13.10	13.25	13.41	13.58	13.75	13.93
65	12.44	12.52	12.66	12.80	12.94	13.09	13.24	13.40	13.56	13.73	13.90
60	12.43	12.51	12.65	12.79	12.93	13.08	13.23	13.38	13.54	13.71	13.88
55	12.43	12.51	12.64	12.78	12.93	13.07	13.22	13.37	13.53	13.69	13.86
50	12.42	12.50	12.64	12.78	12.92	13.06	13.21	13.36	13.51	13.67	13.83
45	12.42	12.50	12.63	12.77	12.91	13.05	13.19	13.34	13.50	13.65	13.81
40	12.42	12.50	12.63	12.77	12.90	13.04	13.18	13.33	13.48	13.63	13.79

* Computed from the standard thermodynamic formulas shown above

The writer made a certain number of checks. These are presented here.

t_i	R.H. _i	$v_{i\text{act}}$	
		1-Mathematical solution (Table 44)	2-Slide rule No 5
40°	100%	12.69	12.69
40°	80%	12.67	12.67
40°	70%	12.66	12.66
40°	60%	12.65	12.65
40°	40%	12.63	12.63
45°	80%	12.82	12.81*
50°	80%	12.97	12.96*
60°	70%	13.25	13.25
65°	95%	13.48	13.48
65°	75%	13.43	13.43
70°	100%	12.68	12.68

* Slight deviation (negligible)

As a conclusion of these checks, slide rule No 5 (for determining the specific volume of air and water vapor mixtures was found extremely satisfactory.

Slide rule No 6 (for determining the rate of ventilation required for moisture removal) can be considered as being a combination of the four following formulas

$$1) W_{i\text{sat}} \times \text{R.H.}_i = W_{i\text{act}}$$

$$2) W_{i\text{act}} - W_{o\text{sat}} + \Delta W$$

$$3) K = \frac{W_T}{100 \Delta W}$$

$$4) V_L = K \times \left(\frac{v_{i\text{act}}}{60} \right)$$

The combination of these four equations gives the true mathematical formula for moisture removal:

$$V_L = K \times \left(\frac{v_{i\text{act}}}{60} \right) = \frac{\left(\frac{W_T}{60 \times 100} \right) \times v_{i\text{act}}}{\Delta W} = \frac{\left(\frac{W_T}{60 \times 100} \right) \times v_{i\text{act}}}{W_{i\text{act}} - W_{o\text{sat}}} = \text{cfm/Hen}$$

In all these formulas the meaning of the terms are as follow:

$W_{i\text{sat}}$ = Water vapor content of saturated air at inside temperature (Lb/Lb dry air)

R.H._i = Relative humidity of inside air (%)

$W_{i\text{act}}$ = Water vapor content of moist air (unsaturated) at inside temperature (Lb/Lb dry air)

$W_{o\text{sat}}$ = Water vapor content of saturated air at outside temperature (Lb/Lb dry air)

ΔW = Water carrying capacity of one pound of outside air (Lb/Lb dry air)

K = Durrty variable = $\frac{W_T}{100 \Delta W}$

V_L = Ventilation rate required for moisture removal (cfm/Hen)

$v_{i\text{act}}$ = Specific volume of the mixture of one pound of dry air and the actual weight of moisture hold by it at the house temperature and relative humidity (Cu.ft/Lb dry air)

Equation 1 was changed to a logarithmic form:

$$+(\log W_{i\text{sat}}) - (0) + (\log R.H._i) - (\log W_{i\text{act}}) = (0)$$

The values of $\log W_{i\text{sat}}$ were plotted and values of t_i were substituted for them.

A middle support was required for conversion of $\log W_{i\text{act}}$ values to

W_{iact} values. Equation 2 was arranged in the following order:

$$+(W_{iact}) - (0) + (W_{osat}) - (\Delta W) = (0)$$

A second middle support was required for conversion of W values to $\log W$ values. The third equation was changed to a logarithmic form:

$$+(\log 100 \Delta W) - (0) + (-\log W_p) - (+\log K) = (0)$$

A third middle support was required but the same $\log K$ scale was used. The fourth equation was changed to a logarithmic form:

$$+(\log K) - (0) + \left(\frac{\log v_i}{60}\right) - (\log V_L) = (0)$$

Reference to any textbook on nomographs and slide rules (46) will explain how the slide rule was prepared from the above equations.

Since (a) the last scale for v_{iact} is very small, (b) the variation of the value of v_{iact} for a given temperature is very little, and (c) 80% relative humidity is the most common relative humidity percentage encountered in poultry houses, the writer thought of modifying the last scale by substituting t_i values for the v_{iact} values corresponding to 80% relative humidity. Such modification does not impair the accuracy of the slide rule very much and there is no need of finding v_{iact} as a preliminary step. Using this t_i scale instead of the v_{iact} scale the same degree of accuracy is obtained for the three applications given above. (Accuracy within 0.01 cfm per hen from the true mathematical solution).

Values of W_{isat} and W_{osat} were taken from the psychrometric table published in Heating, Ventilating, Air Conditioning Guide, 1951 (6).

In order to illustrate the use of slide rule No 6, consider the following applications for inside temperatures of 20, 30, 40 and 50°F.

Application No. 1 ($t_i = 20^\circ\text{F}$)

Given: Rate of water production in the house = 1.875 lb/hr, 100 hens
(or 45 lb/Day, 100 hens)

Indoor temperature = 20°F

Indoor relative humidity = 85%

Outdoor temperature = -20°F

Outdoor relative humidity = 100%

Wanted: The minimum rate of ventilation required under the above conditions to remove all the moisture produced.

True mathematical solution: (Heating, Ventilating, Air Conditioning Guide, 1951)

$$W_{\text{sat.}} (20^{\circ}\text{F}) = 21.52 \times 10^{-4} \text{ lb/Lb dry air}$$

$$W_i \text{ 85\% } (20^{\circ}\text{F}) = 21.52 \times 0.85 = 18.290 \times 10^{-4} \text{ lb/Lb dry air}$$

$$W_{\text{sat.}} (-20^{\circ}\text{F}) = 2.630 \times 10^{-4} \text{ lb/Lb dry air}$$

$$\Delta W = 18.290 - 2.630 = 15.66 \text{ lb/Lb dry air}$$

$$v_{\text{act}} (20^{\circ}\text{F}, 85\%) = 12.12 \text{ (Use of slide rule No. 5)}$$

$$V_L = \frac{W_T \times v_{\text{act}}}{6000 \Delta W} = \frac{1.875 \times 12.12}{6000 \times 15.66 \times 10^{-4}} = \frac{1.875 \times 12.12}{0.6 \times 15.66} = 2.419 \text{ cfm/Hen}$$

Solution by use of the exact slide rule: (From left to right)

1-Put the first arrow opposite $t_i = 20^{\circ}\text{F}$ and read $W_{\text{act}} = 18.3$
opposite R.H.i = 85%

2-Put the second arrow opposite $W_{\text{act}} = 18.3$ and read $\Delta W = 15.5$
opposite $t_o = -20^{\circ}\text{F}$

3-Put the third arrow opposite $\Delta W = 15.5$ and read $K = 120$
opposite $W_T = 1.875$ (or 45)

4a-Put the fourth arrow opposite $K = 120$ and read $V_L = 2.41 \text{ cfm/Hen}$
opposite $t_i = 20^{\circ}\text{F}$

4b-Put the fourth arrow opposite $K = 120.5$ and read $V_L = 2.41 \text{ cfm/Hen}$
opposite $v_{\text{act}} = 12.12$ (from Table 141 or slide rule No. 5)

Percentage of error:

$$\text{a. and b. \% error} = \left(\frac{2.41 - 2.419}{2.419} \right) 100 = -\frac{0.009 \times 100}{2.419} = 0.37\%$$

Application No. 2 ($t_i = 30^\circ\text{F}$)

Given: Rate of water production in the house = 1.875 lb/Hr, 100 hens
 (or 45 lb/Day, 100 hens)
 Indoor temperature = 30°F
 Indoor relative humidity = 90%
 Outdoor temperature = 10°F
 Outdoor relative humidity = 100%

Wanted: The minimum rate of ventilation required under the above conditions to remove all the moisture produced.

True mathematical solution: (Heating, Ventilating, Air Conditioning Guide, 1951)

$$\begin{aligned}
 W_{i\text{sat.}} (30^\circ\text{F}) &= 34.54 \times 10^{-4} \text{ lb/Lb dry air} \\
 W_i 90\% (30^\circ\text{F}) &= 34.54 \times 0.90 = 31.086 \times 10^{-4} \text{ lb/Lb dry air} \\
 W_{o\text{sat.}} (10^\circ\text{F}) &= 13.15 \times 10^{-4} \text{ lb/Lb dry air} \\
 \Delta W &= 31.086 - 13.15 = 17.936 \times 10^{-4} \text{ lb/Lb dry air} \\
 v_{i\text{act}} (30^\circ\text{F}, 90\%) &= 12.39 \text{ (Use of slide rule No. 5)} \\
 V_L &= \frac{W_T \times v_{i\text{act}}}{6000 \Delta W} = \frac{1.875 \times 12.39}{6000 \times 17.93 \times 10^{-4}} = \frac{1.875 \times 12.39}{0.6 \times 17.93} = 2.15 \text{ cfm/Hen}
 \end{aligned}$$

Solution by use of the exact slide rule: (From left to right)

- 1-Put the first arrow opposite $t_i = 30^\circ\text{F}$ and read $W_{i\text{act}} = 30.80$
 opposite R.H._i = 90%
- 2-Put the second arrow opposite $W_{i\text{act}} = 30.80$ and read $\Delta W = 17.7$
 opposite $t_o = 10^\circ\text{F}$
- 3-Put the third arrow opposite $\Delta W = 17.7$ and read $K = 106$
 opposite $W_T = 1.875$ (or 45)
- 4a-Put the fourth arrow opposite $K = 106$ and read $V_L = 2.17$ cfm/Hen
 opposite $t_i = 30^\circ\text{F}$

4b-Put the fourth arrow opposite $K = 100$ and read $V_L = 2.17$ cfm/Hen
opposite $v_{iact} = 12.39$ (from Table 44 or slide rule No. 5)

Percentage of error:

$$a. \text{ and } b. \quad \% \text{ error} = \left(\frac{2.17 \times 2.15}{2.15} \right) 100 = \frac{+0.02 \times 100}{2.15} = +0.93\%$$

APPLICATION No 3 ($t_i = 40^\circ\text{F}$)

Given: Rate of water production in the house = 1.875 lb/Hr, 100 hens
(or 15 lb/Day, 100 hens)

Indoor temperature = 40°F

Indoor relative humidity = 80%

Outdoor temperature = 0°F

Outdoor relative humidity = 100%

Wanted: The minimum rate of ventilation required under the above conditions to remove all the moisture produced.

True mathematical solution: (Heating, Ventilating, Air Conditioning Guide, 1951)

$$W_{isat} (40^\circ\text{F}) = 52.13 \times 10^{-4} \text{ lb/Lb dry air}$$

$$W_i 80\% (40^\circ\text{F}) = 52.13 \times 0.80 = 41.704 \times 10^{-4} \text{ lb/Lb dry air}$$

$$W_{osat} (0^\circ\text{F}) = 7.872 \times 10^{-4}$$

$$\Delta W = 41.704 - 7.872 = 33.832$$

$$v_{iact} (40^\circ\text{F}, 80\%) = 12.67 \text{ (from Table 44)}$$

$$V_L = \frac{W_T \times v_{iact}}{6000 \Delta W} = \frac{1.875 \times 12.67}{6000 \times 33.832 \times 10^{-4}} = \frac{1.875 \times 12.67}{0.6 \times 33.832} = 1.1703 \text{ cfm/Hen}$$

Solution by use of the exact slide rule: (From left to right)

1-Put the first arrow opposite $t_i = 40^\circ\text{F}$ and read $W_{iact} = 41.65$
opposite R.H._i = 80%

2-Put the second arrow opposite $W_{iact} = 41.65$ and read $\Delta W = 33.75$
opposite $t_o = 0^\circ\text{F}$

3-Put the third arrow opposite $\Delta W = 33.75$ and read $K = 55.6$

opposite $W_T = 1.875$ (or 45)

4a-Put the fourth arrow opposite $K = 55.6$ and read $V_L = 1.18$ cfm/Hen

opposite $t_i = 40^\circ\text{F}$

4b-Put the fourth arrow opposite $K = 55.6$ and read $V_L = 1.18$ cfm/Hen

opposite $v_{i\text{act}} = 12.67$ (from Table 44 or slide rule No. 5)

Percentage of error:

$$\text{a. and b. } \% \text{ error} = \left(\frac{1.18 - 1.1703}{1.1703} \right) 100 = \frac{+0.0097 \times 100}{1.1703} = +0.83\%$$

Application No. 4 ($t_i = 50^\circ\text{F}$)

Given: Rate of water production in the house = 1.875 lb/Hr, 100 hens

(or 45 lb/Day, 100 hens)

Indoor temperature = 50°F

Indoor relative humidity = 80%

Outdoor temperature = 20°F

Outdoor relative humidity = 100%

Wanted: The minimum rate of ventilation required under the above conditions to remove all the moisture produced.

True mathematical solution: (Heating, Ventilating, Air Conditioning Guide, 1951)

$$W_{i\text{sat}} (50^\circ\text{F}) = 76.58 \times 10^{-4} \text{ lb/Lb dry air}$$

$$W_i \text{ 80\% } (50^\circ\text{F}) = 76.58 \times 0.80 = 61.264 \times 10^{-4} \text{ lb/Lb dry air}$$

$$W_{o\text{sat}} (20^\circ\text{F}) = 21.52 \times 10^{-4} \text{ lb/Lb dry air}$$

$$\Delta W = 61.264 - 21.52 = 39.744 \times 10^{-4} \text{ lb/Lb dry air}$$

$$v_{i\text{act}} (50^\circ\text{F}, 80\%) = 12.97 \text{ (from Table 44)}$$

$$V_L = \frac{W_T \times v_{i\text{act}}}{6000 \times \Delta W} = \frac{1.875 \times 12.97}{6000 \times 39.744 \times 10^{-4}} = \frac{1.875 \times 12.97}{0.6 \times 39.744} = 1.019 \text{ cfm/Hen}$$

Solution by use of the exact slide rule: (From left to right)

1-Put the first arrow opposite $t_i = 50^\circ\text{F}$ and read $W_{i\text{act}} = 61.3$

opposite R.H._i = 80%

2-Put the second arrow opposite $W_{i_{act}} = 61.3$ and read $\Delta W = 40.1$

opposite $t_o = 20^\circ\text{F}$

3-Put the third arrow opposite $\Delta W = 40$ and read $K = 46.9$

opposite $W_T = 1.875$ (or 45)

4a-Put the fourth arrow opposite $K = 46.9$ and read $V_L = 1.025$ cfm/Hen

opposite $t_i = 50^\circ\text{F}$

4b-Put the fourth arrow opposite $K = 46.9$ and read $V_L = 1.025$ cfm/Hen

opposite $v_{i_{act}} = 12.97$ (from Table 44 or slide rule No. 5)

Percentage of error:

$$\text{a. and b. } \% \text{ error} = \left(\frac{1.025 - 1.019}{1.019} \right) 100 = + \frac{0.006 \times 100}{1.019} = +0.59\%$$

In order to check more thoroughly the accuracy of this slide rule, the writer worked out 10 additional applications: ($W_T = 2.0$ lb/Hr, 100 hens or 48 lb/Day, 100 hens)

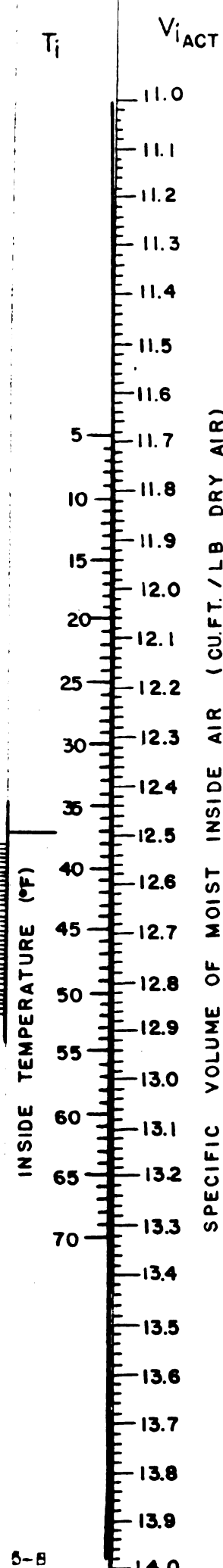
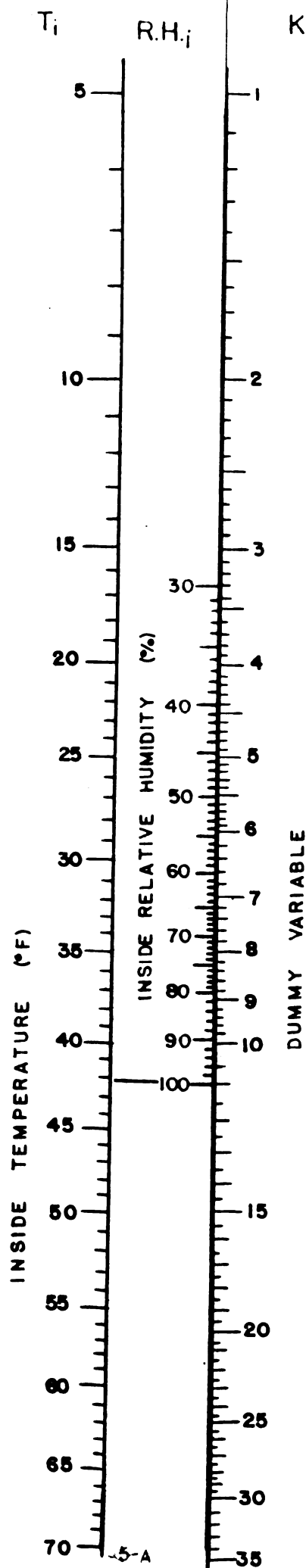
Check	t_i	R.H.i	$W_{i_{act}}$	t_o	ΔW	K	V_L	Diff.	% error
							Use of slide rule	True math. solut.	
No. 1	50	80	61.35	20	40	50	1.09	1.087	+0.003
No. 2	50	80	61.35	10	48.2	41.3	0.90	0.897	+0.003
No. 3	50	80	61.35	0	53.6	37.4	0.813	0.81	+0.003
No. 4	50	80	61.35	-10	57	35.1	0.763	0.762	+0.001
No. 5	40	80	41.75	10	28.5	70.2	1.488	1.478	+0.01
No. 6	40	80	41.75	0	33.9	59	1.26	1.251	+0.009
No. 7	40	80	41.75	-10	37.3	53.65	1.14	1.14	-
No. 8	30	80	27.5	10	14.5	137.5	2.82	2.78	+0.04
No. 9	30	80	41.75	0	19.8	101.5	2.085	2.05	+0.035
No. 10	30	80	41.75	-10	23.2	86.2	1.785	1.763	+0.033

From the above applications and checks, it can be said that slide rule No. 6 seems to maintain an accuracy within 1 or 2% for the whole range of inside temperature from 20° to 50°F , and since it is based on the true mathematical formula, unmodified, a similar degree of accuracy can be expected for inside temperatures outside this range. It should be

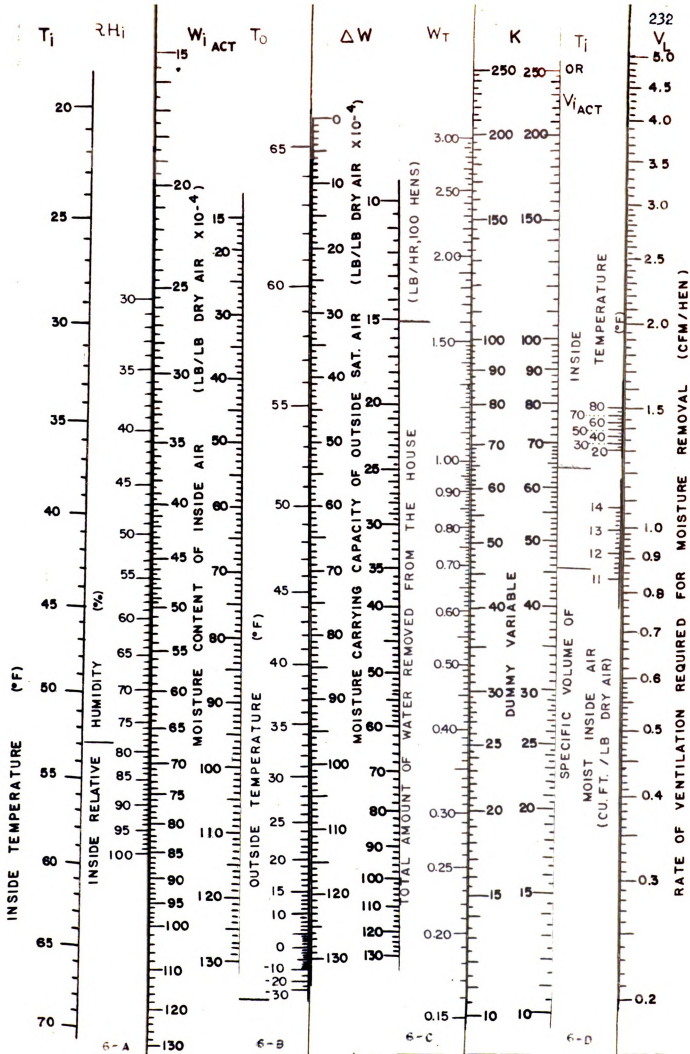
noted also that the use of the upper scale (t_i) on the fourth runner from the left does not impair the accuracy of the solution and since it makes the reading much easier and saves a few steps, the writer recommends its use in place of the lower (V_{iact}).

A few of the advantages of slide rule No. 6 are:

- 1-Little space is needed.
- 2-No straight edge is required.
- 3-No psychrometric chart of any kind is needed.
- 4-The ventilation rate is given in cfm per hen which makes it very suitable for purpose of comparison.
- 5-The accuracy seems to be within 1% for any inside temperature.



SLIDE RULE FOR DETERMINING
THE SPECIFIC VOLUME OF MOIST AIR



6-A

6-8

6-c

6-D

FOLD
HERE

SLIDE RULE FOR DETERMINING THE RATE OF VENTILATION REQUIRED FOR MOISTURE REMOVAL

c. Exact slide rule for design and analysis. For purpose of simplification, the previous two slide rules for moisture removal (slide rules Nos. 4 and 6) are based on an assumed outside relative humidity of 100%. This assumption makes them fitted for ordinary design purposes, since it is customary to use 100% relative humidity for design conditions. For analysis, however, it is necessary to deal with outside relative humidity values different from saturation and two slide rules (slide rules Nos 7 and 8) were designed for this purpose.

Slide rule No.7 permits finding the moisture carrying capacity of the outside air and is based on the following formulas.

- 1) $W_{i\text{sat.}} \times R.H._i = W_{i\text{act}}$ (Left portion of the slide rule)
- 2) $W_{o\text{sat.}} \times R.H._o = W_{o\text{act}}$ (Left portion of the slide rule)
- 3) $\Delta W = W_{i\text{act}} - W_{o\text{act}}$ (Center portion of the slide rule)

Equation 1 was changed to a logarithmic form:

$$(\log W_{i\text{sat}}) - (0) + (\log R.H._i) - (\log W_{i\text{act}}) = (0)$$

The values of $\log W_{i\text{sat.}}$ were plotted and values of t_i were substituted for them. A middle support was required for conversion of $\log W_{i\text{act}}$ values to $W_{i\text{act}}$ values.

Equation 2 was changed to a logarithmic form:

$$-(\log W_{o\text{act}}) + (\log R.H._o) - (0) + (\log W_{i\text{sat}}) = (0)$$

The values of $\log W_{o\text{sat}}$ were plotted and values of t_o were substituted for them. A middle support was required for conversion of $\log W_{o\text{act}}$ values to $W_{o\text{act}}$ values.

Equation 3 was changed to the form:

$$+(\log W_{i\text{act}}) - (\Delta W) + (0) - (\log W_{o\text{act}}) = (0)$$

Values of $W_{i,sat.}$ and $W_{o,sat.}$ were taken from the psychrometric table published in Heating, Ventilating, Air Conditioning Guide, 1951.

Slide rule No. 8 permits finding the rate of ventilation required for moisture removal once the moisture carrying capacity of the outside air is known. It is based on the following formulas:

$$4) K = \frac{W_T}{100 \Delta W}$$

$$5) V_L = K \times \left(\frac{V_{i,act}}{60} \right)$$

These two formulas are identical to formulas 3) and 4) used in the derivation of slide rule No. 6 and will therefore not be explained further.

In order to illustrate the use of the above slide rules, consider the same applications used for slide rule No 6. Perhaps applications with outside relative humidity different from 100% would have given better illustration of the real usefulness of these two new slide rules, but more space would have been required to present computations for the true mathematical solution given by the new set of conditions.

Application No. 1 ($t_1 = 20^\circ\text{F}$)

Given: Rate of water production in the house = 1.875 lb/Hr, 100 hens

(or 45 lb/Day, 100 hens)

Indoor temperature = 20°F

Indoor relative humidity = 85%

Outdoor temperature = -20°F

Outdoor relative humidity = 100%

Wanted: The minimum rate of ventilation required under the above conditions to remove all the moisture produced.

True mathematical solution: (Heating, Ventilating, Air Conditioning Guide, 1951)

2.419 cfm/Hen (Refer to application 1 of slide rule No. 6)

Solution by use of slide rules Nos. 7 and 8:

a. Slide rule No. 7: (From left to right)

1-Put the first arrow opposite $t_i = 20^\circ\text{F}$ and read $W_{i\text{act}}$ (or W_i) = 18.22

opposite R.H._i = 85%

2-Put the third arrow opposite $t_o = -20^\circ\text{F}$ and read $W_{o\text{act}}$ (or W_o) = 2.63

opposite R.H._o = 100%

3-Put the second arrow opposite $W_{o\text{act}}$ (or W_o) = 2.63 and read

$\Delta W = 15.6$ opposite $W_{i\text{act}}$ (or W_i) = 18.22

b. Slide rule No. 8: (From left to right)

1-Put the first arrow opposite $\Delta W = 15.6$ and read $K = 120$

opposite $W_T = 1.875$

2-Put the second arrow opposite $K = 120$ and read $V_L = 2.41$ cfm

per hen opposite $t_i = 20^\circ\text{F}$

Percentage of error:

$$\% \text{ error} = \left(\frac{2.41 - 2.419}{2.41} \right) 100 = -\frac{0.009 \times 100}{2.41} = 0.37\%$$

Application No. 2 ($t_i = 30^\circ\text{F}$)

Given: Rate of water production in the house = 1.875 lb/Hr, 100 hens

(or 45 lb/Day, 100 hens)

Indoor temperature = 30°F

Indoor relative humidity = 90%

Outdoor temperature = 10°F

Outdoor relative humidity = 100%

Wanted: The minimum rate of ventilation required under the above conditions to remove all the moisture produced.

True mathematical solution: (Heating, Ventilating, Air Conditioning Guide, 1951)

2.15 cfm/Hen (Refer to application 2 of slide rule No. 6)

Solution by use of slide rules Nos. 7 and 8:

a. Slide rule No. 7: (From left to right)

1-Put the first arrow opposite $t_i = 30^\circ\text{F}$ and read $W_{i\text{act}}$ (or W_i) = 31.1
opposite R.H._i = 90%

2-Put the third arrow opposite $t_o = 10^\circ\text{F}$ and read $W_{o\text{act}}$ (or W_o) = 13.2
opposite R.H._o = 100%

3-Put the second arrow opposite $W_{o\text{act}}$ (or W_o) = 13.2 and read
 $\Delta W = 18.0$ opposite $W_{i\text{act}}$ (or W_i) = 31.1

b. Slide rule No. 8: (From left to right)

1-Put the first arrow opposite $\Delta W = 18.0$ and read $K = 104$
opposite $W_T = 1.875$

2-Put the second arrow opposite $K = 104$ and read $V_L = 2.14$ cfm
per hen opposite $t_i = 30^\circ\text{F}$

Percentage of error:

$$\% \text{ error} = \left(\frac{2.14 - 2.15}{2.14} \right) 100 = - \frac{0.01 \times 100}{2.14} = 0.47\%$$

Application No. 3 ($t_i = 40^\circ\text{F}$)

Given: Rate of water production in the house = 1.875 lb/Hr, 100 hens
(or 45 lb/Day, 100 hens)

Indoor temperature = 40°F

Indoor relative humidity = 30%

Outdoor temperature = 0°F

Outdoor relative humidity = 100%

Wanted: The minimum rate of ventilation required under the above conditions to remove all the moisture produced.

True mathematical solution: (Heating, Ventilating, Air Conditioning Guide, 1951)

1.1703 cfm/Hen (Refer to application 3 of slide rule No. 6)

Solution by use of slide rules Nos. 7 and 8:

a. Slide rule No. 7: (From left to right)

1-Put the first arrow opposite $t_i = 40^\circ\text{F}$ and read $W_{i\text{act}}$ (or W_i) = 41.8

opposite R.H._i = 80%

2-Put the third arrow opposite $t_o = 0^\circ\text{F}$ and read $W_{o\text{act}}$ (or W_o) = 7.88

opposite R.H._o = 100%

3-Put the second arrow opposite $W_{o\text{act}}$ (or W_o) = 7.88 and read

$\Delta W = 3\frac{1}{4}$ opposite $W_{i\text{act}}$ (or W_i) = 41.8

b. Slide rule No. 8: (From left to right)

1-Put the first arrow opposite $\Delta W = 3\frac{1}{4}$ and read $K = 55.5$

opposite $W_T = 1.875$

2-Put the second arrow opposite $K = 55.5$ and read $V_L = 1.175$ cfm

per hen opposite $t_i = 40^\circ\text{F}$

Percentage of error:

$$\% \text{ error} = \left(\frac{1.1750 - 1.1703}{1.1703} \right) 100 = + \frac{0.0047 \times 100}{1.1703} = +0.43\%$$

Application No. 4 ($t_i = 50^\circ\text{F}$)

Given: Rate of water production in the house = 1.875 lb/hr, 100 hens
(or 45 lb/Day, 100 hens)

Indoor temperature = 50°F

Indoor relative humidity = 80%

Outdoor temperature = 20°F

Outdoor relative humidity = 100%

Wanted: The minimum rate of ventilation required under the above conditions to remove all the moisture produced.

True mathematical solution: (Heating, Ventilating, Air Conditioning, 1951)

1.019 cfm/Hen (Refer to application 4 of slide rule No. 6)

Solution by use of slide rules Nos. 7 and 8:

a. Slide rule No. 7: (From left to right)

1-Put the first arrow opposite $t_i = 50^\circ\text{F}$ and read $W_{i\text{act}}$ (or W_i) = 61.5

opposite $R.H._i = 80\%$

2-Put the third arrow opposite $t_o = 20^\circ\text{F}$ and read $W_{o\text{act}}$ (or W_o) = 21.65

opposite $R.H._o = 100\%$

3-Put the second arrow opposite $W_{o\text{act}}$ (or W_o) = 21.65

and read $\Delta W = 40$ opposite $W_{i\text{act}}$ (or W_i) = 61.5

b. Slide rule No. 8: (From left to right)

1-Put the first arrow opposite $\Delta W = 40$ and read $K = 46.75$

opposite $W_T = 1.875$

2-Put the second arrow opposite $K = 46.75$ and read $V_L = 1.015$ cfm

per hen opposite $t_i = 50^\circ\text{F}$

Percentage of error:

$$\% \text{ error} = \left(\frac{1.015 - 1.019}{1.019} \right) 100 = - \frac{0.004 \times 100}{1.019} = 0.39\%$$

In order to check more thoroughly the accuracy of slide rules Nos. 7 and 8, the writer worked out 18 additional applications: ($W_T = 20$ lb/Hr, 100 hens, or 48 lb/Day, 100 hens)

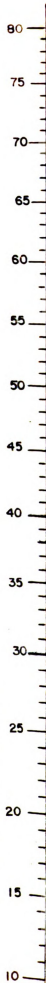
Check	t_i	R.H.i	W_{iact}	t_o	R.H.o	W_{oact}	W	K	V_L	Use of True slide rule math. solut.	Diff.	% error
1	75	70	132	60	70	77.8	54.1	37.15	0.855	0.845	+0.01	+1.83
2	65	80	106.3	50	50	38.4	67.9	22.6	0.664	0.661	+0.003	+0.45
3	60	90	100	50	90	69.1	30.8	65.5	1.455	1.440	+0.015	+1.04
4	50	80	61.6	40	60	31.3	30.2	66.5	1.441	1.441	-0.001	-0.07
5	50	90	69.2	40	100	52.1	17.0	117.7	2.54	2.572	-0.032	-1.25
6	40	80	41.85	30	70	24.2	17.6	114.1	2.40	2.405	-0.005	-0.21
7	40	70	36.45	30	90	31.15	5.35	37.5	7.95	7.80	+0.15	+1.92
8	40	70	36.45	20	70	15.05	21.3	94	1.982	1.965	+0.023	+1.17
9	35	90	38.45	30	40	13.8	24.6	81.4	1.70	1.695	+0.005	+0.30
10	35	80	34.3	20	50	10.8	23.5	85.5	1.782	1.78	+0.002	+0.11
11	30	90	31.1	20	100	21.6	9.75	206	4.29	4.32	-0.04	-0.93
12	30	80	27.65	10	90	11.9	15.7	128	2.63	2.61	+0.02	+0.77
13	30	90	31.1	10	100	13.25	18.0	111	2.23	2.30	-0.02	-0.87
14	30	80	27.65	0	60	4.75	22.8	88.4	1.82	1.805	+0.015	+0.83
15	25	90	24.7	10	70	9.2	15.4	130	2.64	2.65	-0.01	-0.38
16	25	80	21.9	-20	100	2.63	19.2	104.8	2.13	2.123	+0.007	+0.33
17	20	80	17.2	0	80	6.38	11.0	183	3.685	3.69	-0.005	-0.14
18	20	80	17.2	-20	100	2.63	14.6	137.3	2.77	2.767	+0.003	+0.11

From the above applications and check, it can be said that the slide rules Nos. 7 and 8 seem to maintain an accuracy within 1 or 2% for the whole range of inside temperature from 20° to 75°F and since they are based on the true mathematical formula unmodified, a similar degree of accuracy can be expected for inside temperature higher or lower than that range.

Because of the accuracy obtained with slide rules Nos. 4, 6, 7 and 8 and the ease of their use the writer believes:

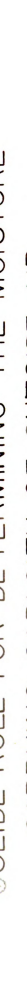
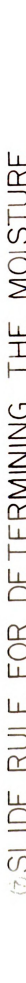
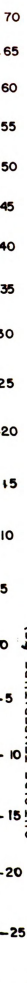
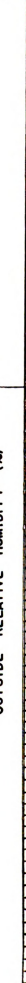
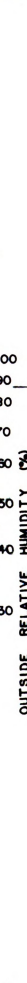
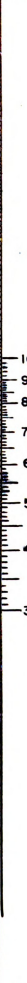
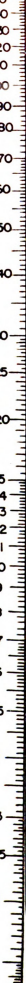
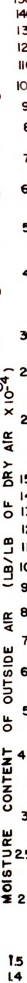
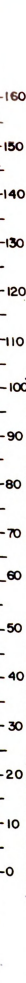
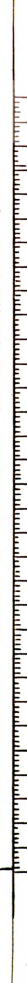
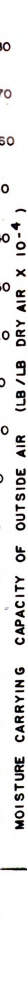
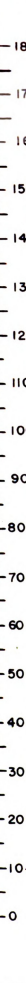
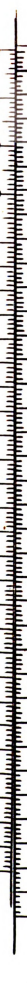
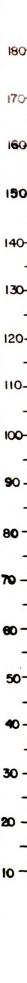
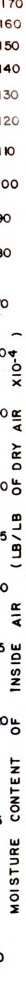
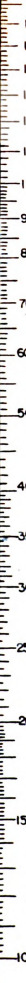
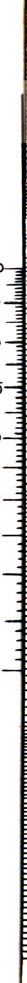
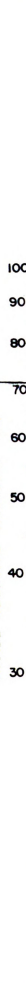
- 1-That they could be used in place of any nomograph or composite chart whenever an answer having a degree of accuracy within or near 1% is desired.
- 2-That they should be used to advantage even when no great accuracy is needed, because they require no straight edge and occupy only a limited space.

INSIDE TEMPERATURE (°F)



7-A

INSIDE RELATIVE HUMIDITY (%)

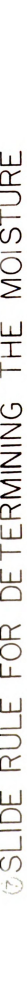


SLIDE RULE FOR DETERMINING THE MOISTURE
CARRYING CAPACITY OF OUTSIDE AIR

240

RH. 0

OUTSIDE TEMPERATURE (°F)



7-C

7-B

T_i

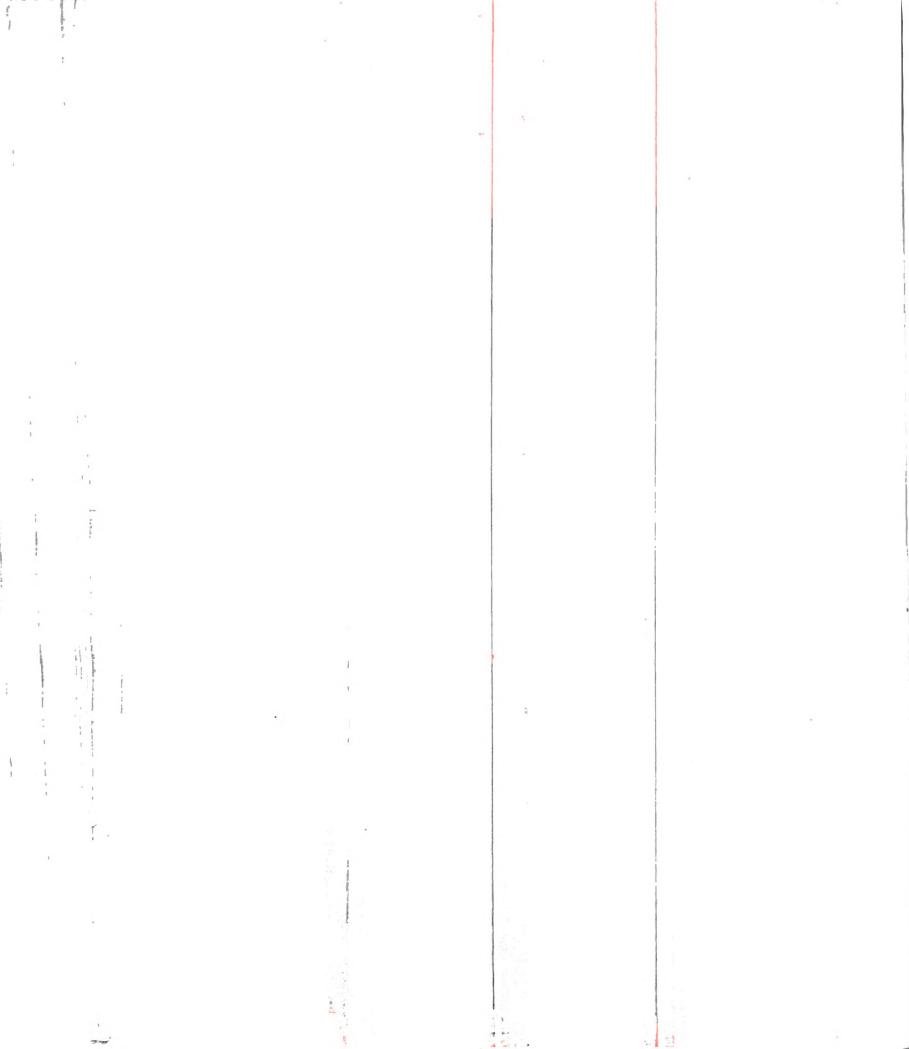
R.H._i

ΔW

W₀

RH. 0

T₀



MOISTURE CARRYING CAPACITY OF OUTSIDE AIR (LB/LB DRY AIR X 10⁻⁴)

TOTAL AMOUNT OF WATER REMOVED FROM THE HOUSE (LB/HR, 100 HENS)

ΔW

W_T

DUMMY VARIABLE

INSIDE TEMPERATURE (°F)

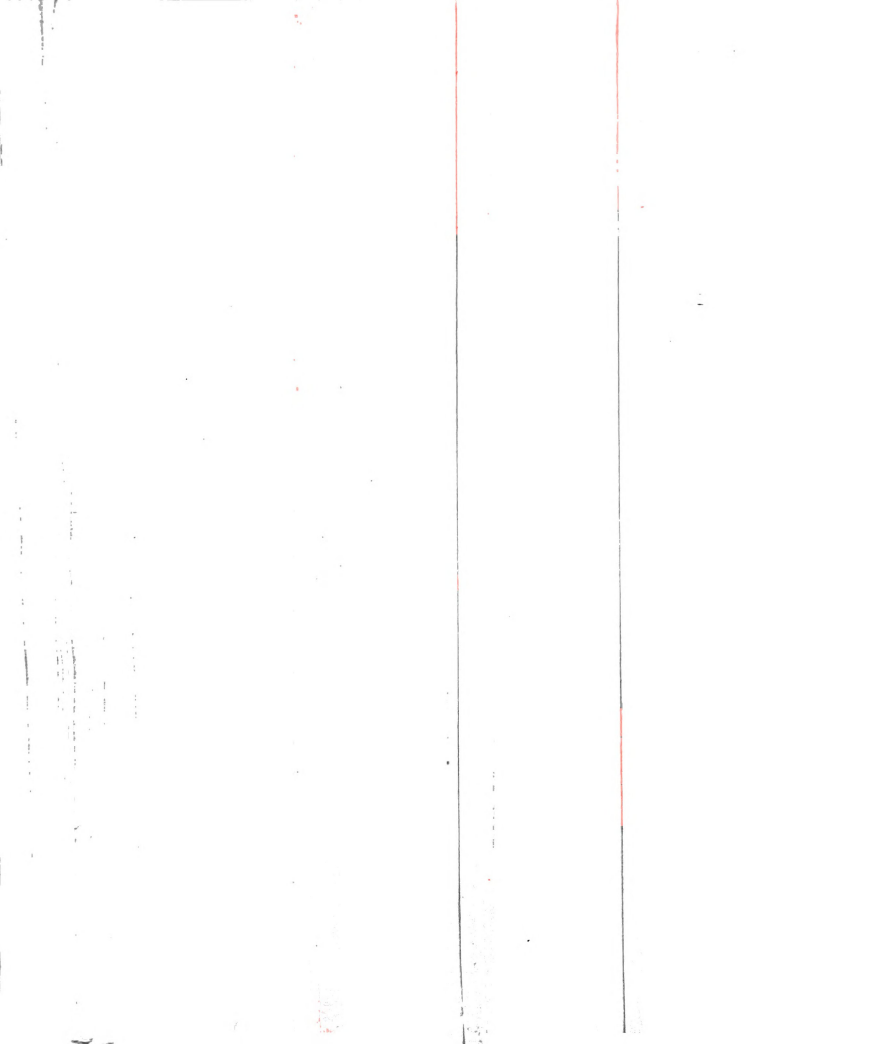
T_i

V_L

VENTILATION RATE REQUIRED FOR MOISTURE REMOVAL (CFM/HEN)

SLIDE RULE FOR DETERMINING THE RATE OF VENTILATION REQUIRED FOR MOISTURE REMOVAL

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2. Slide rule for determining the ventilation rate required for temperature control

a. Approximate design slide rule. In the construction of this slide rule (slide rule No 9), the following equation was used:

$$Q_s = \frac{60 V_s D}{52} + AC$$

$$\text{or } \frac{60 V_s}{52} = \frac{Q_s}{D} - AC$$

$$\text{or } 1.15385 V_s = \frac{Q_s}{D} - AC$$

where Q_s = Total sensible heat production per hen
(Btu/Hr, hen)

60 = Used to reduce the ventilation rate from one
hour to one minute

V_s = Ventilation rate (cfm/Hen)

D = Temperature difference

$52 = \frac{v}{0.24} = \frac{12.48}{0.24}$ (where v is the specific volume
of a mixture of moist air at approximately
33°F and 0.24 is the specific heat of air
at constant pressure) = Cubic feet of air
raised one °F by one Btu

AC = Heat loss characteristic of the building
(Btu/Hr, °F, hen)

This four-variable equation was separated into two equal parts each containing two variables and made equal to a third variable (dummy variable), for instance K.

$$1) K = \frac{Q_s}{D}$$

$$2) 1.15385 V_s = K - "AC"$$

Equation 1 was changed to a logarithmic form:

$$\log K = \log Q_s - \log D$$

$$+(\log Q_s) - (0) + (-\log D) - (\log K) = (0)$$

A middle support was required on the slide rule for conversion of $\log K$ values to values of K . Equation 2 was arranged in the following order:

$$+(K) - (0) + (-AC) - (1.15365 V_s) = (0)$$

Reference to any textbook on nomographs and slide rules (46) will explain how the slide rule was prepared from the above equations.

This slide rule is based on the assumption that $\frac{v_{i\text{act}}}{0.24} = 52$; therefore

it involves a certain error when values of $\frac{v_{i\text{act}}}{0.24}$ are different from 52.

This error is below 1% for the range of inside temperature from 30 to 35°F.

Since most of the design or critical temperatures used by designers are around 30 and 35°F, it may be concluded that this slide rule is in general very satisfactory for design purpose. The percentages of expected error have been computed in the section of nomographs, and are summarized here for both $\frac{v_{i\text{act}}}{0.24} = 52$ (as used in this slide rule) and $\frac{v_{i\text{act}}}{0.24} = 53$ (as used

by many other designers).

t_i	% of expected error when using $Z = \frac{v_{i\text{act}}}{0.24} = 52$	% of expected error when using $Z = \frac{v_{i\text{act}}}{0.24} = 53$
80	- 12.0	- 10.5
75	- 10.4	- 8.5
70	- 9.0	- 7.1
65	- 7.7	- 5.8
60	- 6.4	- 4.5
55	- 5.1	- 3.19
50	- 3.92	- 2.0
45	- 2.73	- 0.81
40	- 1.19	+ 0.92
35	- 0.40	+ 1.52
32	+ 0.25	+ 2.17
30	+ 0.73	+ 2.65
25	+ 1.84	+ 3.77
20	+ 2.88	+ 4.8
15	+ 3.92	+ 5.8
10	+ 5.0	+ 7.0
5	+ 6.1	+ 8.0

In order to illustrate the use of slide rule No 9, consider the following applications.

Application No. 1

Given: Sensible heat production = 45 Btu/Hr, hen

Desired indoor temperature = 40°F

Design outdoor temperature = 0°F

AC value of the house = 1.0

Wanted: The maximum rate of ventilation required to maintain the above temperature difference (40 - 0 = 40°F)

True mathematical solution:

$v_{i\text{act}}$ (40°F, 80%) = 12.67 (From Table 14)

$$60 V_s = \frac{v_{i\text{act}}}{0.24} \left(\frac{Q_s - AC}{D} \right)$$

$$\begin{aligned}
 V_s &= \frac{12.67}{0.24} \left(\frac{45}{10} - 1 \right) \frac{1}{60} \\
 &= 52.7 (1.125 - 1) \frac{1}{60} \\
 &= \frac{52.7 \times 0.125}{60} = 0.1098 \text{ cfm/Hen}
 \end{aligned}$$

Solution by use of the slide rule: (From left to right)

1-Put the first arrow opposite $Q_s = 45$ and read $K = 11.25$

opposite $\Delta t = 40$

2-Put the second arrow opposite $K = 11.25$ and read $V_s = 0.11$

opposite $AC = 1.0$

Percentage of error:

$$\left(\frac{0.11 - 0.1098}{0.1098} \right) 100 = \frac{+0.0002 \times 100}{0.1098} = + 0.18\%$$

Application No. 2

Given: Sensible heat production/hen = 45 Btu

Desired temperature differential = 40°F

Average air infiltration rate = 0.20 cfm/Hen

Wanted: The "AC" value required for the house to maintain this temperature differential, with no ventilation other than the infiltration.

True mathematical solution:

$$v_{i_{act}} (40^\circ\text{F}, 80\%) = 12.67 \text{ (From Table 14)}$$

$$AC = \frac{Q_s}{D} - \frac{60 V_s}{\left(\frac{v_{i_{act}}}{0.24} \right)}$$

$$= \frac{45}{40} - \frac{60 \times 0.20}{\left(\frac{12.67}{0.24} \right)}$$

$$= 1.125 - \frac{60 \times 0.20}{52.7} = 1.1250 - 0.2277 = 0.8973 \text{ Btu/Hr, } ^\circ\text{F, hen}$$

Solution by use of the slide rule: (From left to right)

1-Put the first arrow opposite $Q_s = 45$ and read $K = 11.25$

opposite $\Delta t = 40$

2-Put the second arrow opposite $K = 11.1$ and read $AC = 0.90$

opposite $V_s = 0.20$

Percentage of error:

$$\left(\frac{0.90 - 0.8973}{0.8973} \right) 100 = + \frac{0.0027 \times 100}{0.8973} = +0.30\%$$

Application No. 3

Given: Sensible heat production/hen = 45 Btu
Average air infiltration rate = 0.20 cfm/hen
AC value of the house = 2

Wanted: The maximum temperature differential possible with no ventilation other than the infiltration

True mathematical solution:

$$v_{iact} (40^\circ F, 80\%) = 12.67 \text{ (From Table 44)}$$

$$\begin{aligned} D &= \frac{Q_s}{\frac{60 V_s}{\left(\frac{v_{iact}}{0.24} \right)} + AC} \\ &= \frac{45}{\frac{60 \times 0.20}{\left(\frac{12.67}{0.24} \right)} + 2} = \frac{45}{\frac{12}{52.7} + 2} = \frac{45}{0.2277 + 2} = 20.2^\circ F \end{aligned}$$

Solution by use of the slide rule:

1-Put $AC = 2$ opposite $V_s = 0.2$ and read $K = 22.3$
opposite the right arrow

2-Put the left arrow opposite $Q_s = 45$ and read $\Delta t = 20.1^\circ$
opposite $K = 22.3$

Percentage of error:

$$\left(\frac{20.1 - 20.2}{20.2} \right) 100 = - \frac{0.1 \times 100}{20.2} = -0.50\%$$

In order to check more thoroughly the accuracy of this slide rule, the writer worked out 18 additional applications ($Q_s = 50$ Btu/hr, hen)

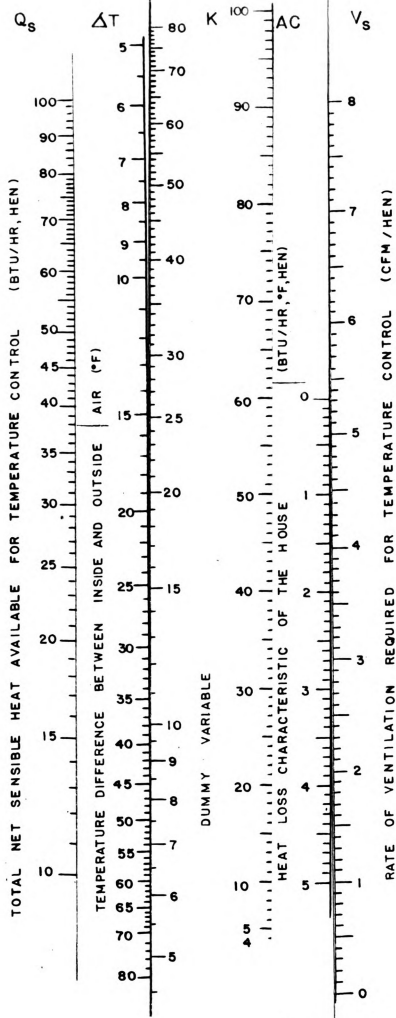
Check	t_1	R.H.i	t_o	Δt	AC	V_s	Diff.	Percentage of error	
						Use of slide rule		Actual error*	Expected error*
No. 1	75	70	60	15	1.5	1.61	1.75	-0.14	-8%
No. 2	65	70	50	15	1.5	1.61	1.707	-0.097	-5.2%
No. 3	60	70	50	10	1.5	3.05	3.22	-0.17	-5.3%
No. 4	50	80	40	10	1.5	3.05	3.152	-0.102	-3.2%
No. 5	50	80	30	20	1.5	0.87	0.9003	-0.0303	-3.36%
No. 6	40	80	20	30	1.5	0.145	0.1466	-0.0016	-1.1%
No. 7	40	80	0	40	1.0	0.215	0.2199	-0.049	-2.28%
No. 8	35	80	25	10	1.5	3.05	3.0792	-0.0292	-0.95%
No. 9	35	80	10	25	1.5	0.135	0.1399	-0.0049	-1.1%
No. 10	30	80	20	10	1.5	3.05	3.011	+0.039	+1.3%
No. 11	30	80	10	20	1.5	0.87	0.86	+0.01	+1.16%
No. 12	30	85	0	30	1.5	0.145	0.14335	+0.00175	+1.2%
No. 13	30	90	-10	40	1.0	0.215	0.21508	-0.00008	-0.04%
No. 14	25	90	5	20	1.5	0.87	0.851	+0.019	+2.23%
No. 15	25	90	-15	40	1.0	0.215	0.2128	+0.0022	+1.05%
No. 16	20	90	0	20	1.0	1.30	1.2625	+0.0375	+2.97%
No. 17	20	90	-20	40	1.0	0.215	0.2104	+0.0046	+2.19%
No. 18	10	90	-30	40	1.0	0.215	0.2059	+0.0091	+4.4%

* When V_s is equal to approximately 0.20 cfm/Hen or less, the actual error will be bigger than the expected error if a great care is not taken in making the readings.

From the above applications and checks, it may be concluded that for values of V_s greater than 0.40 cfm/Hen, the actual error is quite close to the expected error as computed previously. This means that for the range of temperature from 15 to 50°F and for values of V_s greater than 0.40, the error may be expected to fall within 0 to 5%. Such a range of accuracy is satisfactory enough for most design problem. For values of V_s smaller than 0.40 cfm per hen, a great care must be taken in making the readings, otherwise the actual error may be much bigger than the expected error.

A few of the advantages of slide rule No 9 are:

- 1-It requires little space
- 2-No straight edge is required
- 3-The AC value of the house per hen is used instead of the total AU_{aver} value, which makes it easier to compare houses of any capacity, exposed area and insulating value.
- 4-The ventilation rate is given in cfm per hen which makes it very suitable for purpose of comparison and analysis.
- 5-The slide rule has only two runners which means that any answer can be obtained quickly in only two operations.



SLIDE RULE FOR DETERMINING THE RATE OF VENTILATION REQUIRED FOR TEMPERATURE CONTROL



b. Corrected slide rule for design and analysis. In order to reduce the amount of error given by the approximate slide rule (slide rule No 9), the writer added a third slide involving a factor of correction. The value of the correction factor was found as shown in the following table.

t_1	$v_1(80\%)$	$z = \frac{v_1(80\%)}{0.24}$	Correction factor $z \div 52$
80	13.98	58.25	1.12
75	13.78	57.42	1.104
70	13.61	56.71	1.0906
65	13.44	56.00	1.077
60	13.28	55.33	1.064
55	13.12	54.66	1.051
50	12.97	54.04	1.04
45	12.82	53.42	1.0273
40	12.67	52.82	1.012
35	12.53	52.21	1.004
32	12.45	51.87	0.9975
30	12.39	51.62	0.9927
25	12.25	51.04	0.9815
20	12.12	50.50	0.9711
15	11.990	49.96	0.9608
10	11.855	49.307	0.948
5	11.722	48.83	0.939
0	11.59	48.28	0.9285

The equation of this third slide is:

$$V_S \text{ corr.} = V_S \text{ appr.} \times \text{C.F.}$$

where $V_S \text{ corr.}$ = Corrected ventilation rate
(cfm/Man)

$V_S \text{ appr.}$ = Approximate ventilation rate
(cfm/Man)

$$\text{C.F.} = \text{Correction factor} = \frac{z}{52} = \frac{v_1(80\%)}{0.24} \div 52$$

This equation was changed to a logarithmic form:

$$\begin{aligned} \log V_S \text{ corr.} &= \log V_S \text{ appr.} + \log \text{C.F.} \\ + (\log V_S \text{ appr.}) - (0) + (\log \text{C.F.}) - (\log V_S \text{ corr.}) &= (0) \end{aligned}$$

Values of $\log Z.F.$ were plotted and values of t_i were substituted for them.

Reference to any textbook on nomographs and slide rules (46) will explain how slide rule No 10 was prepared.

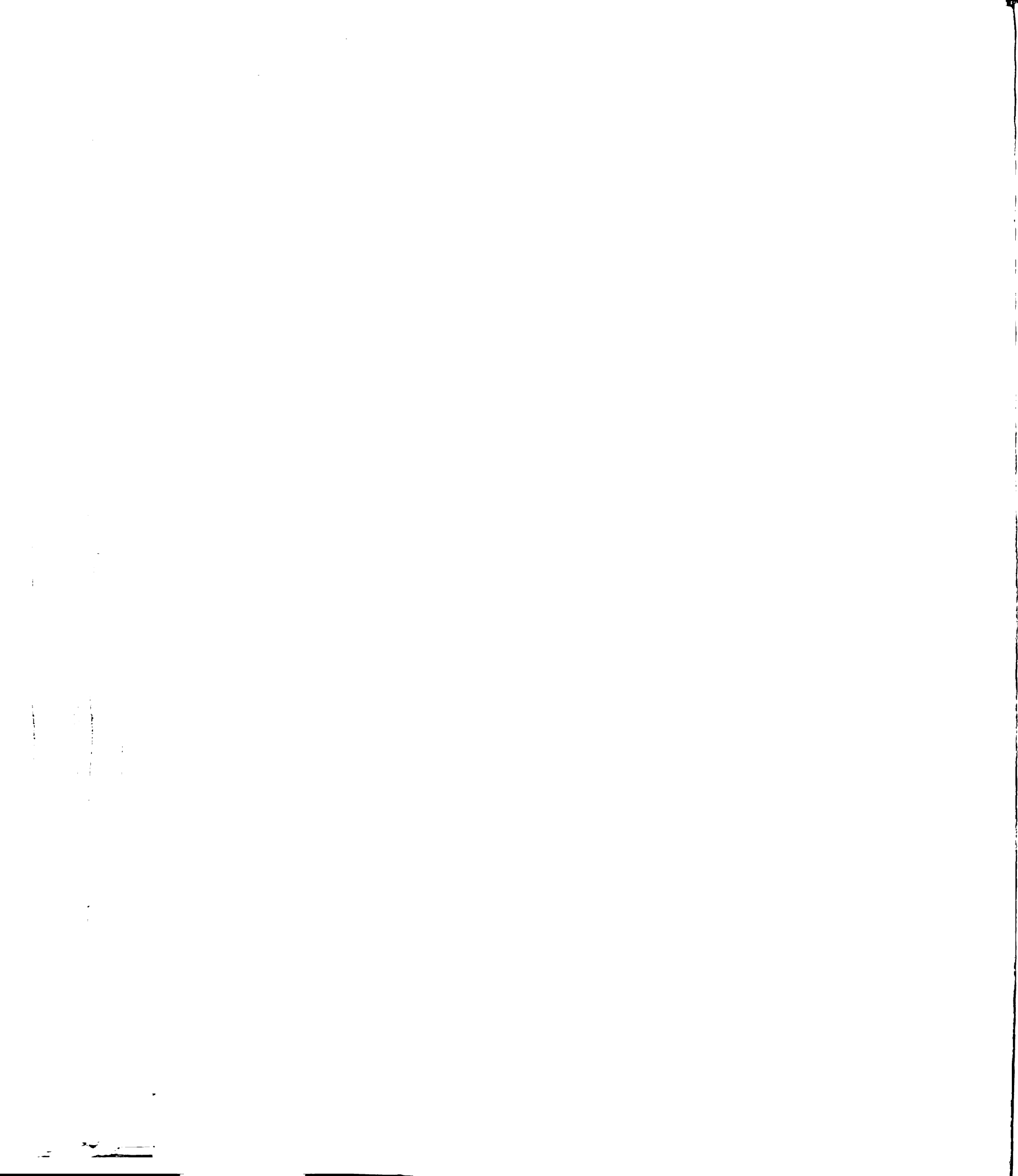
The writer used this slide rule for the same 18 applications solved with the approximate slide rule. The corrected ventilation rates and the percentages of error obtained are presented here: ($Q_s = 50$ Btu/Hr, hen)

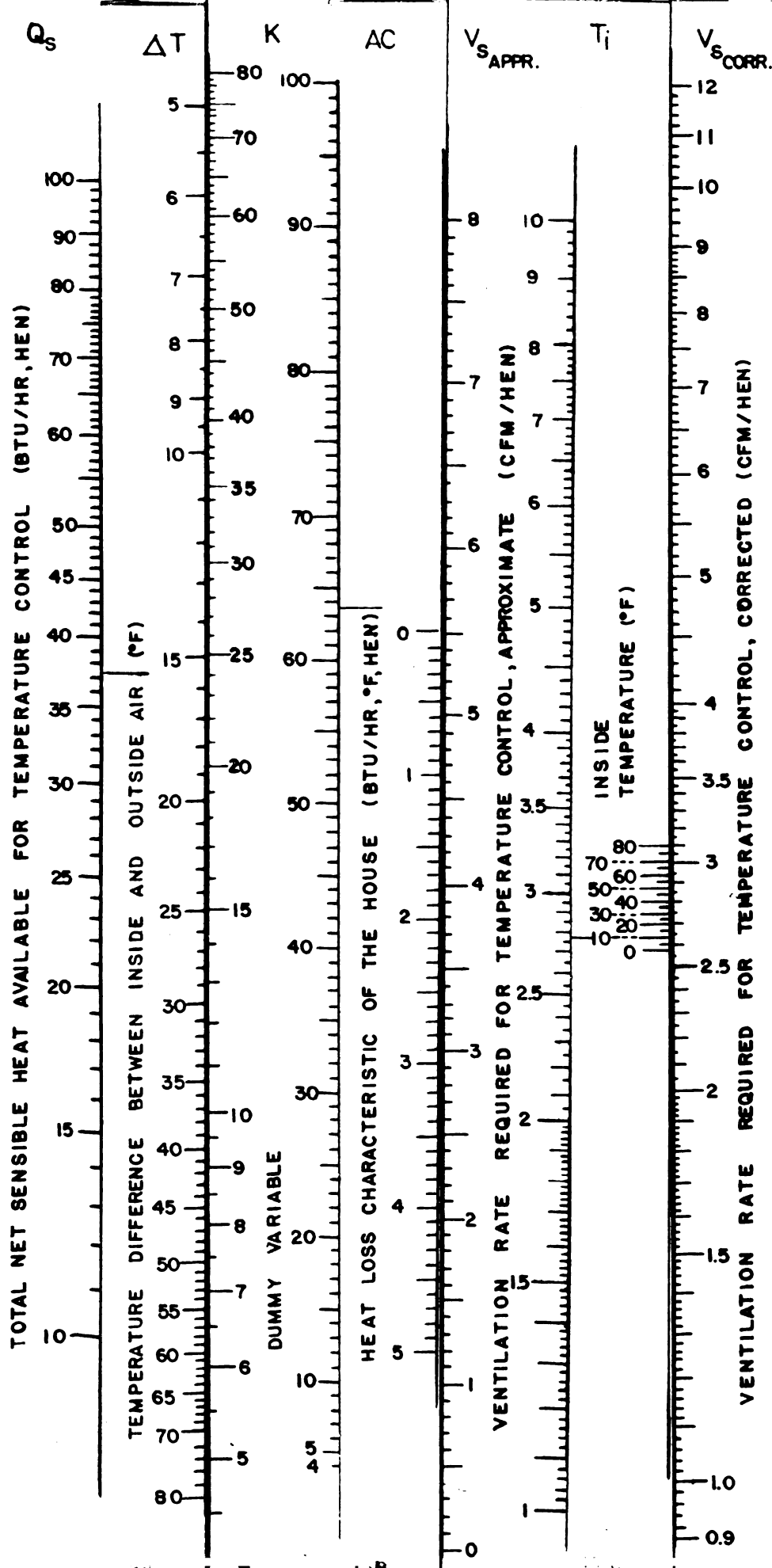
Check	t_i	R.H. ₁	t_o	t	AC	V_s		True math. solut.	Actual error (%)	
						Appr. slide rule	Corrected slide rule		Appr. slide rule*	Corrected slide rule*
No. 1	75	70	60	15	1.5	1.61	1.73	1.75	- 8%	+ 1.71%
No. 2	65	70	50	15	1.5	1.61	1.725	1.707	- 5.2	+ 1.05
No. 3	60	70	50	10	1.5	3.05	3.225	3.22	- 5.3	+ 0.16
No. 4	50	80	40	10	1.5	3.05	3.16	3.152	- 3.2	+ 0.25
No. 5	50	80	30	20	1.5	0.87	0.90	0.9003	- 3.36	- 0.02
No. 6	40	80	20	30	1.5	0.145	0.146	0.1466	- 1.1	- 0.04
No. 7	40	80	0	40	1.0	0.215	0.2175	0.2199	- 2.28	- 1.1
No. 8	35	80	25	10	1.5	3.05	3.05	3.0792	- 0.95	- 0.95
No. 9	35	80	10	25	1.5	0.435	0.435	0.4399	- 1.1	- 1.1
No. 10	30	80	20	10	1.5	3.05	3.015	3.011	+ 1.3	+ 0.16
No. 11	30	80	10	20	1.5	0.87	0.855	0.86	+ 1.16	- 0.58
No. 12	30	85	0	30	1.5	0.145	0.143	0.14335	+ 1.2	- 0.25
No. 13	20	90	-10	40	1.0	0.215	0.213	0.21508	- 0.04	- 0.97
No. 14	25	90	5	20	1.5	0.87	0.85	0.851	+ 2.23	- 0.12
No. 15	25	90	-15	40	1.0	0.215	0.211	0.2128	+ 1.05	- 0.85
No. 16	20	90	0	20	1.0	1.30	1.26	1.2625	+ 2.97	- 0.20
No. 17	20	90	-20	40	1.0	0.215	0.2085	0.2104	+ 2.19	- 0.90
No. 18	10	90	-30	40	1.0	0.215	0.204	0.2059	+ 4.4	- 0.92

* When V_s is equal to approximately 0.40 cfm/Hen or less, the actual error may be bigger than the one indicated here, if a great care is not taken in making the readings.

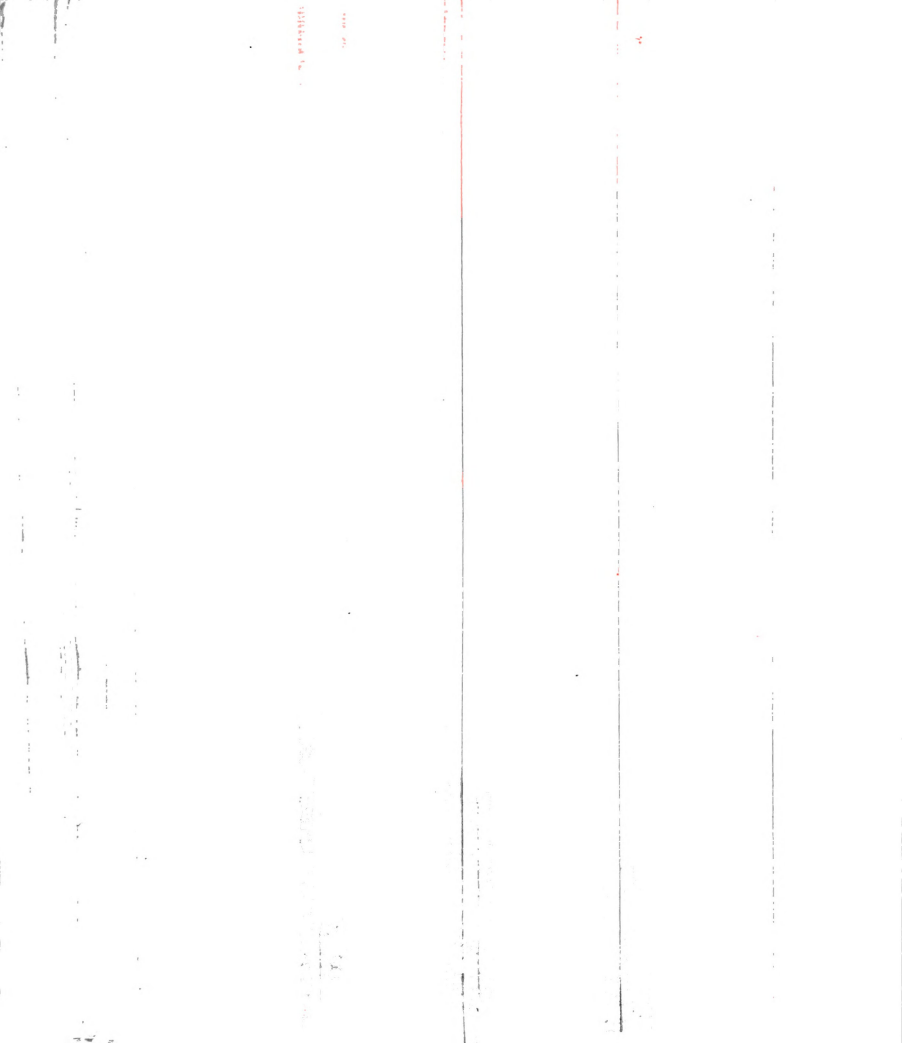
To conclude from the above table, it can be said that the corrected slide rule for temperature control (slide rule No 10) gives answers generally within 1 or 1.5% error, for the whole range of inside temperatures from 10 to 75°F. This is much better than the approximate slide rule which was giving errors as big as 8% (Check No.1) because of the assumption of a constant v_i value equal to 52. 0.24

The advantages of this slide rule are the same as for the approximate slide rule. In addition it gives a higher degree of accuracy as compared to the true mathematical solution.





SLIDE RULE FOR DETERMINING THE RATE OF
VENTILATION REQUIRED FOR TEMPERATURE CONTROL



F. The Problem of Temperature Control

The problem of temperature control without the use of any artificial heat is related to many factors among which are:

- 1-Use of proper insulation.
- 2-Use of methods to restrict or stop completely the ventilation at the lower temperatures.
- 3-Use of the biggest number of birds consistent with good management practice.
- 4-Use of deep litter as a source of secondary heat.
- 5-Use of insulated glass (Thermopane, etc.).
- 6-Reduction of air infiltration to a minimum.
- 7-Use of heat exchanger.

The factors numbered from 2 to 6 have been or will be discussed in other sections. This section will deal with the determination of the proper amount of insulation to use and with the discussion of the design and use of heat exchangers.

1-Use of proper insulation

a. Economical advantages of insulation. Insulation as an aid to temperature control is usually recognized as being favorable to egg production (49) (4) (22) and feed saving (42) (79), even though at least one experiment (13) seems to prove greater egg production under uncontrolled conditions.

The Insulation Board Institute (49) mentions some results obtained at Iowa State College for demonstration record flocks: "Twenty-three of the flocks were housed in insulated buildings, and 21 in non-insulated structures. The following differences are of interest:

- 1-Hens in insulated houses laid 16 more eggs per bird.
- 2-During the year, the hens in insulated houses consumed 6.1 pounds of feed less per bird (This corresponds to a ton and a half per 500 birds).

3-It took one pound less feed to produce a dozen eggs from the hens in the insulated house.

4-Investment per hen was 3¢ cents more for those housed in insulated buildings. However, this was almost entirely offset by the expense per hen which was 36 cents less for birds in insulated houses.

5-Income per hen soared to 62 cents more per bird in insulated laying houses. Labor returned 4¢ cents more per hour in the care of birds in laying houses which were insulated.

6-Money invested in the flocks in insulated houses returned 27% while only 14.3% was realized on these flocks in non-insulated houses."

These field results seem to indicate definite advantages in using insulation. With feed prices high, the saving of feed alone would probably offset the initial cost of insulation.

b. Cost of insulation. Benefits from the use of insulation in increasing thicknesses follows the law of diminishing returns, and from a purely investment standpoint the returns of the first inch of insulation would be greater than from additional thicknesses. The following table (Table 45) from data by Jefferson (52) shows the reduction in heat transfer as increasing inches of insulation are added to a standard uninsulated ceiling.

The decreasing efficiency shown here is, however, offset in the case of rigid insulating boards by saving the cost of the inside wood sheathing otherwise required with loose fill commercial insulation or dry shavings.

Table 17. Law of diminishing returns applied to increasing thickness of insulation*

	Heat transfer coefficient U (Btu/°F, sq.ft)	Insulating value or resistance $R = \frac{1}{U}$	Decrease in heat transfer ΔU (Btu/°F, sq.ft)	Efficiency of the additional inch of insulation as compared to the first inch (%)
1-Uninsulated ceiling	0.620	1.61	-	-
2-One inch of insulation	0.202	4.95	0.418	100%
3-Two inches of insulation	0.121	8.27	0.081	19.4%
4-Three inches of insulation	0.086	11.57	0.035	8.4%
5-Four inches of insulation	0.067	14.92	0.019	4.5%

* From data by Jefferson (52)

c. Dry shavings as an insulant. According to Jefferson (52): "Wood shavings and other similar types of native materials are not suggested as a substitute for commercial insulation in all cases, but where they are available, they offer an economical solution to the problem of insulating many types of farm building." He adds: "Most of the objections frequently raised against the use of dry wood shavings and similar materials are not necessarily disadvantages but are obstacles of prejudice. The use of shavings for insulation does not in itself constitute a fire hazard. Although shavings are inflammable and if ignited will burn along with all the other readily inflammable materials in a frame building, they may actually be a fire retardant. When used in the wall space, they destroy the flue action of these spaces and retard the spread of flames. There is likewise no evidence to support the objection that these insulating materials would harbor insects and vermin any more than the same building without this type of insulation."

Cher (92), Meyer and Flickle (68) and many other poultry house designers state that, if wood shavings, dry sawdust, ground corn cobs or chopped straw are used as insulant, it is better to mix one pound of ordinary hydrated lime with each basket of these materials in order to discourage rodents and insects. Chertier (20) considers that mixing the lime with the shavings makes a very irritating dust for the workers. He recommends instead (1) putting two to four pounds of lime at the bottom of the studs (depending on the spacing), (2) adding shavings up to the middle of the wall height, (3) pouring some more hydrated lime, (4) adding shavings up to the top, (5) placing half an inch of lime at the top.

d. Relation between insulation and ventilation. Insulation and ventilation work together and it is impossible to get a good temperature and humidity control without proper insulation.

The air change required to remove the moisture varies with temperature and relative humidity outside and inside the house. The following example shows that an uninsulated house requires a much higher rate of ventilation than an insulated house with respect to moisture removal.

Given: Outside temperature = 10°F

Outside relative humidity = 80%

	Uninsulated house	Well insulated house
Inside temperature	15°F	35°F
Inside relative humidity	80%	75%

Wanted: The rate of ventilation required for the removal of 30 pounds of moisture per day per one hundred hens (or 1.25 lb/Hr, 100 hens).

Answer: Uninsulated house = 8.76 cfm/hen

Insulated house = 1.16 cfm/hen

e. Determination of the economical insulation requirement for temperature control. The amount of insulation required to restrict heat losses through the walls, ceilings, doors and windows can be calculated so that the inside temperature of the house can be maintained at the desired point.

The insulation requirements first depend on the exposed area in square

feet per animal. It also depends on the heat produced by the type of animal concerned, and the minimum temperature differential desired with the minimum ventilation used during extremely cold spells.

The formula for finding the amount of insulation required is derived from the following simplified heat balance equation which neglects the secondary sources of heat losses in poultry houses:

$$\text{Sensible heat produced by the birds} = \text{Heat loss due to the building} + \text{Heat loss due to the minimum air change}$$

$$Q_s = A \cdot U \cdot D + \frac{V_{\min} \cdot D}{52}$$

where Q_s = Total sensible heat available in the house
(Btu/Hr, bird)

$V_{\min}$ = Minimum air flow rate during extremely
cold spells = Average infiltration rate
where intermittent ventilation is used

$D = \Delta t$ = Difference between inside and outside
temperature ($^{\circ}\text{F}$)

$C = U_{\text{aver.}}$ = Average heat transfer coefficient
of the building

$52 = \frac{v}{0.24} = \frac{12.48}{0.24}$ (where v is the specific
volume of one pound of air at approximately
 33°F and 0.24 is the specific heat of air
at constant pressure) = Number of cubic feet
of air raised one $^{\circ}\text{F}$ by one Btu.

A = Total exposed area per bird (sq.ft/bird)

v = Specific volume of one pound of dry air

0.24 = Specific heat of air at constant pressure

Solving for AC or $AU_{aver.}$ we have:

$$AC = \frac{Q_s}{D} - \frac{V_{min}}{52}$$

Solving for C or $U_{aver.}$ we have:

$$C = \frac{Q_s}{AD} - \frac{V_{min}}{52A} \quad \text{or } R_t = \frac{1}{C} = \frac{1}{\frac{Q_s}{AD} - \frac{V_{min}}{52A}}$$

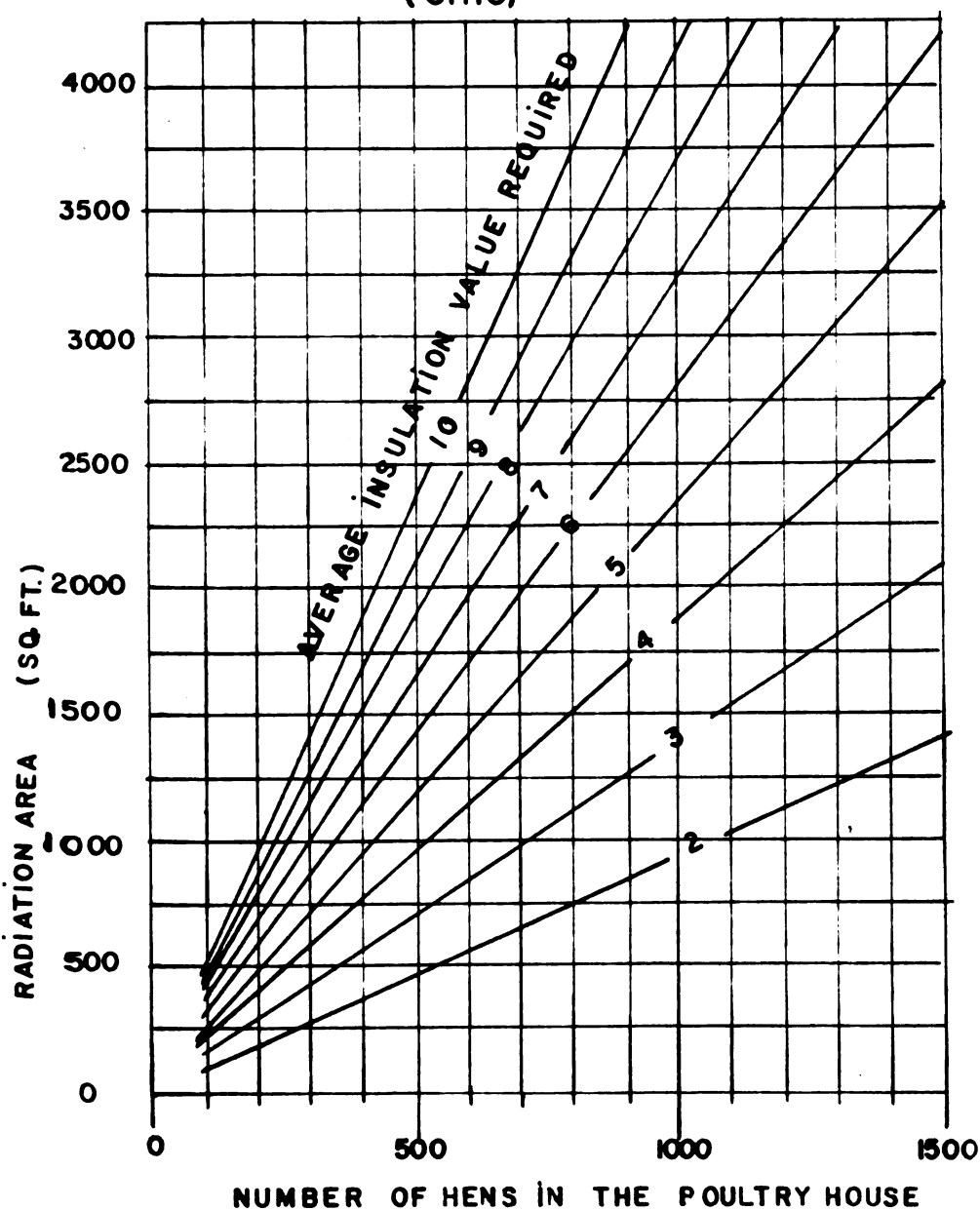
aa. Shier's chart for the insulation value required for temperature control.
Using the later formula and making some basic assumptions he thought suitable for Ohio weather conditions, Shier (92) prepared in 1943 the following type of chart (Figure 31) for finding the R_t value (or average insulating value) to be recommended for poultry houses having different numbers of laying birds and different total exposed area (or radiation area). The writer believes that such a type of chart is very useful for fast determination of the most economical insulation value to use for a building. Similar charts could be made for other states. Once the economical C or $U_{aver.}$ value is determined, it is a simple matter to design or adjust the construction of walls, ceilings, doors and windows in such a way to get a value close to the one recommended.

Since Mr. Shier is no longer living, the writer was unable to get any detailed information concerning the basic assumptions used in the drawing of his chart.

The radiation area (or exposed area) is made up of the sum of the areas of the ceiling, side walls, end walls, windows and doors.

FIGURE 31

SHIER CHART FOR DETERMINING THE
AVERAGE INSULATION VALUE NEEDED TO
MAINTAIN DESIRABLE INSIDE TEMPERATURES(92)
(OHIO)



bb. Ryan and Pile chart for the insulation value required for temperature control. More recently (1971), Ryan and Pile (88) presented two charts (Figures 32 and 33) similar to the Chier chart, for application to Minnesota conditions.

The conditions assumed in the derivation of the charts were as follows:

1-Outside temperature = 15°F

2-Outside relative humidity = 100%

3-Inside temperature = 45°F

4-Inside relative humidity = 75%

The heat and moisture production of the birds, was taken from Mitchell and Kelley (73). The minimum rate of ventilation or infiltration used in the computations is not given specifically although it seems to be 1/4 cubic foot of air per minute (minimum allowable volume of air change recommended by the writers under restricted ventilation).

The radiation area (or exposed area) is made of the sum of the areas of the ceiling, side walls, end walls, windows and doors.

A comparison of Chier's recommendations with those of Ryan and Pile is presented here (Table 1.6).

Table 16. Comparison of average insulating value recommendations for Ohio and Minnesota poultry houses

Number of hens	Total radiation area (sq. ft)	Radiation area per hen * *	Average insulating value recommendations		
			Ohio		Minnesota
			(Shier) (92)	(Ryan and Pile) (88)	
500	500	1	2.20	0 *	Smaller
	1000	2	4.15	3.00	value
	1500	3	5.30	5.80	Greater value
	2000	4	8.10	8.70	
	2500	5	10.50 *	11.60	
1000	1000	1	2.20	1.50	Smaller
	2000	2	4.35	4.30	
	3000	3	6.50	6.15	value
	4000	4	8.70	9.00	Greater
	5000	5	11.50	12.85	value
1500	1500	1	2.20	2.00	Smaller
	3000	2	4.30	4.80	value
	4500	3	6.35 *	7.7	Greater
	6000	4	8.80 *	10.50	value
	7500	5	10.90 *	13.50	

* Obtained by extrapolation

* * Radiation area or exposed area = Total area of the exposed ceiling and walls, the latter area including the area occupied by the windows and doors.

Values are found to agree quite closely except for very small and very high radiation area (1 sq. ft and 5 sq. ft per hen).

FIGURE 32

RYAN AND PILE CHART FOR DETERMINING THE AVERAGE
INSULATION VALUE NEEDED TO MAINTAIN DESIRABLE INSIDE TEMPERATURES (88)

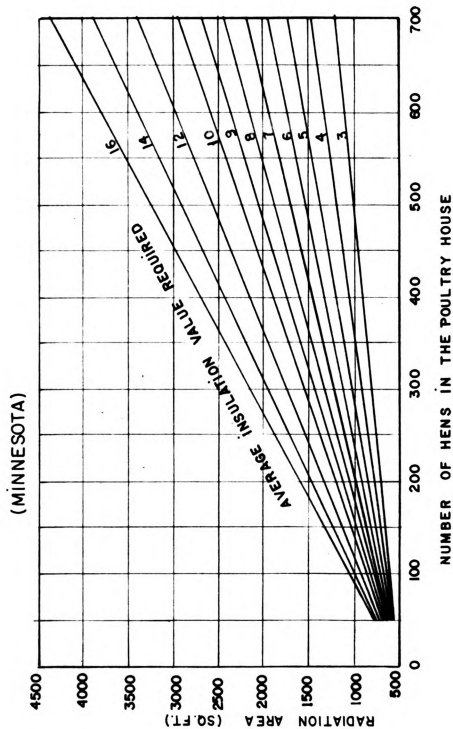
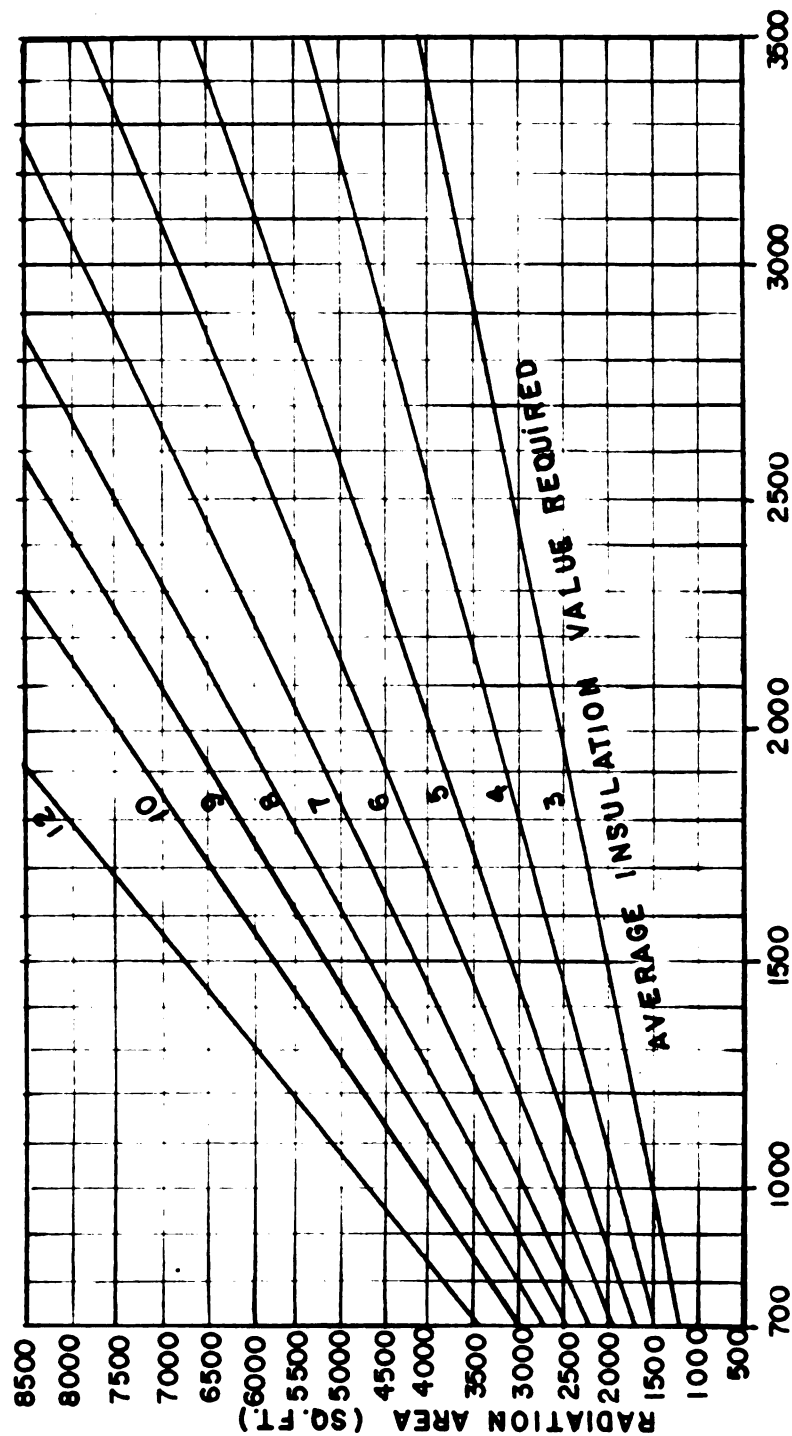


FIGURE 33

RYAN AND PILE CHART FOR DETERMINING THE AVERAGE

INSULATION VALUE NEEDED TO MAINTAIN DESIRABLE INSIDE TEMPERATURES(88)

(MINNESOTA)



NUMBER OF HENS IN THE POULTRY HOUSE

cc. Other recommendations. Ashby, Miller et al (4) suggest the following insulation values (Table 47) as minimums for poultry laying houses 20' x 20' in size or larger. Because of the larger wall area per hen in small houses, the heat loss per hen is bigger and greater insulating value is required.

Table 47. Insulating values desirable in walls and ceilings
in the different climatic zones of United States *

Climatic zone		Desirable insulating values (Walls - Ceilings)	
Zone 1	Coldest parts	10	- 12
		or 8	- 15
	Warmer parts	6	- 8
		or 5	- 12
Zone 2		4	- 6
		or 3	- 10
Zone 3 and 4		1.4	- 4

* From Ashby, Miller et al (4)

According to Bell and Moore (14) (Michigan): "for laying houses, a resistance value ("R") of at least ten is required and a value of twelve to fifteen is recommended. The rating of ordinary insulation shows a value of about 3.05 per inch, on the average. Therefore, insulating to the depth of three to four inches is necessary on the walls and ceiling of a Michigan laying house."

The Insulation Board Institute (49) recommends the following insulating values for poultry houses (Table 48):

Table 49. Insulating values required for poultry houses to maintain an inside temperature of 35°F. **

Expected outside temperature	Temperature difference (°F)	Required maximum wall and ceiling heat transfer coefficient ($U_{aver.}$)	Recommended minimum insulating value (R_t)
15	20	0.21	4.76
10	25	0.17 *	5.88
5	30	0.13 *	7.69
0	35	0.10 *	10.00
- 5	40	0.08 *	12.50
-10	45	0.063 *	15.87
-15	50	0.05 *	20.00

** From the Insulation Board Institute (42), based on a heat production of 35.7 Btu per hour per hen at 35°F.

* With storm windows

dd. Use of the nomographs and slide rule to find the insulating value required for temperature control. One disadvantage of Shier's chart and Ryan and Pile charts is that they are good only for local conditions or for a certain limited region. As to the other recommendations presented here they are too general and do not consider the radiation area and similar factors. For more specific use, the nomographs and slide rule for determining the rate of ventilation required for temperature control are more suitable. The method of finding the AC value required has been described in the applications accompanying these nomographs or slide rule, and will not be repeated here.

2. Design and use of heat exchangers in poultry houses

During periods of low temperatures, the animal heat alone is often insufficient to vaporize moisture from animals, to offset heat losses through building walls and to warm the required supply of ventilating air. Heat exchangers are for the purpose of providing a source of supplementary heat, by using the otherwise wasted heat energy of the exhaust air to warm the incoming fresh air. They are especially suited for areas having relatively cold winter.

Heat exchangers are reported by Giese and Bond (35) to have been used very effectively in both dairy barns and poultry houses. They say: "It was possible to ventilate barns at full ventilation rate with outside air temperatures from 16 to 19°F lower than when ventilated by some means in which the incoming air was untempered."

Heat exchangers may be either of the parallel or of the counterflow principle. As to their fabrication, it can be varied tremendously. In their review of literature, Giese and Bond (35) describe some heat exchangers used in farm buildings as being built in the following manner.

Heat exchanger	Warm air ducts	Cold air ducts	Principle
No. 1	An 8" square duct, 20 ft long	Four 3-in galvanized steel pipes	Counterflow
No. 2	An 16" square duct, 30 ft long	Sixteen 3-in pipes	Counterflow
No. 3	An 11 x 21" air duct, 45 ft long	Eight 4-in pipes	Parallel

They also report that several units were manufactured by the Loudon Machinery Company and installed in cooperation with the Public Service Company of Northern Illinois. These units have proven eminently satisfactory.

The greatest obstacle to the use of heat exchangers in animal shelters is the ordinarily low temperature differential existing between the warm exhaust air and the cold incoming air. This condition requires that the heat transfer be accomplished by either a large heat transfer surface or a very high air velocity. Since the air velocity affects the operational cost in overcoming friction, they concluded that an exchanger with a large surface area would seem to be the most economical. They therefore designed a plate type of heat exchanger which is presented in Figures 34, 35 and 36. The advantages they claim for this heat exchanger are:

- 1-Large heat transfer surface
- 2-Low operating cost
- 3-Low first cost
- 4-Ease of fabrication
- 5-Ease of operation
- 6-Small space requirement

In the construction of this plate-type heat exchanger (Figures 34, 35, 36) they used 26" x 72" sheets of corrugated aluminum roofing to serve as the plates or fluid separators, since aluminum is an excellent heat transfer material and is quite resistant to corrosion. The spacing between the sheets is obtained by the use of 1/4" spacers. The number of sheets to use may be any number (20, 30, 40, etc.) according to the requirements of the different animal shelters. The opposite edges of the plates are fastened to form an accordion-like unit with alternate flow sections through the stack for the warm and cold fluids. The entire stack is enclosed with sheet metal and provided with entrances and exits for the warm and cold fluids. All galvanized surfaces are painted with zinc chromate.

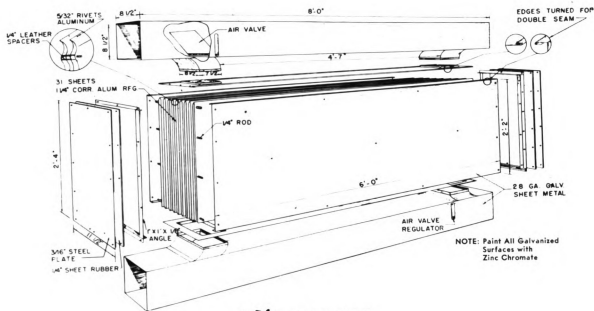


Fig. 34 plate-type heat exchanger

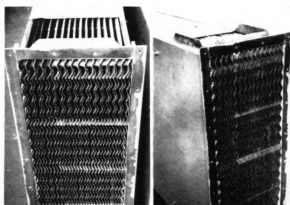


Fig. 35 End views of exchanger with end plate removed



Fig. 36 Laboratory test arrangement for plate-type heat exchanger

Laboratory testing has shown the heat transfer coefficient of such a cross flow heat exchanger to be:

$$h_c = 2.284 \times 10^{-3} \left(\frac{W}{cu} \right)^{0.657}$$

where h_c = Local heat transfer coefficient on either side
of separating wall, Btu/Hr, sq.ft, °F.

W = Fluid rate, lb/Hr

c = Fluid flow area, sq. ft.

u = Absolute viscosity of fluid, lb/Hr, ft.

The value of h_c corresponding to this equation may be found directly by reference to Figure 37 (Log log graph). Giese and Bond (35) found out that this experimental formula gives values quite close to the theoretical formula:

$$h_c = 3.23 \times 10^{-4} \left(\frac{2W}{cu} \right)^{0.75}$$

Laboratory tests also showed the static pressure to vary as the square of the capacity. Figure 37b was plotted from their results and shows the characteristic the blower must meet. It should be remembered that the blower must also overcome any additional friction in any ducts attached to the exchanger.

FIGURE 37_A

LOCAL HEAT TRANSFER
COEFFICIENTS FOR GIESE
AND BOND PLATE TYPE
HEAT EXCHANGER (35)

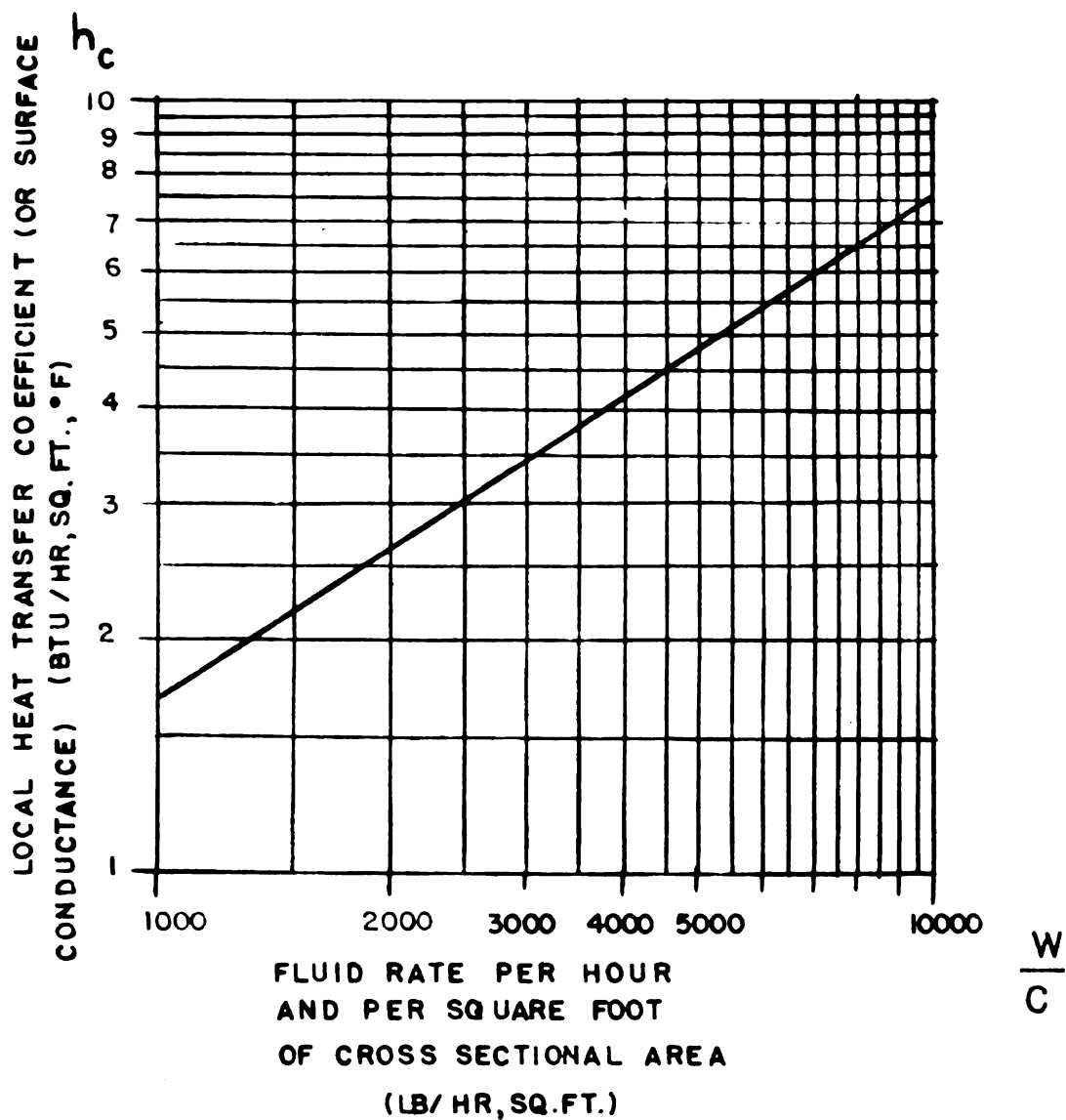
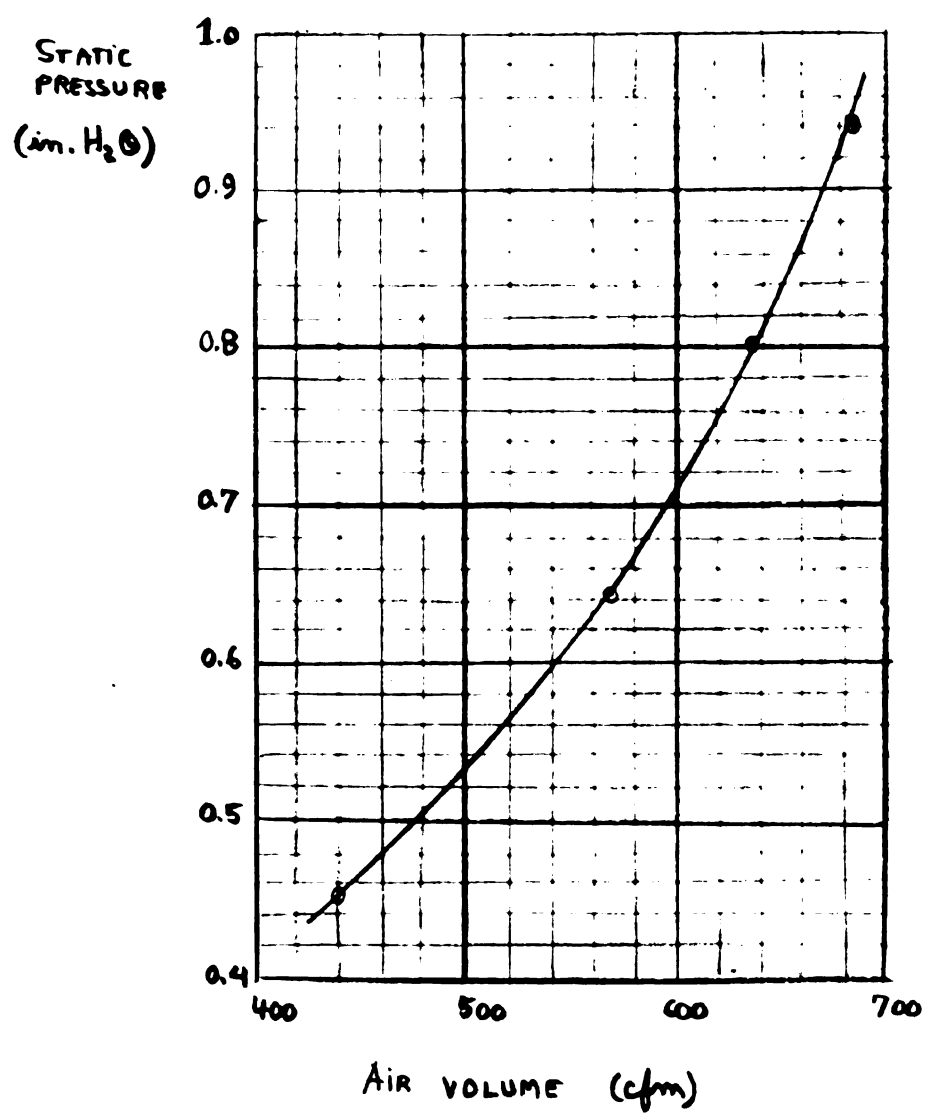


FIGURE 37b

BLOWER CHARACTERISTIC REQUIREMENT FOR GIESE AND BOND
PLATE TYPE HEAT EXCHANGER (35)



For the purpose of heat and water balance in poultry houses, Glase and Ford (24) give several charts which are presented here. The inside temperature is assumed as being 10°C . In the chart of Figure 30, the dotted lines represent the air-flow required to maintain a constant temperature of 10°C , while the solid line shows the air-flow requirement to remove the total moisture produced. The temperature lines are based on an assumption of 36 Btu of sensible heat per bird as given by Mitchell and Kelley (73) for a four-week old active bird not in production. It should be noted that since Mitchell and Kelley's values (73) are lower than the more recent values obtained by Gla, Harvey and Ashby (72) the heat production used is probably smaller than the actual heat production by about six Btu per hour per bird as discussed in a previous section. Furthermore, no secondary sources of heat are considered in this chart, although it is evident that solar heat and especially the heat from deep litter are usually very appreciable. The moisture removal line was constructed on the assumption of 75% inside relative humidity and 70% outside relative humidity and therefore the chart is limited to these conditions. Although the quantity of moisture produced is used to construct the heavy line is not given, the writer found it to be equal to 1.32 pound of water per hour per one hundred hens which correspond to thirty-three pounds per day per one hundred hens. This value is somewhat low for an active bird not in production and should be increased still more if the design is to be applied to poultry houses containing hens producing eggs.

The chart in Figure 32 is drawn to find the supplemental amount of heat required at different outside temperatures and for different insulating values, if the house is to be maintained at 10°C and if all the moisture is to be removed as soon as it is produced. Figure 30 being apparently in error, Figure 32 would also be in error. Further research on the total sensible heat available and the total amount of water to be removed from the

house is required to remove the heat loads and make the design of heat exchanger more accurate.

The following example is given to show the procedure to follow in designing a plate-type counterflow heat exchanger for a poultry house.

Application

Given: 1000-bird poultry house

Average weight of the birds = 4 pounds

Heat production of the birds = 35 Btu/hr, bird (from Mitchell and Miller) (72)

Moisture production of the birds = 32.3 lb/hr, 1000 birds (35)

Relative humidity to be maintained = 40%

Outside design temperature = 50°F

Wanted: The number and length of plates necessary for the construction of an aluminum plate-type counterflow heat exchanger, when using aluminum plates 2 inches wide separated by 1/4 inch spacers.

Solution:

Step 1: Find the minimum air flow required for moisture removal

(Refer to Figure 28, for $t_o = 50^\circ\text{F}$)

$$\begin{aligned} V_L \text{ (or } M_L) &= 4.40 \text{ lb/hr, bird} \\ &= 4400 \text{ lb/hr, 1000 birds} \end{aligned}$$

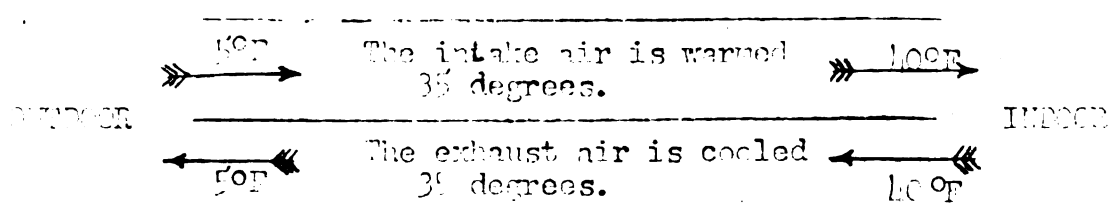
Step 2: Find the supplemental heat required, $Q_{\text{suppl. req.}}$ (Btu/hr).

(Refer to Figure 29, for $t_o = 50^\circ\text{F}$ and $AG = 1$)

$$\begin{aligned} Q_{\text{suppl. req.}} &= 26 \text{ Btu/hr, bird} \\ &= 26,000 \text{ Btu/hr, 1000 birds} \end{aligned}$$

Step 3: Check the maximum amount of heat which can be saved by the heat exchanger, when assuming a 100% efficiency, $Q_{\text{max. (100\%)}}$.

Schematic of a 100% efficient heat exchanger:



computation of the maximum heat saving possible

$$\begin{aligned} \text{Heat, (Btu.)} &= V_2 \times 1.01 \times \Delta t_{\text{max}} \\ &= 1.10 \times 1.01 \times 36 = 39.9 \text{ Btu/lr, bird} \\ &= 11,770 \text{ Btu/lr, 300 birds.} \end{aligned}$$

Interpretation of the results

Since the maximum amount of heat which can be saved by the heat exchanger, 11,770 Btu., is bigger than the amount of supplemental heat required, 9,000 Btu., it seems that the exchanger can be expected to save 2,770 Btu. to allow the house temperature to remain at or above 44°F with a ventilation rate of 1.10 lb./hr, bird (or approximately 2.3 acft./bird), when the outside temperature is 50°F. In other words, the heat otherwise lost in the exhaust air will be recovered and used to warm the house for the heat losses through the walls and ceiling of the house.

Step 4: Find the arithmetic mean temperature difference for design conditions *

$$\Delta t_{\text{arithm.}} = \frac{t_1 - t_2}{2} = \frac{51 - 35}{2} = \frac{16}{2} = 8.0^\circ \text{ (the arithmetic mean temperature difference)}$$

Step 5: Find the heat transfer characteristic requirement of the heat exchanger (A)

$$\text{Heat, (Btu.)} = AU \Delta t_{\text{arithm.}}$$

$$AU = \frac{\text{Heat, (Btu.)}}{\Delta t_{\text{arithm.}}} = \frac{12,800}{8.0} = 1,600 \text{ Btu/lr}$$

Step 6: Assume the number of aluminum sheets to use

Assume 11 sheets of 10-foot sheets

Step 7: Compute the fluid flow area

Derivation of the formula for the fluid flow area of 24-inch sheets separated by 1/4 inch spacers

$$\begin{aligned} \text{Fluid flow area} = C &= \frac{(24'' - 3'') \times (1/4'') \times \left(\frac{N}{2}\right)}{144} \\ &= \frac{21'' \times 1/4'' \times \frac{N}{2}}{144} = \frac{21 \times N}{4 \times 6 \times 2} = \frac{N}{16} \end{aligned}$$

Application of the formula

$$C = \frac{N}{16} = \frac{11}{16} = 0.6875 \text{ sq. ft.}$$

*The logarithmic mean temperature difference is usually used for counterflow heat exchanger design. However, according to Hise and Bond (35), the terminal temperature differences in the exchanger considered here are very nearly equal so that the arithmetic mean temperature difference, $\Delta t_{\text{arithm.}}$, can be used in place of the logarithmic mean temperature difference $\Delta t_{\text{log.}}$ between the fluids.

Step 8: Find the fluid velocity W in ft./hr.

$$\frac{W}{2} = \frac{2200}{2} = 1100 \text{ ft./hr. (ass. st.)}$$

Step 9: Find the value of the surface conductance h_c

(Refer to Figure 37, for $W = 1100 \text{ ft./hr. (ass. st.)}$)

$$h_c = 1.50$$

Step 10: Compute the overall heat transfer coefficient U

Derivation of the formula for the overall heat transfer coefficient

Since the local coefficient of heat transfer h_c for the warm and the cold side of the plates is approximately the same.

$$U = \frac{1}{\frac{1}{h} + \frac{1}{h}} = \frac{1}{\frac{2}{h}} = \frac{h}{2}$$

Application of the formula

$$U = \frac{h}{2} = \frac{1.50}{2} = 0.75$$

Step 11: Find the total heat transfer area required

$$A = \frac{Q}{U} = \frac{618}{0.75} = 824 \text{ sq. ft.}$$

Step 12: Check the number of standard 10-foot sheets required

Derivation of the formula for the number of standard 10-foot sheets required

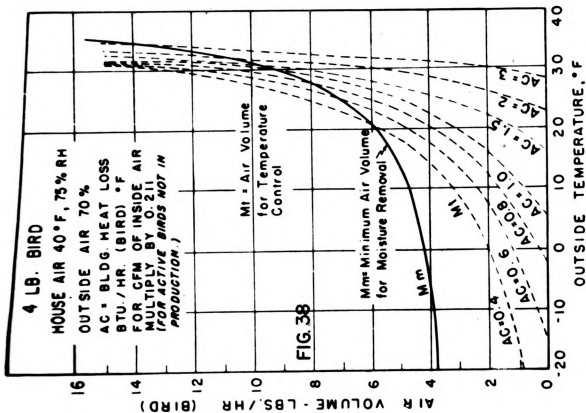
$$N = \frac{A}{21 \times 10} = \frac{A}{210 \text{ sq. ft.}} = \frac{A}{210}$$

Application of the formula

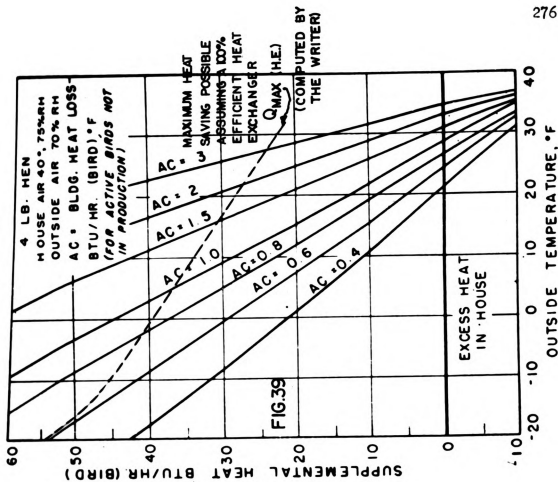
Since this is quite close from the assumed 14 sheets, the use of 14 sheets would be satisfactory.

Step 13: Select the capacity of the blower

The capacity of the blower will be determined by the range of air volumes required in the barn. For the exchanger designed and tested by Giese and Bond (35) the static pressure varied as the square of the capacity. Reference to Figure 37b will show the characteristics the blower must meet. A fan or blower catalog will give the size and power required for a particular need. It should be remembered that the blower must also overcome any additional friction in any ducts attached to the exchanger.



GIESE AND BOND VENTILATION CHART (35)



GIESE AND BOND CHART
FOR HEAT EXCHANGER DESIGN (35)

[illegible]

L. The Problem of Condensation on Walls and Ceilings Surfaces

Condensation will occur on a surface whenever the dew point temperature of the room air is greater than the wall surface temperature and will continue to occur until the weight of vapor decreases below the saturation-humidity ratio for water vapor at the given wall surface temperature. Since surface condensation is caused by water vapor coming into contact with the materials having surface temperatures below its dew point, two factors can be used to avoid it:

(1) Reduce as far as possible the dew-point temperature of the surrounding vapors (or in other words, reduce the vapor content of the air so as not to exceed the saturation-humidity ratio of the wall surface temperature). This is usually an operating problem and can be obtained by eliminating unnecessary water spillage and by providing effective ventilation for the removal of moisture vapor in the house. The control of vapor formation and its elimination from the space as soon as possible is the first requirement in most condensation problems.

(2) Increase the temperatures of the surfaces with which the vapor comes in contact. This is a construction rather than an operating problem and can be obtained by adding insulation to outside walls and ceiling, by the use of double sash windows and doors, and by the circulation of warmer air over the surface.

The use of a direct air stream from a fan against the surface also results in an increase of the surface conductance and a raising of the surface temperature (11).

For any set of temperature and humidity conditions, there is a definite relation between the condensation possibilities and the insulation of exposed parts of the structures. This relation has been presented in

a certain number of charts. Two charts previously published and a slide rule prepared by the writer are presented here.

1-Edgar chart for wall-surface condensation

In July 1949, Edgar (29) published a wall-surface condensation chart (Figure 40) based on psychrometric tables and the following formula:

$$R_t = 0.61 \left(\frac{t_i - t_o}{t_i - t_{dp}} \right) = 0.61 \left(\frac{\Delta t}{t_i - t_{dp}} \right)$$

$$\text{or } U = 1.65 \left(\frac{t_i - t_{dp}}{t_i - t_o} \right) = 1.65 \left(\frac{t_i - t_{dp}}{\Delta t} \right)$$

where R_t = Total resistance of the wall to heat transfer =

$$\frac{1}{U}$$

U = Overall heat transmission coefficient of the wall (Btu/Hr, sq.ft, °F)

t_i = Inside air temperature (°F)

t_o = Outside air temperature (°F)

Δt = Inside-outside air temperature difference (°F)

$t_{dp} = t_d = t_g$ = Wall surface temperature corresponding to the inside dew point temperature

0.61 = Air film heat transfer resistance = $\frac{1}{1.65}$

1.65 = Air film heat transfer coefficient
(Btu/Hr, °F, sq.ft)

In the computation of this chart the overall heat transmission coefficients (U) were considered over the range from 0 to 1.65, the outside temperature over the range from 40 to 100°F, and the inside temperature over the range from 32 to 100°F. With the value of t_{dp} computed, the corresponding critical relative humidity was obtained from a psychrometric table or a psychrometric chart, and the chart was plotted.

The chart may be considered as made of two superimposed charts. The first one would solve the equation

$$t_i - t_{dp} = \frac{U \times \Delta t}{1.65}$$

$$t_{dp} = t_i - U \left(\frac{\Delta t}{1.65} \right)$$

$$t_{dp} = K_1 - RK_2$$

For a given value of t_i and a given value of Δt , the value of t_{dp} varies in straight line with U as shown in the following graph,

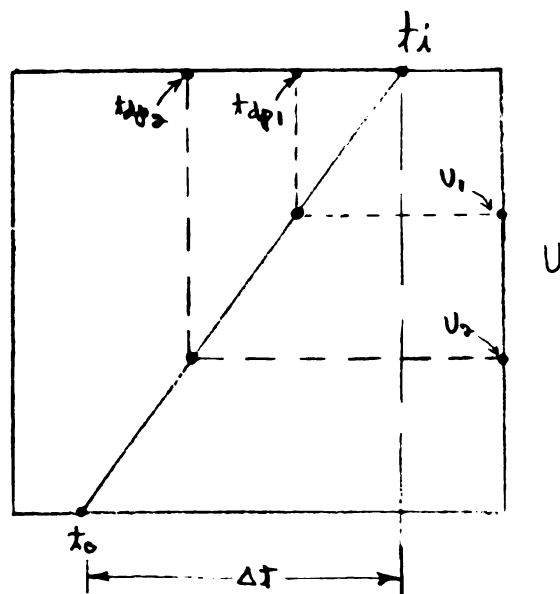
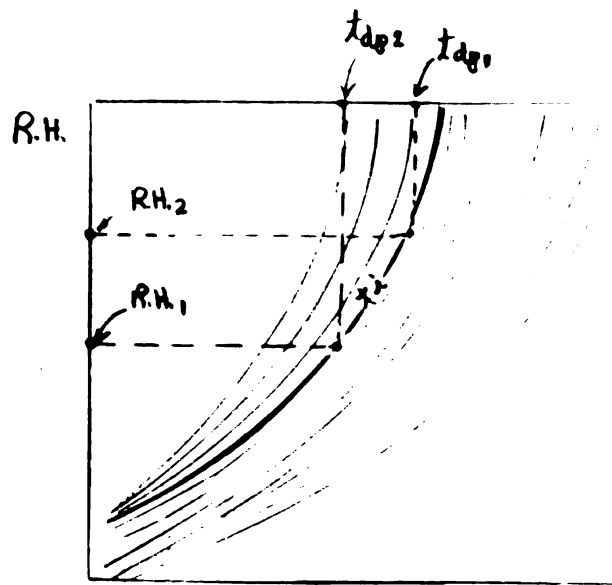


CHART TO SOLVE
THE EQUATION

$$t_{dp} = t_i - U \left(\frac{\Delta t}{1.65} \right)$$

From this graph it can be seen that the first step consists in finding " t_{dp} " by drawing a line from " t_i " to " t_o " and then drawing a horizontal line to the right from " U " to the first line and a horizontal line up from this intersecting point to " t_i ".

The second chart is derived from a psychrometric table or chart. It has t_d as one ordinate and "RH" as the other and " t_i " is shown as curved lines on the graph:



MODIFIED PSYCHROMETRIC CHART
TO FIND THE RELATIVE HUMIDITY
AT WHICH CONDENSATION WILL
OCCUR FOR GIVEN t_a AND t_{dp}
VALUES

The second step consists therefore in drawing a vertical line down from " t_{dp} " to " t_1 " and then drawing a horizontal line to the left, from the intersecting point to "R.H".

Since this chart is rather complicated because of this superposition of two different charts, the four following applications are presented as an illustration of its use.

Application No. 1

Given: Wall heat transfer coefficient (U) = 0.36

Inside temperature = 74°F

Outside temperature = 6°F

Wanted: The critical relative humidity which would cause condensation on the walls

Solution:

1-Draw a line from $t_i = 74^{\circ}\text{F}$ and $t_o = -6^{\circ}\text{F}$

2-Find $t_{dp} = 57^{\circ}\text{F}$ by going horizontally from $U = 0.36$ to this line and then vertically from the intersecting point to the t_i (and t_{dp}) scale.

3-Find $t_0 = 16$ by going vertically from " t_{dp} " = 57°F to " t_i " = 74 (curved line) and then horizontally from the intersecting point to the RH scale.

Application No. 2

Given: Wall heat transfer coefficient (U) = 0.09

Inside temperature = 50°F

Outside temperature = -15.5°F .

Wanted: The critical relative humidity which would cause condensation on the walls

Solution: Same procedure as for application No. 1. Answer = 89%

Application No. 3

Given: Wall heat transfer coefficient (U) = 0.09

Inside temperature = 50°F

Inside relative humidity = 89%

Wanted: The critical outside temperature which would cause condensation on the walls

Solution:

1-Find " t_{dp} " = 46 by going horizontally from RH = 89% to the curved line for $t_i = 50^\circ\text{F}$

2-Draw a line horizontally from $U = 0.09$ to the vertical line for " t_{dp} " = 46

3-Find $t_0 = -15.5^\circ\text{F}$ by drawing a line through $t_i = 50^\circ\text{F}$ and the intersecting point found in step 2.

Application No. 4

Given: Wall heat transfer coefficient (U) = 0.09

Inside temperature = 40°F

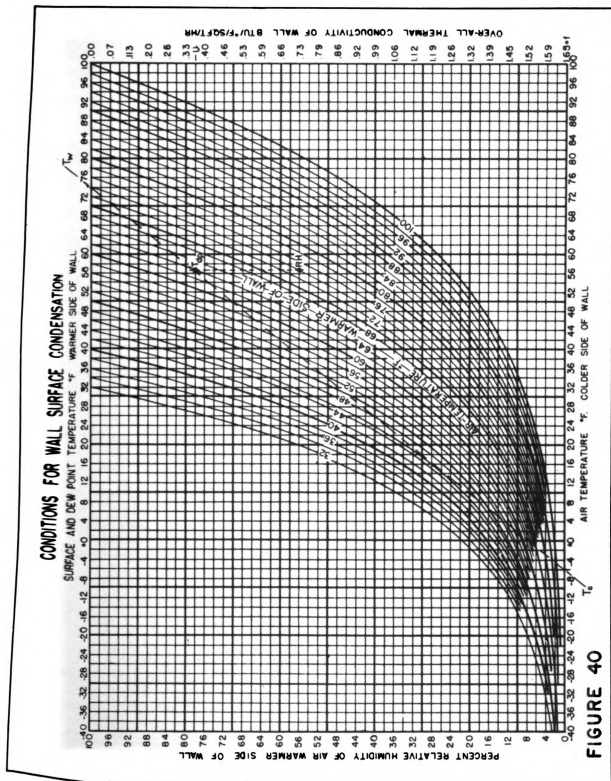
Inside relative humidity = 91%

Wanted: The critical outside temperature which would cause condensation on the walls

Solution:

- 1-Find " t_{dp} " = 3° by going horizontally from $W = 91.6$ to the curved line for $t_i = 40^{\circ}\text{F}$
- 2-Draw a line horizontally from $U = 0.02$ to the vertical line for $t_{dp} = 3^{\circ}$
- 3-Find $t_o = -9^{\circ}\text{F}$ by drawing a line through $t_i = 40^{\circ}\text{F}$ and the intersecting point found in step 2.

Edgar's chart has the advantage of requiring no computation and of being extremely time-saving in this respect. However, the writer always found it difficult to remember the procedure to follow when using it and always had some trouble with all the intersecting lines on it. For this reason the writer worked out a triple slide rule giving the same information. This slide rule will be presented later in this section.



EDGAR CHART FOR WALL SURFACE CONDENSATION (29)

2-The Insulation Board Institute chart for wall surface condensation

Also in 1912 the Insulation Board Institute (12) published a second type of chart (Figure 11) for wall surface condensation. Although the derivation of this chart is not given in the text the writer assumes that the same procedure was used as for the Edgar chart, except that the inside temperature was assumed fixed at a value probably between 40 and 50°F.

By assuming the inside temperature as being fixed, for example at 45°F, the four variable equation solved by the Edgar chart can be reduced to a three variable equation as shown here: $(t_i - t_d) = \frac{0.61 \Delta t}{R_t}$ or $(45 - t_d) = \frac{0.61 \Delta t}{R_t}$. Such a three variable equation has the advantage of yielding a simple type of chart very easy to read.

In the computation of this chart, the resistance value of the wall (R_t) was considered over the range from 2 to 17 and the inside-outside temperature difference over the range from 40 to 80°F. Once the value of t_d is found, the corresponding critical relative humidity is obtained from a psychrometric table or a psychrometric chart, and the chart plotted.

It can be seen from this chart that the greater the insulation value, the higher the critical relative humidity and the smaller the chances for condensation on the surface.

The two following applications are presented as an illustration of this chart.

Application No. 1

Given: Design inside relative humidity = 70%

Design inside temperature = 40°F

Design outside temperature = 20°F

Wanted: The insulation requirement to avoid wall surface condensation

Solution:

1-Find $t = 10 - (-20) = 30^{\circ}\text{F}$

2-Locate the relative humidity of 70% on the left-hand scale and draw a line horizontally to the right to the point where it intersects the 30 degree temperature difference curve

3-From this point, draw a line downward to the lower side, and find $R_t = 3.9$

Application No. 2

Given: Wall resistance = 4.70

Design inside temperature = 50°F

Outside design temperature = 20°F

Wanted: The maximum permissible relative humidity that can be carried for a particular type of construction.

Solution:

1-Find $t = 50 - (-20) = 70^{\circ}\text{F}$

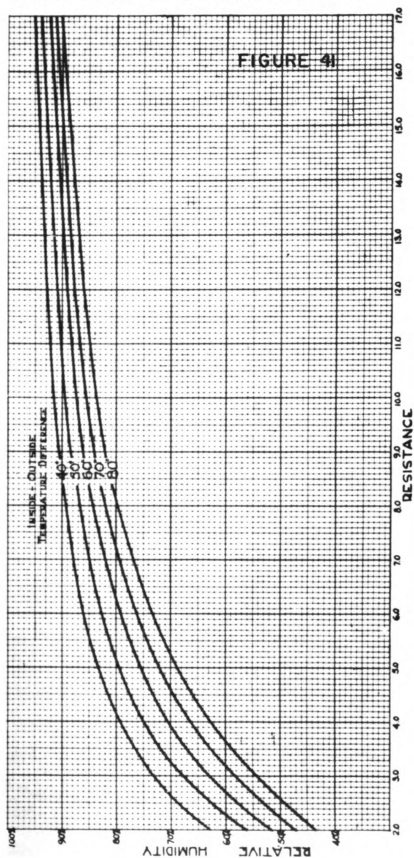
2-Locate 5.7 on the lower scale and draw a line vertically to the 70 degree temperature difference curve

3-From this intersecting point draw a line horizontally to the left-hand scale, and find RH = 75%

This chart has the big advantage of being easy to read. However, it offers some limitations since the inside temperature is assumed fixed at a value of probably 45°F , and no lines are drawn for temperature differentials of less than 40 degrees. Since the inside temperature is fixed, the writer believes that this chart should be used only as an approximation. For inside temperatures varying much from 45°F , Edgar's chart or the writer's slide rule should be used.

INSULATION BOARD INSTITUTE CHART FOR

WALL SURFACE CONDENSATION (49)



1717

2-Slide rule for wall-surface condensation

Because of the complexity of use of Edgar's chart, the writer thought of presenting this chart in a slide rule form. This slide rule is based on the same equation as the one used by Edgar:

$$R_t = 0.61 \left(\frac{t_i - t_o}{t_i - t_{dp}} \right) = \frac{0.61 \Delta t}{t_i - t_d}$$

By considering $t_i - t_o$ as equal to Δt , the equation is reduced to a four variable equation. This equation was separated into two equal parts each containing two variables and made equal to a third variable (dummy variable), say V .

$$1-R_t = \frac{0.61 \Delta t}{K}$$

$$2-K = t_i - t_d$$

The first equation was changed to a logarithmic form:

$$\begin{aligned} \log R_t &= \log 0.61 \Delta t - \log K = 0 \\ + (\log R_t) - (0) + (-\log 0.61 \Delta t) - (+\log K) &= (0) \end{aligned}$$

A middle support was required for conversion of $\log K$ values to K values, i.e. $\log t_i - t_{dp}$ values to $t_i - t_{dp}$ values. Equation 2 was arranged in the following order:

$$+(K) - (0) + (t_i) - (+t_{dp}) = (0)$$

The values of t_{dp} were plotted and the values of $W_{dp sat.}$ were substituted for them.

A middle support was required for conversion of $W_{dp sat.}$ values to $\log W_{dp sat.}$ values. The the last part of the slide rule was completed by using the equation:

$$R.H.C = \frac{W_{dp sat.}}{W_{i sat.}}$$

where $R.H._c$ = Critical relative humidity at which condensation will occur on wall-surface

$W_{dp_{sat}}$ = Moisture content of saturated air at the dew point temperature for the wall

$W_{i_{sat}}$ = Moisture content of saturated air at the inside air temperature

This equation was changed to a logarithmic form:

$$\log R.H._c = \log W_{dp_{sat}} - \log W_{i_{sat}}$$

$$+(\log W_{dp_{sat}}) - (0) + (-\log W_{i_{sat}}) - (\log R.H._c) = (0)$$

The values of $\log W_{i_{sat}}$ were plotted and the values of t_i were substituted for them.

The following applications are presented in order to illustrate the use of this slide rule.

Application No. 1

Given: Wall heat transfer resistance = 10

Inside temperature = 40°F

Outside temperature = 0°F

Wanted: The critical relative humidity which would cause condensation on the walls

True mathematical solution: (Heating, Ventilation, Air Conditioning Guide, 1951)

$$K = \frac{0.61 \Delta t}{R_t} = \frac{0.61 \times 40}{10} = 2.44^\circ$$

$$t_{dp} = t_i - K = 40 - 2.44 = 37.56^\circ\text{F}$$

$$W_{i_{sat}}(37.56^\circ\text{F}) = 47.2 \times 10^{-4} \text{ lb/Lb dry air (Use of slide rule No.7)}$$

$$W_{i_{sat}}(40^\circ\text{F}) = 52.13 \times 10^{-4} \text{ lb/Lb dry air}$$

$$R.H._c = \frac{47.2 \times 100}{52.13} = 90.5\%$$

Solution:

1-Put the first arrow opposite $R_t = 10$ and read $t_i - t_{dp} = 2.42$

opposite $\Delta t = 40$

2-Put the second arrow opposite $t_i - t_{dp} = 2.42$ and read $W_{dp} = 47.2$

opposite $t_i = 40^\circ\text{F}$

3-Put the third arrow opposite $W_{dp} = 47.2$ and read R.H.c 90.5%

opposite $t_i = 40^\circ\text{F}$

N.B. = By using Edgar's chart, the answer is 90%, and by using the Insulating Board Institute chart it is 91.5%.

Percentage of error given by the slide rule = 0

Application No. 2

Given: Wall heat transfer resistance = 10

Inside temperature = 50°F

Outside temperature = 0°F

Wanted: The critical relative humidity which would cause condensation on the walls

True mathematical solution: (Heating, Ventilating, Air Conditioning Guide, 1951)

$$K = \frac{0.61 \Delta t}{R_t} = \frac{0.61 \times 50}{10} = 3.05^\circ$$

$$t_{dp} = t_i - K = 50 - 3.05 = 46.95^\circ\text{F}$$

$$W_{i \text{ sat.}} (46.95^\circ\text{F}) = 68.2 \times 10^{-4} \text{ lb/Lb dry air (Use of slide rule No.7)}$$

$$W_{i \text{ sat.}} (50^\circ\text{F}) = 76.58 \times 10^{-4} \text{ lb/Lb dry air}$$

$$\text{R.H.c} = \frac{68.2 \times 100}{76.58} = 89.06\%$$

Solution:

1-Put the first arrow opposite $R_t = 10$ and read $t_i - t_{dp} = 3.04$

opposite $\Delta t = 50$

2-Put the second arrow opposite $t_i - t_{dp} = 3.04$ and read $W_{dp} = 68$

opposite $t_i = 50^\circ\text{F}$

3-Put the third arrow opposite $W_{dp} = 68$ and read $R.H._c = 89\%$

opposite $t_i = 50^\circ F$

N.B. = By using Edgar's chart the answer is 90% and by using the Insulating Board Institute chart it is 89.5%.

Percentage of error given by the slide rule:

$$\% \text{ error} = \left(\frac{89 - 89.06}{89.06} \right) 100 - \frac{0.06 \times 100}{89.06} = -0.07\%$$

Application No. 3

Given: Wall heat transfer resistance = 5

Inside temperature = $50^\circ F$

Outside temperature = $10^\circ F$

Wanted: The critical relative humidity which would cause condensation on the walls

True mathematical solution: (Heating, Ventilating, Air Conditioning Guide, 1951)

$$K = \frac{0.61 \times \Delta t}{R_t} = \frac{0.61 \times 40}{5} = 4.88^\circ$$

$$t_{dp} = t_i - K = 50 - 4.88 = 45.12^\circ F$$

$$W_{i \text{ sat.}} (45.12^\circ F) = 63.65 \times 10^{-4} \text{ lb/Lb dry air (Use of slide rule No.7)}$$

$$W_{i \text{ sat.}} (50^\circ F) = 76.58 \times 10^{-4} \text{ lb/Lb dry air}$$

$$R.H._c = \frac{63.65 \times 100}{76.58} = 83.1\%$$

Solution:

1-Put the first arrow opposite $R_t = 5$ and read $t_i - t_{dp} = 4.9$

opposite $\Delta t = 40$

2-Put the second arrow opposite $t_i - t_{dp} = 4.9$ and read $W_{dp} = 63$

opposite $t_i = 50^\circ F$

3-Put the third arrow opposite $W_{dp} = 63$ and read $R.H._c = 83\%$

opposite $t_i = 50^\circ F$

N.B. = By using Edgar's chart the answer is 84% and by using the Insulating Board Institute chart it is 83.5%.

Percentage of error given by the slide rule:

$$\% \text{ error} = \left(\frac{83 - 83.1}{83.1} \right) 100 = - \frac{0.1 \times 100}{83} = -0.12\%$$

Application No. 4

Given: Wall heat transfer resistance = 8

Inside temperature = 32°F

Outside temperature = -28°F

Wanted: The critical relative humidity which would cause condensation on the walls

True mathematical solution: (Heating, Ventilating, Air Conditioning Guide, 1951)

$$K = \frac{0.61 \times \Delta t}{R_t} = \frac{0.61 \times 60}{8} = \frac{36.6}{8} = 4.58^\circ$$

$$t_{dp} = t_i - K = 32 - 4.58 = 27.42^\circ\text{F}$$

$$W_{i \text{ sat.}} (27.42^\circ\text{F}) = 30.50 \times 10^{-4} \text{ lb/Lb dry air (Use of slide rule No. 7)}$$

$$W_{i \text{ sat.}} (32^\circ\text{F}) = 37.88 \times 10^{-4} \text{ lb/Lb dry air}$$

$$\text{R.H.}_c = \frac{30.50 \times 100}{37.88} = 80.5\%$$

Solution:

1-Put the first arrow opposite $R_t = 8$ and read $t_i - t_{dp} = 4.58$

opposite $\Delta t = 60$

2-Put the second arrow opposite $t_i - t_{dp} = 4.58$ and read $W_{dp} = 30.7$

opposite $t_i = 32^\circ\text{F}$

3-Put the third arrow opposite $W_{dp} = 30.7$ and read $\text{R.H.}_c = 81\%$

opposite $t_i = 32^\circ\text{F}$

N.B. = By using Edgar's chart, the answer is 81.5%. However, by using the Insulating Board Institute chart, the answer is 84.5%. This 3% difference is probably due to the fact that the Insulating Board Institute chart is based on a fixed inside temperature of around 45°F.

Percentage of error given by the slide rule:

$$\% \text{ error} = \left(\frac{81 - 80.5}{80.5} \right) 100 = + \frac{0.5 \times 100}{80.5} = +0.62\%$$

Application No. 5Given: Wall heat transfer resistance = 8

Inside temperature = 65°F

Outside temperature = 50°F

Wanted: The critical relative humidity which would cause condensation
on the wallsTrue mathematical solution: (Heating, Ventilating, Air Conditioning Guide, 1951)

$$K = \frac{0.61 \times \Delta t}{t_i - t_o} = \frac{0.61 \times 15}{65 - 50} = \frac{9.15}{15} = 1.13^\circ$$

$$t_{dp} = t_i - K = 65 - 1.13 = 63.87^\circ\text{F}$$

$$W_{i\text{sat.}}(63.87^\circ\text{F}) = 126.8 \times 10^{-4} \text{ lb/Lb dry air (Use of slide rule No.7)}$$

$$W_{i\text{sat.}}(65^\circ\text{F}) = 132.6 \times 10^{-4} \text{ lb/Lb dry air}$$

$$\text{R.H.}_c = \frac{126.8 \times 100}{132.6} = 95.7\%$$

Solution:1-Put the first arrow opposite $R_t = 8$ and read $t_i - t_{dp} = 1.13$ opposite $\Delta t = 15$ 2-Put the second arrow opposite $t_i - t_{dp} = 1.13$ and read $W_{dp} = 126.5$ opposite $t_i = 65^\circ\text{F}$ 3-Put the third arrow opposite $W_{dp} = 126.5$ and read $\text{R.H.}_c = 95\%$ opposite $t_i = 65^\circ\text{F}$ N.B. = By using Edgar's chart, the answer is 96%. It was impossible to use the Insulating Board Institute chart since this chart is only for temperature differentials of more than 40°F.Percentage or error given by the slide rule:

$$\% \text{ error} = \left(\frac{95 - 95.7}{95.7} \right) 100 = \frac{-0.7 \times 100}{95.7} = -0.73\%$$

A few of the advantages of this slide rules are:

- 1-No straight edge is required.
- 2-There are no superimposed curves.
- 3-This slide rule is good for any inside temperature and any inside percent of relative humidity.

Edgar's chart, the Insulation Board Institute chart and the writer's slide rule were all prepared on the assumption of an inside air film heat conductivity of 1.65 or an air film resistance of approximately 0.61. An interesting fact to consider is the effect of variations in the value of the inside surface coefficient on this problem of condensation.

Strahan and Marsh (97) state in this respect: "It has been observed that smooth inside surfaces seem to show condensation more quickly than comparatively rough ones. In our own experience we have observed this on smooth plaster walls whereas apparently similar conditions did not produce condensation on painted wood walls." These authors presented some condensation charts for heat conductivity of 1.6, 1.4, 1.2 and 1.0. These charts are not presented here because they are limited to walls having a total heat resistance of 6.25 and do not apply to other walls. According to Strahan and Marsh (97) a rough wall (wood) is represented by an approximate air film heat transfer coefficient of 1.4 while a smooth plaster wall is represented by an approximate value of 1.0.

More specifically, it can be said that the air film heat transfer coefficient is affected by the temperature and emissivity of the surface, air velocity, and temperature difference between the surface and the air. Figure 42 gives the heat transfer coefficients (surface conductance) for different materials and air speeds at 20°F mean temperature.

FIGURE 42

HEAT TRANSFER COEFFI-
CIENTS (SURFACE CONDUCT-
ANCE FOR DIFFERENT MATERIALS
AND AIR SPEEDS (AT 20°F.)*

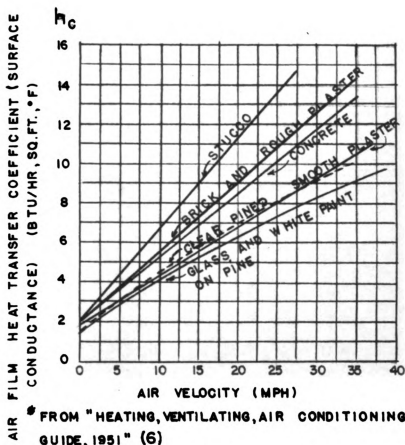


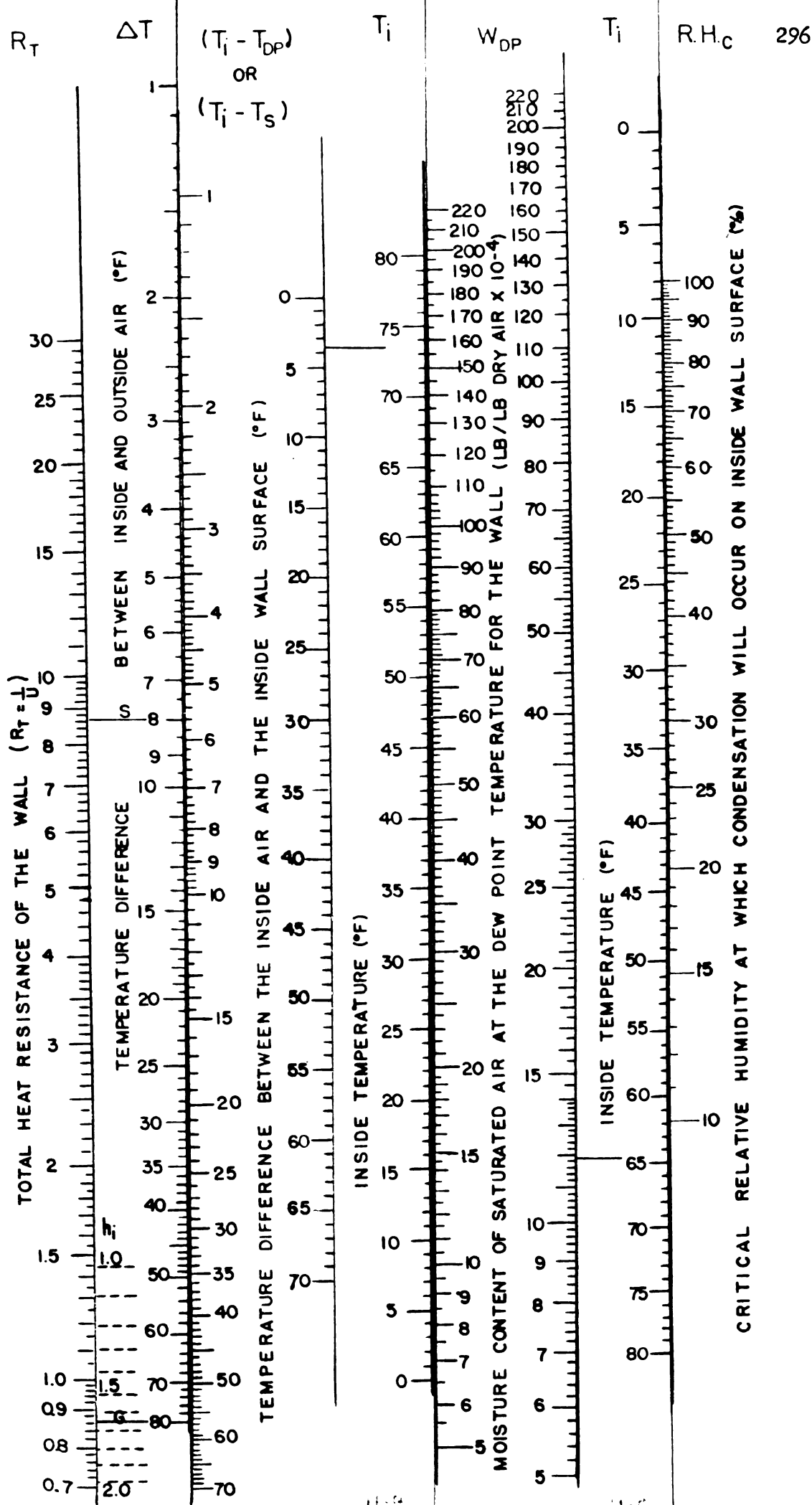
Table (55) illustrates the effect of air motion by stating: A.W. Clyde found that when the stable air is in motion the condensation point on the walls is from 1 to 4°F below the dew point in still air. This may be important if the prevention of condensation is a controlling design factor."

Because of the possible variation in the inside air film heat transfer coefficient, the writer included on his slide rule values of h_i from 1.0 to 2.0. A comparison is made here (Table 49) of the critical relative humidity for walls of 1.65, 1.4, 1.2 and 1.0 air film coefficients, to show the effect of variation of the air film heat transfer coefficient on condensation.

Table 49. Effect of inside air film heat transfer coefficient on the critical relative humidity causing condensation on wall and ceiling surfaces.*

Conditions				Inside air film heat transfer coefficient (h_i)				Maximum difference
				1.65	1.40	1.20	1.00	
				Inside air film heat transfer resistance				
R_t	t_i	t_o	Δt	0.606	0.715	0.833	1.00	(% R.H.)
8	65	50	15	95.5	95.0	94.2	93.5	2.0
10	40	0	40	90.5	89.5	87.0	84.5	6.0
10	50	0	50	89.5	87.8	85.2	82.0	7.5
5	50	10	40	83.2	82.0	78.5	74.0	9.2
8	32	-28	60	81.2	78	75.2	71.3	9.9

* Computed by use of the writer's slide rule.



SLIDE RULE FOR DETERMINING WALL SURFACE CONDENSATION



M. The Problem of Condensation Within Walls and Ceilings

Condensation within the structure may be considered as surface condensation transferred to the interior parts of the structure. In fact, an analysis of temperature gradient through a composite wall show that if no barrier is opposed to the flow of moisture vapor through the wall, this vapor will finally reach a plane where the temperature corresponds to its dew point and condensation will occur within the wall. It is therefore very important to prevent vapor from getting to the interior sections of a wall.

Barre and Sammet (11) list as follow the conditions favorable to condensation of moisture vapor within walls:

- 1-A large vapor pressure difference between the two sides of the wall
- 2-A high vapor permeability of the warm side material with a relatively lower permeability on the cold side
- 3-A large temperature difference on opposite sides of the wall (or a high insulating value in the wall).

To control condensation within walls, three ways are therefore possible:

- 1-Reducing the vapor pressure on the warm side (Proper rate of ventilation)
- 2-Increasing the vapor resistance on the warm side of the wall (Use of vapor barriers)
- 3-Reducing the amount of insulation (Smaller temperature difference)

Since insulation is desirable for the purpose of temperature control, only the first two of the above methods of control are recommended.

Building materials offer a resistance to water vapor flow in a way somewhat similar to the resistance they offer to heat flow, and vapor diffusion is from the region of higher vapor pressure to the region of lower vapor pressure, just like the heat flow is from the region of higher temperature to the region of lower temperature. Therefore, the theory of vapor

transmission may be compared to that of heat transmission, and formulas similar to those used for heat transmission may be derived.

$$Q = AU (T_i - T_o) \quad (\text{Heat transfer formula}) \quad (\text{Symbols not explained here})$$

$$W = Au (P_1 - P_2) \quad (\text{Moisture transfer formula})$$

where W = Total moisture vapor flow through the walls
(grains/hr)

u = Overall permeability (or moisture transfer coefficient) of the wall. (grains/hr, sq.ft, inch Hg)

A = Area of the wall (sq.ft)

P_1 = Vapor pressure on the humid side of the wall
(inch Hg)

P_2 = Vapor pressure on the other side of the wall
(inch Hg)

$$U = \frac{1}{\frac{1}{C_1} + \frac{1}{C_2} + \frac{1}{C_3} + \dots + \frac{1}{C_n}} \quad (\text{Heat transfer formula}) \quad (\text{Symbols not explained here})$$

$$u = \frac{1}{\frac{1}{u_1} + \frac{1}{u_2} + \frac{1}{u_3} + \dots + \frac{1}{u_n}} \quad (\text{Moisture transfer formula})$$

where u = Overall permeability of the wall (grains/hr, sq.ft. inch Hg)

$u_1 u_2 u_3 u_n$ = Vapor permeability of the different layer of material forming the composite wall.

Permeability (u or K) is expressed in terms of grains of water transmitted per hour per square foot of surface, for a vapor pressure difference of one inch of mercury or one pound per square inch (psi) and a thickness of material as indicated. Reciprocals of permeability, $D = \frac{1}{u} = \frac{1}{K}$ are designed as vapor resistances.

In the application of the latter formula; it is assumed that the permeability is directly proportional to the vapor pressure drop between two different planes and that the resistance to vapor is additive for several materials in series. This theory is not true:

1-When the vapor encounters change to a liquid or solid state. Such a change of state completely changes the mechanism by which moisture is transferred thorough the material.

2-When the materials are hygroscopic. Such materials absorbed vapor somewhat in the proportion to the relative humidity, and not directly proportional to the pressure of the vapor in contact with the material.

A further point to be considered is that the vapor pressure or dew point temperature drop per degree of temperature drop is much greater in high temperature than it is in low temperature ranges.

Due to the uncertainties as to the exact mechanism for the transfer of vapor through various types of structure, the A.S.H.V.E. Guide 1950 (5) recommends the application of the above formulas to be used with caution. The following values or permeability for various materials can be used as a guide (Table 50).

Table 50. Permeability of various materials to water vapor *

Material	Permeability (Grains/Hr, sq.ft.inch Hg and given thickness)	Authority
1-Poil-surfaced reflective insulation, double-faced	0.08 to 0.13	b
2-Gypsum lath with metallic aluminum backing	0.09 to 0.39	b
3-Poil roofing - smooth, 40 to 45 lb per roll 108 sq.ft.	0.13 to 0.17	b
4-Sheathing paper, asphalt impregnated glossy	0.17 to 2.05	b
5-Plywood, 2 coats of asphalt paint	0.43	b
6-Duplex or laminated papers, 30-60-30	0.52 to 0.86	b
7-Duplex paper, coated with metallic oxides	0.52 to 1.29	b
8-Insulation backup paper, treated	0.86 to 3.42	b
9-Brick masonry, 4 in.	1.1	a
10-Plaster, 2 coats of aluminum paint	1.15	b
11-Plywood, 2 coats of aluminum paint	1.29	b
12-Duplex or laminated papers 30-30-30	1.37 to 2.58	b
13-Fir sheathing, $\frac{3}{4}$ in.	2.9	a
14-Insulating sheathing, surface-coated	3.9 to 4.36	b
15-Paint film	3.4	a
16-Plaster, 3 coats of lead and oil	3.68 to 3.84	b
17-Pine, lap siding	4.9	a
18-Insulating cork blocks, 1 in.	6.19	b
19-Plaster, wood lath	11.0	b
20-Sugar cane fiberboard, $\frac{3}{4}$ in.	12.5	a
21-Plaster base and plaster, $\frac{3}{4}$ in.	14.7	a
22-Plaster, fiberboard or gypsum lath	19.73 to 20.57	b
23-Insulating lath and sheathing, board type	25.68 to 34.27	b
24-Mineral wool, unprotected, 4 in.	29.07	b

* A.S.H.V.E. Guide 1950 (5)

a) From data by L.G. Miller (69), Calculating Vapor and heat transfer through walls, Heating and Ventilating, Vol.35, No. 11, November, 1938.

b) From data by L.V. Teesdale (104), How to overcome condensation in building walls and others, Heating and Ventilating, Vol.36, No.4, April, 1939.

Barre (9) also presents some values as a result of his extensive test on vapor permeability of different materials. The values he found are presented here (Table 51):

Table 51. Permeability of various materials to water vapor *

Description of material	Thick- ness (in)	Vapor permeability (Grains/lr, sq.ft, psi) and given thickness K	Vapor resistance $D = \frac{1}{K}$
a-BUILDING BOARDS			
1-White pine board	3/4	2.83	0.353
2-Cedar bevel siding	-	3.69	0.271
3-Cedar bevel siding with 3 coats white lead paint	-	2.06	0.485
4-Douglas fir plywood (G2S)	1/4	7.48	0.133
5-Douglas fir plywood (G2S)	3/8	2.19	0.108
6-Douglas fir plywood (G2S) with 2 coats white lead paint	1/4	3.95	0.253
b-BUILDING PAPERS - ROOFING PAPERS			
1-Red rosin sheathing paper	-	123.13	0.008
2-Asphalt-saturated felt (15 lb)	-	2.72	0.368
3-Asphalt-saturated felt (30 lb)	-	1.80	0.556
4-Roofing paper (51 lb)	-	0.12	8.34***
5-Roofing paper (69 lb)	-	0.06	16.7***
c-INSULATION BOARDS			
1-Homosote	1/2	32.77	0.026
2-Homosote with 1 coat aluminum paint	1/2	24.56	0.041
3-Homosote with 2 coats aluminum paint	1/2	0	∞
4-Celotex vapor-seal insu- lating sheathing	25/32	3.99	0.251
5-Celotex insulating lath	1/2	58.13	0.017
6-Celotex with 1 coat aluminum paint	1/2	13.95	0.072
7-Celotex with 2 coats aluminum paint	1/2	0.72	1.388
8-Structural insulating board (uncoated)	1/2	47.45	0.021**
9-Structural insulating board (coated)	1/2	5.14	0.192**

d-PLASTER

1-Plaster on wood lath (3 coats)	3/4	5.81	0.172
2-Plaster on wood lath (3 coats) with 2 coats aluminum paint	3/4	0	∞
3-Plaster on 3/8" gypsum lath	-	22.9	0.044 ***
4-Plaster on 1/2" ins. lath	1	7.53	0.133
5-Plaster on 1/2" ins. lath painted (2 coats)	1	1.72	0.572

e-MASONRY MATERIALS

1-Brick-wall section laid up with mortar	4	1.61	0.621
2-Tile-wall section laid up with mortar concrete	4	0.25	4.000
3-Concrete	1 1/2	3.38	0.296
4-Concrete with 2 coats liquid-coating asphalt	1 1/2	3.18	0.314

f-FILL INSULATION

1-Rock wool	1	51.89	0.019
2-Sawdust (D.F.)	2	64.3	0.016

g-METAL SHEET

h-STILL AIR	-	257.00	0.004 ***
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* From data by H.J. Barre (9)

* * F.B. Rowley and G.E. Lund (86), Vapor transmission analysis of structural insulating boards, Bull. 22, Univ. of Minnesota, Eng. Expt., Sta., 1944.

* * * Figures presented by H.J. Barre (9) without mention of their origin.

Barre (9) found that laboratory tests indicate one noteworthy variation from the theoretical analysis: "In walls containing a highly permeable insulation, condensation occurs not at a theoretical dew point within the insulation, but at the less permeable boundary on the cold side of the wall. This is explained in the following figure." (Figure 43)

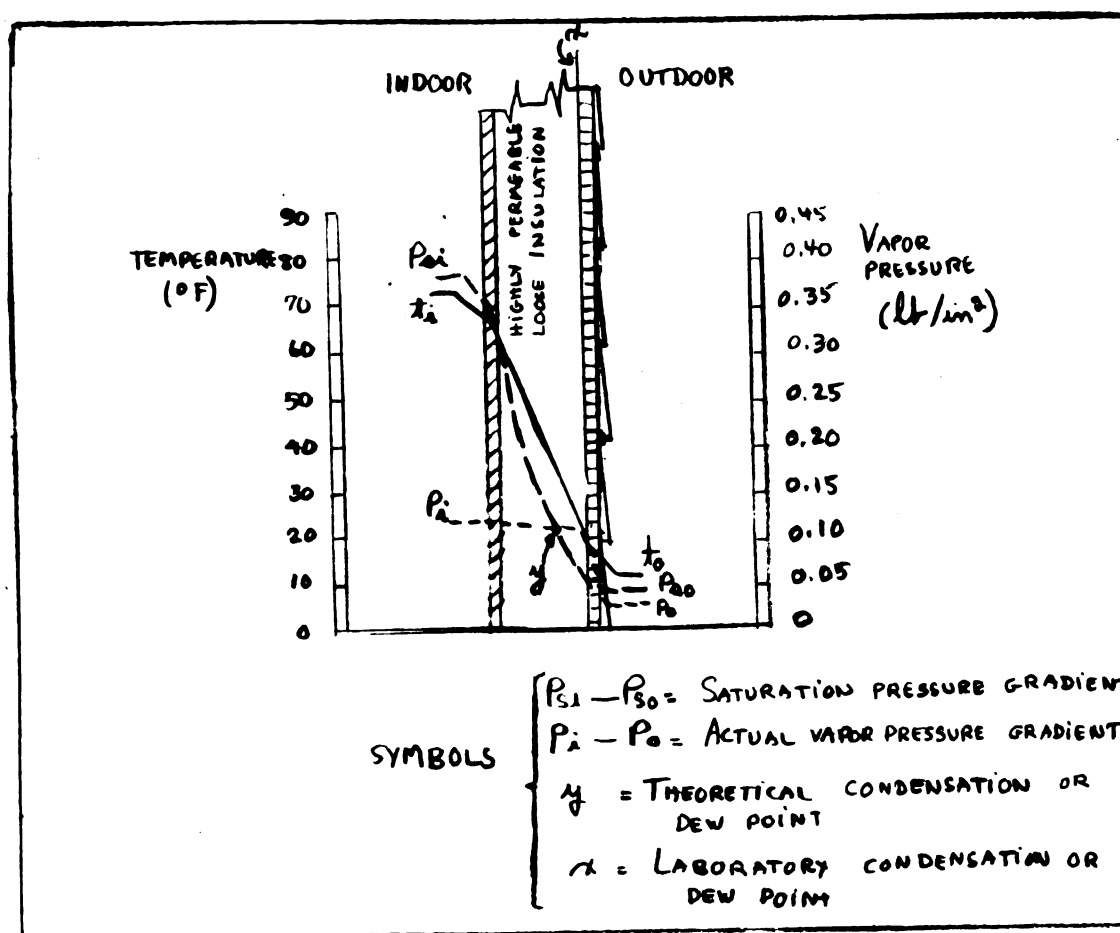
Barre (9) also mentions that for a particular material, vapor permeability varies according to the physical conditions.

1-At relative humidity of less than 75 to 80%, the rate of vapor transfer, in general, is proportional to the vapor pressure

2-But at relative humidity higher than 75 to 80% this is not true, relative humidity being the governing factor rather than vapor pressure

FIGURE 1.3

TEMPERATURE AND VAPOR PRESSURE IN AN INSULATED WALL *



* From Barre and Sammet (11)

3-The vapor permeability appears to be increased by wetting the material

4-The vapor permeability seems to increase as the temperature drops to 32°F but again decrease at temperatures below 32°F.

He concludes that: "the lack of complete correspondence in the test data and variations due to temperature and relative humidity affect the reliability of computations in which the permeability data are used. For this reason, and because of the nature of the practical problems to which the data may be applied, calculated vapor flow should be regarded as an indicative rather than an absolute quantity."

From the above permeability tests, it can be said that:

1-Loose-fill insulation and uncoated insulating boards are relatively permeable, as are lightweight asphalt-saturated and lightweight rosin-saturated building felts and uncoated plaster

2-Unpainted wood appears to be moderately permeable

3-Materials such as heavy roofing felt weighting forty or more pounds per square (1 square = 100 ft²), surfaces having two or more coats of aluminum paint in a varnish vehicle, and building felts saturated and coated with asphalt appear to be relatively impermeable

4-Sheet metal and glass may be regarded as completely impermeable

Goodman (38) states: "Tests made by United States Forest Product Laboratory indicate that any of a number of materials may be built into a wall to make satisfactory vapor barriers. Among these are:

1-Metallic surfaced reflective paper or foils

2-Gypsum lath backed with aluminum foil

3-Asphalt saturated, glossy finished sheathing paper

4-Smooth-surfaces rolled asphalt roofing

"Laminated kraft paper 30-40-30 and 30-50-30, which are made up of two layers of kraft paper with asphalt between. (The 40 and 50 in the grade identification numbers refer to the number of pounds of asphalt to each five hundred square feet of the finished product)

Tarred paper does not make a satisfactory vapor barrier. Asphalt is specified."

Ashby, Miller et al (4) recommend the following: Shiny heavily asphalted paper, aluminum foil lightweight roll roofing, and two coats of asphalt or aluminum-flake paint.

In the application of vapor barriers, the following fundamental principles should be remembered:

- 1-The vapor barrier should be placed as near to the warm surface of the wall as practicable
- 2-The vapor barrier should be continuous with no direct openings through the barrier. Usually a two-inch lap over a framing member will make a sufficiently tight joint for membrane barriers
- 3-All openings for electrical fixtures and joints around window and door casings should be carefully sealed.
- 4-Space packed with fill insulation should breathe to the cold (outer or upper) side to assist in drying out moisture that might penetrate through the barrier.
- 5-On the outer side of the wall a good felt building paper can be used. Its purpose on the outer side is to keep wind and rain out. It should not be vapor proof.

W. The Problem of Condensation on The Interior Surface of Cold Attic Walls

Control of condensation on the interior surface of cold attic walls may be obtained in two ways:

- 1-By application of vapor barrier to the warm side of the ceiling
- 2-By ventilation of the attic

The first method is preferable from the standpoint of heat conservation since ventilation is always a source of heat loss. However even with vapor barrier a certain degree of ventilation is necessary. Attic ventilation is also useful to keep the roof cool in the summer.

When ventilating the attics, precautions must be taken to see that the air is adequately distributed throughout the space to be ventilated. For residential gravity attic ventilation the A.S.H.V.E. Guide 1950 (5) recommends the total area of the inlet openings to be the same as that of the outlet openings and correspond to one-quarter square inch per square foot of floor area. They recommend the distribution of these openings with due regard to the type of construction, outside wind velocities, and all factors which affect the circulation of air. In animal shelters, ventilation of the attic is usually done by having only two big openings covered with louvres under the opposite eaves. Sometimes, however, vents are used in the cornice in conjunction with louvres or ventilators at the high point of the roof (68).

O. The Wet Litter Problem

The primary problem of the poultryman is to keep the floor litter dry (below 35 or 40% moisture content, wet basis) (4), since this will eliminate the job of changing the litter, reduce the number of dirty eggs and tend to provide more sanitary conditions for the birds. Proper insulation and ventilation are absolutely necessary for the maintenance of dry litter, since they result in the following three factors all of which are conducive to dry litter conditions:

- 1-Heat conservation
- 2-Temperature control
- 3-Proper amount of water vapor removal

Any scientific attempt to control the condition of litter in the poultry house must presuppose a knowledge of the moisture and heat problem with which one is dealing. This involves a study of the moisture and heat produced or taken into the poultry house by all agencies and the means by which it may be removed from the house. A part of this study was done in the section dealing with "Heat and moisture production of hens" and the one on "Secondary sources of heat and moisture production in poultry houses." The factors other than insulation and ordinary ventilation affecting the evaporation of this moisture will be discussed here.

1-Unfavorable effect of bird concentration on litter condition

According to Bates (12) bird concentration is a factor of great importance in litter condition. He states: "In the houses studied the litter remained relatively dry under the roost where the birds spent less time and the litter tended to be wet in the front of the house where they concentrated. This would add to the belief that moisture concentration in litter is not a problem of condensation but one of removing the moisture deposited by the droppings."

2-Basic principles of water evaporation from poultry houses litters

The water eliminated in the droppings must first be evaporated from the litter before it can be removed by the ventilation system. At least four factors affect the rate of evaporation of water from a free water surface or from a wet litter. They are:

- 1-The difference between the vapor pressure at the water surface (or in the wet litter) and the vapor pressure of the overlying air
- 2-The amount of exposed surface area
- 3-The amount of air movement
- 4-The amount of heat transferred to the litter or generated in the litter.

Parker (32) consulted a bacteriologist, Dr. F.S. Orcutt of Virginia Polytechnic Institute, concerning the decomposition and possible heat production of built-up litter. According to Dr. Orcutt: "Anaerobic bacteria produce considerably more heat than aerobic; therefore, conditions that promote the growth of anaerobic bacteria should be best from the standpoint of heat production in the litter. These conditions are:

- 1-No oxygen
- 2-High temperature
- 3-High moisture content and
- 4-Large number of bacteria for starting the process."

Mitchell and Kelley (73) stated that there is some doubt as to the amount of water that can be removed from the litter by ventilation. Much depends on the floor temperature, the vapor pressure balance of the air entrapped in the litter and that just above, and depth and kind of litter.

Dann (27) reported that the rate of evaporation of water from litter depends on the available heat, the rate of air movement over the litter, the depth of the litter and the frequency of turnover. He also recommended the use of insulated and water-proofed floors because: "Litter is

a very porous material holding much entrained air. Whenever, the temperature of the litter is higher than the air above it the entrained air, due to its temperature, will hold a large number of vapor particles and bring its vapor pressure in equilibrium with that of the remaining moisture in the litter and therefore retard evaporation from deep litter when left in a quiet state. Thus the heat required to maintain the process should be supplied from the air above the litter." This statement seems to be directly contrary to conclusions from experiments on the heating of poultry house floors (70) (13).

According to Shier (92), the dampness of the litter and bedding of poultry and farrowing houses is often caused by condensation of moisture from the air whenever the floor of the house becomes cold enough to be below the dew point of the air in the house. He adds: "In Ohio the remedy usually is better insulation to conserve heat and a suitable ventilation system. A small amount of condensation on the litter is not harmful. The use of vapor-proof paper in the floor will not prevent damp litter, and may in some cases increase the dampness in the litter by preventing the moisture from moving downward to the colder earth below."

3-Special study of some of the factors affecting favorably the rate of water evaporation from poultry house litters

From the basic principles just mentioned it can be concluded that the following points are considered aids to evaporation of litter moisture:

1-Lower absolute humidity of the air

2-Higher moisture content of litter, up to a point near saturation

3-Higher temperature of litter (Heat is a critical factor in evaporation; if the litter is to remain dry, heat must be transferred to the litter or generated in it).

4-Air movement to help bring air of lower absolute humidity into contact with the litter (Use of recirculating fans)

5-Stirring of litter to help bring litter of higher moisture content into contact with the air

6-Maintenance of a rough surface to help provide a greater surface area exposed to the air

7-Higher heat production in the house (Higher bird population, insulated glass and material, use of heat exchanger, etc.)

Factors 1, 2, 3 and 7 above are evident or have already been discussed.

The factors to be analyzed here are:

a-Effect of the type of litter

b-Effect of the depth of litter (Use of deep litter)

c-Effect of the stirring of litter

d-Effect of the use of recirculating fans

a-Effect of the type of litter. Charles, Tepper et al (19) made several tests of the moisture holding capacity of different plot house litters. They mentioned that these results can only be considered approximate, but give an indication of the relative adsorptive values. These results are presented in the following table (Table 52):

Table 52. Moisture holding ability of litters *

Litter	Absorbtive ability (Lb of water absorbed per lb of litter material)
1-Peat Moss	4.5
2-Sugar Cane (Coarse)	4.4
3-Straw	4.4
4-Sugar Cane (fine)	4.2
5-Peat Moss and Straw	4.2
6-Peat Moss and Shavings	3.7
7-Cardust	3.3
8-Hay	3.2
9-Shavings	2.5
10-Oat Hulls	1.4
11-Sand and Gravel	0.14

* From data by Charles, Tepper et al (19)

According to these results, the peat moss litter absorbed and retained the highest moisture content of any of the litters used and sand and gravel litter retained the least moisture throughout the experiment. They add however, that it is still to be established whether or not it is desirable to select for poultry litter a material having a relatively high absorptive capacity for water which retains absorbed moisture, or one having a low absorptive capacity and a low moisture retention value.

From his conversations with poultrymen and from his own observations, the writer has a tendency to believe that the best type is probably an intermediate one. This opinion is based on the fact that some poultrymen experiencing some trouble with straw (high water absorption) were better satisfied with their litter condition after using shavings (intermediate water absorption). It is to be noted, however, that the ease with which shavings are stirred by the hens, as compared to straw, may be of some importance in explaining the better results obtained with shavings.

b. Effect of the depth of litter (Use of built-up litter). Built-up litter, as compared to shallow litter, seems to be considerably more suitable for the evaporation of liquid water.

Dann (27) however, makes the following remarks concerning the depth of the litter: "We have found that litter over four inches deep is not to be desired. It acts as a cushion of entrained air pockets which become saturated with moisture and through which evaporation proceeds slowly or not at all." He recommends stirring of the litter either mechanically or by the hens in order to break up these moisture saturated air pockets.

Winter (108), Kennard et al (61) (62) and James (50) advocate built-up on the basis that

- 1-It produces some heat
- 2-It remains dryer

3-It does not have to be changed until the end of the season.

These men also present information on built-up litter management practices.

White and Schwantes (197) who conducted tests on well insulated laying houses in Minnesota, stated that in four houses the litter became objectionably wet until built-up litter was used. After this the litter remained in good condition. In one other similar house the litter became objectionably wet and was changed every three to five weeks.

Moyer and Blickle (68) state: "Covering the floor with 6 to 8 inches of litter early in the fall and adding more litter regularly during the winter is widely practiced to combat wet litter."

One of the advantages of deep litter seems to be that deep litter, finely broken, has the capacity to store excess moisture not removed by the ventilation system in periods of unfavorable outdoor humidity, without packing, and to release it to the ventilation system in periods of favorable drying weather. Shallow litter, to the contrary, has a tendency to pack with an increase in moisture content and does not have the ability to release the moisture so readily because of this packed condition.

c. Effect of litter stirring. The frequency of turnover of litter also has a marked effect on evaporation. It breaks up the moisture saturated air pockets of packed wet litters and exposes the different wet surfaces of the litter to the air above it.

The litter can either be stirred up by hand or by the use of a special stirring machine. However, probably the easiest way consists of having the hens stir the litter themselves by feeding them grain directly in the litter. During cold or damp weather, the grain could be fed in smaller quantity and at more frequent intervals during the day.

d. Effect of the use of recirculating fans. The principle of the use of recirculating fans is explained by Bates (12) as follows: "Since the air near the ceiling in a poultry house is usually lower in relative humidity and higher in temperature than the air near the floor, it was thought that by having a more thorough mixing of the air within the house, the strata of warm air near the ceiling would be broken and mixed with the cooler air at the floor. By this action the temperature of the air at the floor would be slightly increased as would its ability to hold moisture. It was thought that by this action more moisture could be removed from the litter of a poultry house than had previously been done".

Discussing air movement as an aid to evaporation of water from litter, Dann (27) states: "As to the rate of air movement over litter we have found the optimum to be about 5 to 6 feet per minute (or 300 to 360 feet per hour), which creates no perceptible draft and yet changes the air sufficiently often to keep it pure and to move off the moisture-laden atmosphere."

Bates (12) concluded from his tests and theoretical computations on the use of recirculating fans in poultry houses:

1-The moisture content of the litter in house A (insulated house) did not increase after the recirculation system was installed while there was an increase of 3% in the moisture content of the litter in B, the check house. This fact seemed significant.

2-The recirculation of air at the rate of 2.45 cfm per heavy hen brought about a more uniform temperature in the house than when a floor exhaust alone was used. No advantage was evident when a higher recirculating rate was used, and although recirculating rates as high as 6.25 and 6.80 cfm per heavy hen seemed to produce no harmful effects on the birds, the poultrymen in charge of the birds was of the opinion that too much air was being recirculated. From the standpoint of fan cost and power consumption such high rates are also undesirable.

3-The recirculation of 2.56 cfm per heavy hen in combination with the exhausting of 2.07 cfm per heavy hen maintained satisfactory litter conditions in the house in which the winter tests were carried out.

4-The recirculation of 2.31 cfm per heavy hen in combination with the exhausting of 2.54 cfm per hen satisfactorily removed the moisture produced by the birds in house A (insulated house) during the period analyzed in March, but not the period analyzed in December.

5-An exhaust rate of three to four cfm per heavy hen was necessary for the most satisfactory removal of moisture and ammonia fumes produced by the birds under winter ventilating conditions in an insulated house such as A.

6-All the tests carried out indicate that there was a lower temperature and relative humidity difference between the floor and ceiling when the recirculating system was used than when it was not. A more uniform temperature within the house, which means a higher temperature at the surface of the litter is believed to aid in moisture removal.

7-The effect of recirculation in providing higher temperatures at floor or roost level and other desirable conditions merits further study.

Bates considered the total amount of water to be exhausted from the house (birds and litter) as equal to 45.4 pounds per 100 heavy hens (six-pound hens) which according to Table 30 seems reasonable.

Kable (55) states: "A. V. Clyde found that when the house air is in motion the condensation point on the walls is from 1 to 4°F below the dew point in still air." It seems therefore that the use of recirculating fans may help in the prevention of wall-surface condensation.

Hinton and McCalmont (42) state that Funness of the Massachusetts State College had good results in an uninsulated 100-hen pen by recirculating 300 cfm and drawing in 100 cfm. They say that: "the recirculation

within the house helps to dry the litter and seems to help in preventing frost accumulation on the ceiling. The temperature has been kept higher than would have been possible with natural ventilation."

Some research would be required in houses exactly similar in other respects, to determine whether the advantage of recirculation is really worth its cost.

h. Special study of some of the factors favorably affecting litter conditions by some other way than improving the rate of water evaporation

In addition to the factors previously mentioned, there are a few factors affecting the litters' condition by some other way than improving the rate of water evaporation from them. These factors are:

- 1-Effect of frequent cleaning of the dropping boards
- 2-Effect of the addition of new litter
- 3-Effect of the addition of lime

a. Effect of frequent cleaning of the dropping boards. One way of improving the moisture conditions in poultry houses, and of leaving more available heat for litter drying, consists of cleaning frequently the dropping boards. Such a practice leaves more available heat for drying the liquid water voided in the pen litter.

Discussing this point, Hill (43) states: "Accepting Jull's (1949) figures for moisture output, there are 46 pounds of water produced daily of which 12 pounds are respiratory and 34 pounds are excretory in origin (for the one hundred 4-pound hen flock). If droppings are cleaned daily, it may be assumed that only 20 of the 34 pounds of excretory water remain to be removed by heat and ventilation."

It should be noted, however, that this practice is quite burdensome and requires much labor. This probably explains the actual tendency toward the use of deep litter in combination with dropping pits as a substitute for dropping boards.

b. Effect of the addition of new litter. Charles, Topper et al (19) found that when new litter additions were made to their experimental pens, a surprisingly rapid rate of moisture increase occurred. Shavings required two weeks to achieve a stable and normal rate of gain. During this two weeks they gained from 23 to 32% of moisture, averaging 26%. Peat moss required but one week to gain 26%. After this initial period of moisture gain, the new litter settled down to a more uniform rate of increase, corresponding to that held previously by similar types of litter. The conclusion of their tests is that additions of fresh relatively dry litter to poultry pens during the winter months does not result in continued dry litter conditions; the newly added litter picks up moisture rapidly and in a few weeks reaches the moisture level of the old litter.

The same authors (19) also add that excessive dampness about the waterers can hardly be controlled by litter additions, since these additions merely served to absorb the spillage and a wet soggy mass resulted about the waterers.

c. Effect of the addition of lime. Bates (12) ran a test to determine the effect of adding lime to wet and packed litter, since it is believed by many poultrymen that the addition of ground limestone causes the litter to dry rapidly. The test consisted of stirring into the litter one pound of ground limestone for each heavy hen in the house (or one pound for each 3.5 square feet of floor).

He concluded from his test: "The addition of limestone does not in itself cause the litter to dry. It was observed that after lime was added, the birds worked the litter more and there was less packing of the litter. After the litter reaches a moisture content of 36%, it appears that stirring without the addition of lime is of little value."

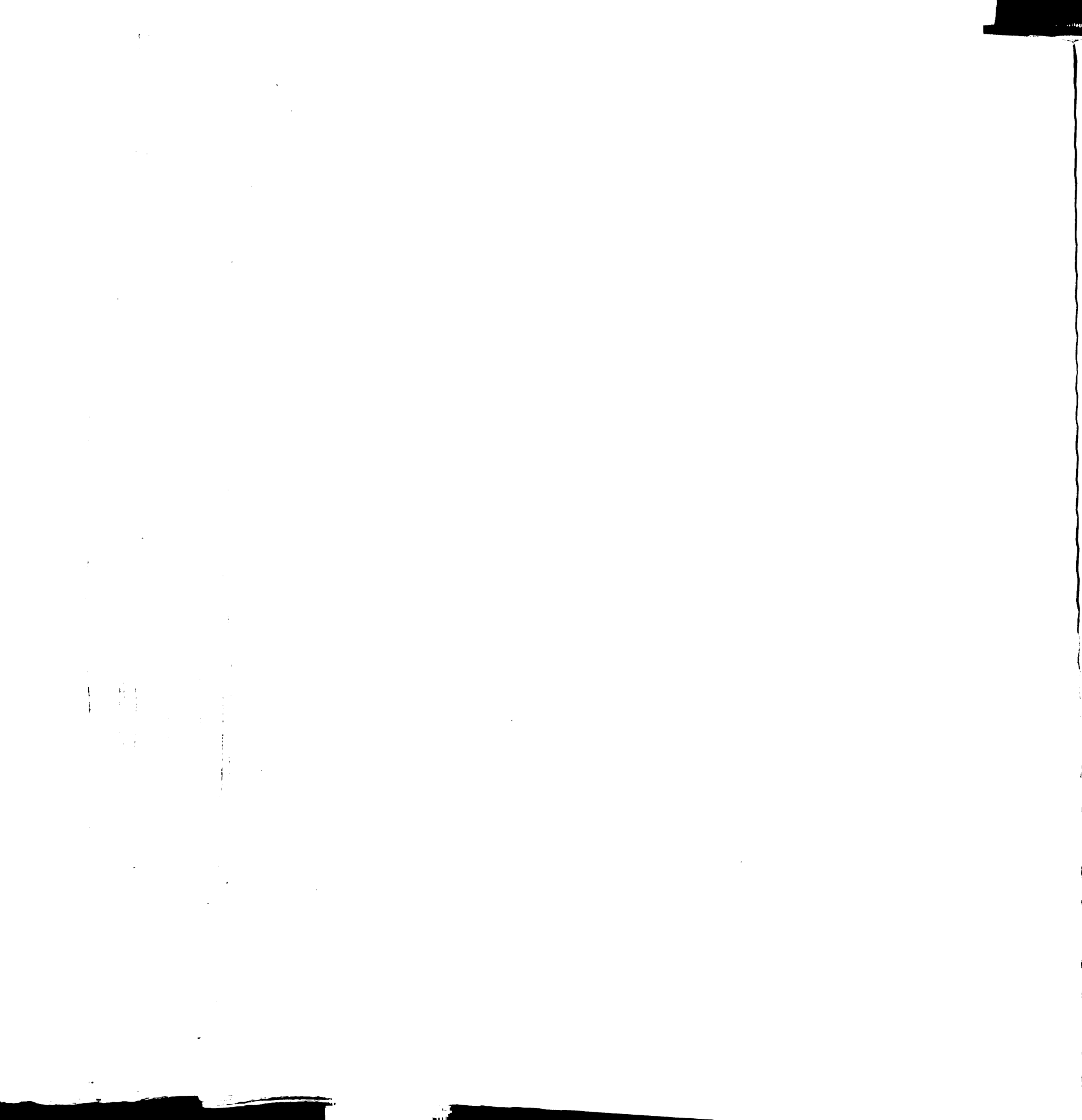
He also said in another section: "The addition of ground limestone to litter having a moisture content below 30 to 40%, will condition the litter so that the birds will work it over and allow the ventilating air to come in contact with a greater surface area of the litter."

C.H. Turner (104) stated as a result of his test on a rather cold poultry house located over an open barn* and having a fan capacity of approximately 2.5 cfm per bird: "Once the litter became damp (above 40% moisture) it was impossible to dry it out by means of the ventilation system even with an increased rate of air delivery. However, by the addition of new dry litter and hydrated spray lime, the old damp litter was reconditioned and remained in good condition throughout the remainder of the season."

Paicment (81) recommends a rate of application similar to Bate's: "In certain cases the application of one pound of hydrated lime per four square feet of floor area is sufficient. However, under average housing conditions where there is some tendency to dampness, the application may be increased to one pound per two and a half square feet of floor area. To be effective, hydrated lime must be applied on a deep litter and stirred frequently." According to this author: "The function of lime is to reduce fermentation in the litter and to perforate the walls of the absorbing material by adhering to it, which favors evaporation."

Hoyer and Blickle (68) state: "The use of hydrated lime at the rate of one pound for each ten to twelve square feet of floor space at intervals of ten to twelve days during the winter also aids in maintaining dry deep litter. When lime is used it is essential to keep the litter stirred to prevent packing or caking on the surface."

* (Walls consisting of asphalt strip shingles laid on 7/8" rough lumber over the rafters; floors made of single thickness unseasoned matched lumber; built-up litter not yet deep enough)



The writer hopes that the above review of literature will help in giving a better understanding of the wet litter problem and will be useful in the planning of further research on this important problem.

P. Design Of The Intake System

1. Importance of the intake system

As the ventilating fan exhausts foul air from a poultry house, fresh air must enter the structure to take its place. This air may enter through cracks in the walls or ceiling, around windows or doors, or may come through specially provided fresh air inlets.

The control of the incoming air is probably as important, if not more, than that of the outgoing air. It is the proper placing of adequate intakes that distributes the air in the house and stimulates the convection currents to make possible satisfactory conditioning of the inside air. Turner (105) illustrates the importance of the air intake system by saying: "Too many ventilation systems have been sold on the basis of what the exhaust fan will accomplish. One of our major problems will be solved by the action of the incoming fresh, dry, cold air rather than by anything that the outgoing air can do. While it is true and appreciated that the fan is essential and provides the moving force, it must be remembered that the outgoing air has done its job by the time it reaches the exhaust point in the stable and the incoming air must do the work which we are asking the system to accomplish."

If the number and size of intakes are insufficient, the velocity of the incoming air may be such that objectionable drafts can possibly result. Barre and Sarnet (11, page 243) state that one basis for estimating the total cross sectional area of intakes is to limit the average air speed through the intake openings to 800 feet per minute.

Where no special intakes are used or where these intakes are not evenly distributed and of proper size, the result is usually an accumulation of strong odor and high moisture in some part of the house (as is present in unventilated houses) with the presence of cold air drafts in other parts of the same house.

Walton and Sprague (106) were early investigators to study the inlet problem on an engineering basis. They say: "Agricultural engineers have been unable to meet the inlet problem on an engineering basis. They have found it necessary to work with experience as their only guide. Results have not been satisfactory." Because of the importance of the inlet problem, and in an effort to provide some information basic to the design of inlet systems, they started, late in 1949, a study to determine inlet air-flow characteristics. Results of their study will be presented here together with the opinions and findings of many other engineers.

2. Principles of fan ventilation - pressure difference conditions causing flow through inlets in actual installations

The principle of fan ventilation is given by Baker (7) as follows: "The fan exhausts air from the stable thus lowering the pressure slightly within the stable which in turn allows outdoor air to enter through any and all openings in the sidewalls."

As to the relative effectiveness of the different air intakes, Saia (89) makes the following statement: "It is justifiable to assume that the effectiveness of an exhausting fan decreases with distance. Because of the resistance to flow, air at the more distant locations from the fan has less movement than that nearer the fan. Thus intakes nearest the fan will supply more air to the stable than those of equal resistance to flow that are located at a greater distance. This might mean that increasing the size of the intakes located further away might be desirable for a more uniform air distribution.

Walton and Sprague (106) checked the pressure differences between inside and outside of several ventilated buildings. These checks were made with a water monometer capable of reading to 0.002 inch. Air flow through these buildings was presumably 90 to 125 cfm per animal unit. The results of this study are summarized here:

a. Pressure difference on very still day. For this series of tests, care was taken to take readings on very still days so that wind pressures would not affect results. The following table (Table 53) shows their findings:

Table 53. Pressure differences through inlets on very still days *

Barn	Remarks	Pressure difference (Inches of water)
1	No inlets	0.025
2	Commercial inlets	0.027
3	" "	0.023
4	" "	0.020
5	No inlets	0.018
6	Homemade inlets; small fan	0.008
7	No inlets; High fan volume; Very tight structure	0.132

* From data by Walton and Sprague (106)

Examination of the table shows pressure differences varying from 0.008 to 0.027 inches of water. The only exception is barn 7. However, barns this tight are rather uncommon and it is included here only as evidence that such well built barns do exist. They concluded that: "typical pressure differences would appear to be in the range of 0.01 to 0.03 inches of water."

b. Pressure difference as affected by wind. It is also important to know how inlets would respond to higher pressures such as those resulting from winds, since when wind blows against the side of a building its velocity pressure becomes static pressure which is effective in causing air flow through inlets.

According to Walton and Sprague (196): "A 25 miles per hour wind seems to be a logical choice for top velocity to consider. This wind velocity has a static pressure equivalent of 0.361 inch of water."

Fairbanks and Goodman (33) also state: "Motors should be powerful enough to maintain practically their full speed when the fan is discharging against a pressure equal to $1/4$ of an inch of water or as commonly stated, against 0.25 inch static pressure. This is about the pressure developed by a wind of 23 miles per hour."

3. Special study of the different types of intakes

One of the most controversial points with regard to present-day, forced-air-ventilation installations is the amount and nature of intake area to provide. A study of the different types of intakes will be presented here.

a. Standard box-type intakes. The same types of standard box-type intakes (L-type and T-type intakes) used in dairy barns are generally used in poultry houses. The recommendations for their application to poultry houses will be given here.

aa. Size of intakes. It has been found that the best size of intake is about 60 square inches (55) (76) (88). This would mean an inlet 6 by 10 inches, 4 by 15 inches or 3 by 20 inches. However, according to N. Mitchell (76), it may be advantageous to use intakes of less than 60 square inches, in the case of small poultry houses, in order to get better distribution of the incoming air.

bb. Number of intakes. The number of air intakes required depends on the total air intake area required and the size of each intake. The standard method of computing the total size of air intakes is to choose a certain desired air velocity through the intakes and vary the air intake area required according to the capacity of the fan used (cfm). N. Mitchell (76) considers a velocity of 700 feet per minute (or 8 mph) as being the maximum

velocity which may be safely used, while Barre and Garnet (11) mention 800 feet per minute. It should be noted, however, that intermediate values from 300 to 500 feet per minute should probably be used to minimize still more the danger of draft. Increasing the total air intake area also reduces resistance to air flow and results in better performance of the fan.

The total intake area required is given by the following formula:

$$\text{Total air intake area (sq.in)} = \frac{\text{Fan capacity (cfm)} \times 144}{\text{Air velocity desired (fpm)}}$$

Ryan and File (82) recommend one 60 square inch intake for each 125 square feet of floor space or its equivalent. Based on a maximum ventilation rate of 1 cfm per hen as they recommend, and on a bird population of 35 hens per 125 square feet of floor (approximately 35 square feet per hen), the corresponding velocity is:

$$\text{fpm} = \frac{\text{cfm/hen} \times 144}{\text{sq.in/hen}} = \frac{1 \times 144}{60/70} = \frac{144 \times 35}{60} = 336 \text{ fpm}$$

Fairbanks and Goodman (31) and H. Mitchell (76) recommend one 60 square inch intake for each 250 square feet of floor space. Based on their recommendation of 2/3 cfm of fan capacity per square foot of floor, the corresponding velocity is:

$$\text{fpm} = \frac{\text{cfm/sq.ft} \times 144}{\text{sq.in/hen}} = \frac{2/3 \times 144}{60/250} = \frac{2 \times 250 \times 144}{3 \times 60} = 1,400 \text{ fpm}$$

Moyer and Blickle (66) recommend 300 to 350 square inches of air intakes for each 1000 cfm capacity of the fan. This corresponds to a velocity of from 410 to 480 feet per minute.

In the use of the above formulas, no consideration was given to the amount of air coming in through the cracks. It is assumed (a) that the cracks simply help in reducing still more the velocity of the air flowing into the house, or else (b) that the air coming through the cracks on one side of the building is going out through the cracks in the opposite side without affecting the flow of air from the intakes to the fan.

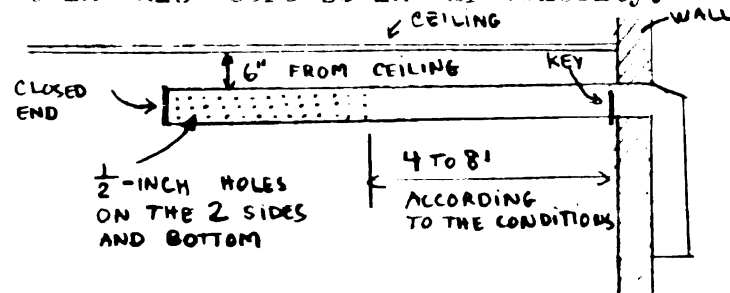
The writer believes that the main effects of increasing intake area are decreasing the velocity and providing a more uniform distribution of the intake air. For this reason there should be no fear about using air velocity as low as 300 to 400 feet per minute or even lower in the computation of the total air intake area required. Some research could be done to see if increasing the number of air intakes may have any effect on the temperature control characteristics of the house and the chances of having some intakes act as outtakes.

cc. location of intakes. In general, the intakes should be eight to ten feet from each other on the wall opposite the fan and on the end walls. Where a larger number of intakes are involved, it may be desirable to locate some of them on the same wall as the fan or fans, to secure complete coverage of the pen area. In no case, however, should an intake be within eight feet of the fan, since "short-circuiting" of the air flow will result in a loss of the ventilating effect in the rest of the pen.

H. Mitchell (76) recommends the placing of one intake in each corner of the house, the remainder being evenly spaced around the walls. The reason for such an even distribution is to provide good circulation of air at all points and leave no dead air pockets in the corners of the house.

According to Ryan and Pile (88) and to H. Mitchell (76), intakes should never be located under or over poultry roosts or manure pits. In poultry houses with the perches (or manure pit) located along one side, these authors recommend placing the fan and exhaust duct about the center of that side and distributing the intakes along the ends and opposite side. Such an arrangement is to prevent the cold air from the intakes from spilling on the birds at night when they are on the perches.

It is stated by Fairbanks (30) and generally admitted that: "Intakes should be so constructed that the velocity of the incoming air is reduced, as it enters the house, to such a degree that no objectionable drafts can possibly result." Chartier (20) developed the following type of diffused air intake to obtain this decrease in air velocity:



This type of air intake seems to be very effective, and is especially suited where air intakes have to be built in the side of the house directly above the roosts. Air is divided into so many streams and directions that the danger of cold air from the intakes spilling on the birds at night is materially reduced.

dd. Construction details. The outside entrance of intakes should be above the snow line and, where possible, should be five feet below the inside openings, since the greater the distance between inside and outside openings, the less chance that warm air which is trapped at the ceiling will back-draft through the intakes (88). If this arrangement places the inside openings closer than twelve inches from the ceiling, the five-foot interval will have to be reduced. The outer opening should be screened with hardware cloth no smaller than 1/2 inch mesh, but small enough to keep out birds and rodents.

The incoming air should be directed upward (88). Very often a baffle or flat surface is used at the ceiling (31). When the air stream hits the surface, it loses its velocity and is broken up and dispersed in all directions. Part of the air mixes with the warm air at the ceiling, while part of it, being cold and heavy, moves down along the wall to the floor. According to N. Mitchell (76), observations have led to the theory that this blanket of cold air next to the wall has some advantage. It would

tend to prevent the warm moist air of the interior of the barn from coming in contact with the walls, windows and doors, thus affording less chance of condensation and decreasing the heat loss through the walls. This theory is based on the fact that in several instances a higher temperature was maintained and less condensation occurred than should have according to calculations for the wall construction and outside weather conditions.

Discussing this theory, Stapleton (96) states: "It seems obvious that some reasonable evaluation of the effect of a curtain of cold inlet air against the building wall is about due. Sufficiently turbulent or at some velocity it

1-Causes mixing to occur where condensation cannot be produced on a surface

2-Changes the condition of wall temperature difference

3-Produces an effect on the film coefficient of the interior wall surface.

These are to the benefit of the problem, but are still somewhat of a mathematical inponderable. Vermont uses this effect induced by infiltration.

Cornell has gone to some trouble to provide a slot from the hay mow plenum to produce this effect. In our poultry house work we assist it with a short ceiling curtain. It seems to be helpful, but it is definitely unevaluated in its effect on heat conservation and control of condensation. In reporting test work it is mentioned sometimes, but in any event it insidiously affects the results, with further divergence of empirical recommendations."

It is generally recommended to insulate all intakes to prevent sweating (33) (33). Fairbank's and Goodman (33) state: "Because cold outdoor air passes through the inlets and they are surrounded by warm air, moisture is sure to condense on the stable side of flues made of one thickness of 7/8 inch boards. To assure dry flues, it is recommended that they be made

of two thickness of $7/8$ inch lumber with good paper between or of a thickness of $7/15$ insulating board with a covering of wood. Durable flashing is needed over the opening in the throat of the flue."

The question sometimes arises as to whether or not it is necessary to have some hinged or sliding doors on the top of air intakes in order to increase the air resistance and decrease the air flow at some times. Discussing this question Fairbanks and Goodman (33) state: "Seldom is it necessary to have doors or other throttling devices on air intakes. Intakes without throttling doors have been used for years in central New York. If, however, throttling doors are installed, they should be cut a full inch narrower than the opening they are to control so that the air passage will never be completely closed." Speaking more specifically for poultry houses, they say in another publication (31) that such doors are rarely needed except on inlets placed under dropping boards on the side of the house exposed to prevailing winds.

Another way of decreasing the air flow is by the use of a flap or wind damper. According to Kalbfleisch and White (57): "If desired, a wind flap or wind damper may be placed in vents on the windward side of the barn. A flap of this type consists of a thin piece of aluminum or galvanized iron that is hinged on a piece of $1/8$ inch wire or rod. For a standard 4- by 16-inch vent this sheet measures about six inches wide and sixteen inches long. The hinge wire is located just below the board that forms the top of the vent. When there is no wind the flap hangs in a vertical position and the air enters the space between the bottom edge of the flap and the barn wall. When a gust of wind develops, it swings the bottom of the flap toward the barn thus closing the vent. This flap has proved effective during windy weather."

It should be remembered that if the system is to operate satisfactorily, all openings such as windows, doors and chutes should be kept closed, all cracks should be plugged, and the house made as tight as possible (88) (92).

ee. Performance of standard box-type intakes. Walton and Sprague (106) made extended laboratory investigation of the air flow characteristics of several commercial inlets having built-in constant flow features and some homemade inlets of the L-type and T-type (overwall). The summary of their findings is presented here.

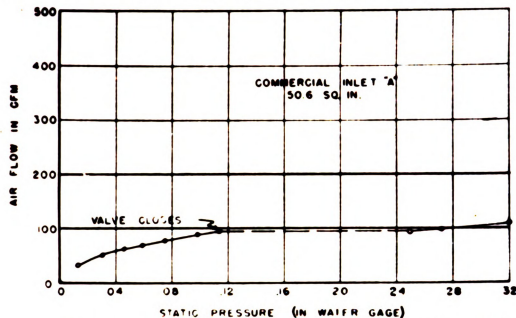
(1) Commercial inlets with built-in constant flow features. Typical commercial units with a built-in valve for regulating the effect of wind were found to close off at a static pressure of approximately 0.114 inch of water, which is the equivalent of a 15.3 miles per hour wind. The authors (106) also noticed that static pressures above the shut-off point did not increase the flow of air through the inlet any appreciable amount. This means that the inlet successfully minimized the effect of winds. One other significant finding was that commercial inlets admit rather small quantities of air.

Figure 14 summarizes flow results for the commercial inlets tested. Results are compared for a static pressure of 0.02 inch of water, since this static pressure represents very closely the pressure difference across inlets in actual installation.

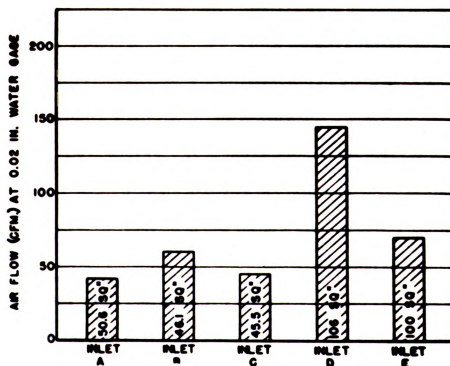
(2) Homemade I-type inlets. Figure 15 gives air flow characteristic for a typical 6- x 10-inch I-type inlet. Results of tests with the three different cross-sectional dimensions being recommended shows that the air flow rates are not materially different.

The effect of covering the inlet entrance with hardware cloth is also shown, as is the result of closing the inlets varying amount on the stable

AIR FLOW CHARACTERISTICS FOR TYPICAL COMMERCIAL INLETS



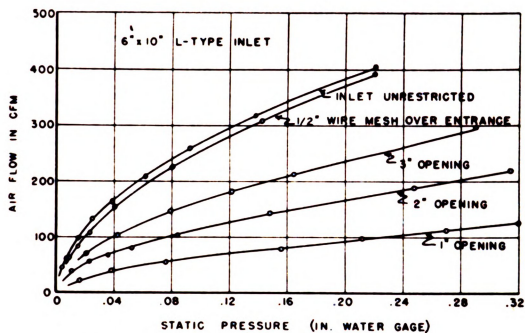
Flow characteristics for a typical commercial inlet. The broken portion of the curve indicates a pressure range which could not be investigated. The pressure increase resulted from closure of the inlet-flow regulating valve



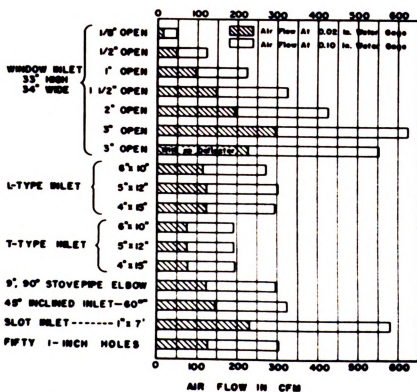
Air-flow results for commercial inlets tested. The larger rate of air flow through inlet D is a result of its larger cross section and its regulating valve design

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AIR FLOW CHARACTERISTICS FOR TYPICAL HOMEMADE INLETS*



Air-flow characteristics for a typical 6 x 10-in. L-type inlet



Air-flow rates at 0.02 and 0.10 in. water gage for homemade inlets

side leaving openings ten inches in length. The restriction offered by the hardware cloth over the inlet entrance is not great.

(3) Homemade T-type inlets (Over wall). The result with different cross-sectional dimensions of T-type inlets also strengthened the results obtained by L-type inlets; the air flow rates were not materially different.

b. Tip-in windows as intakes. Window inlets have not been looked upon with favor by many engineers, but many people working with animal shelter ventilation would welcome their simplicity. This led us to a review of the opinions and findings concerning them.

Kable (55) states that tip-in windows are not recommended as intakes because the sweep of air into the house tends to channel and create drafts. According to him windows can be used as intakes when sawed off and baffled. They may be used as tipping windows in the summer but are not tipped-in for winter ventilation.

Ryan and Pile (88) have somewhat similar attitude toward tip-in windows. They recognize that window ventilation is desirable in moderate weather but they find it unreliable during cold windy weather: the windows might act as outtakes rather than intakes, or drafts may be caused by not properly regulating the windows. Where tip-in windows with side shields are used they recommend the placing of a vertical board at the top of the window to direct the incoming air straight up to the ceiling. The upper part of the board should not come closer than twelve inches to the ceiling. Considering the usual width of animal shelter windows, they say that a tip-in distance of about two inches is generally adequate.

Fairbanks (30) states: "We tried as in the former tests, to supply intake air through the windows and found again that it was unsatisfactory.

The effect of the window intakes are: unequal distribution of air, drafts, and the unsatisfactory results of the human element in judging the adjustment of the window opening."

If there is about 2" inches between the top of the sash and the stable ceiling, Fairbanks and Goodman (33) say that windows, although in general undesirable as intakes, can be used for this purpose under certain limitations. "Side shields should be used. A vertical board about eight inches wide should direct the air straight up to the ceiling. The window is tipped enough to give an opening of 60 square inches."

Walton and Sprague (106) concluded from their laboratory investigation: "Based upon the performance of the frequently recommended L-type inlet, windows 34 inches wide and opened more than one inch at the top offer too little restriction. Without a deflector a window may easily misdirect air. A deflector directing air upward reduces the window inlet capacity but may not eliminate drafts. A 34-inch window opened one inch and equipped with deflector should be equivalent to a 6- x 10-inch L-type inlet in performance."

c. Double-sash windows used as intakes. According to Moyer and Blickle (68) "The lower sash of double hung windows can be raised to provide an air intake. However, a baffle board should be placed across the opening to prevent the air from sweeping across the floor." He adds: "Window intakes require only half the opening of fixed intakes."

Fairbanks and Goodman (31) also state: "Air inlets may be made under windows. The sash is raised from one and a half to three inches depending on the width of the window. The area under the sash should be 60 square inches. A baffle board the top of which is about four inches higher than the bottom of the raised sash is nailed across the window frame, so that incoming air can never blow on the birds but will be directed up along

the inside face of the glass. This arrangement has the marked advantage of reducing the accumulation of moisture on the windows." They also add: "It should be remembered that window inlets cannot be made at windows on which a storm sash is used."

These authors (31) seem to favor the use of double sash windows as intakes in preference to standard built-in inlets, when possible: "If the windows cannot be raised conveniently to form window inlets, built-in inlets may be installed. These are just as satisfactory in operation though they are slightly more expensive to construct."

d. Stove pipe elbow and 45° inclined inlet as intakes. Walton and Sprague (106) concluded from their laboratory investigation that a nine-inch stove pipe elbow and a 45° inlet which are two types of inlets sometimes found in poultry houses, admit slightly greater volumes of air than are admitted by a 6- x 10-inch standard L-type inlet.

e. Long slots or series of holes as intakes. Long slots or series of holes placed adjacent to the walls are sometimes used as inlets to admit fresh air from storages spaces or closed in areas above animal shelters.

Saia (92) discussing the slot system of intakes states: "The slot intake functions to prevent condensation on the sidewalls, introduce fresh air evenly in the stable, stimulate convection currents causing more uniform stable temperatures, and allows the fans to operate at a minimum pressure differential. The row acts as a plenum chamber allowing air, which is up to ten degrees warmer than outside air, to enter the stable uniformly regardless of wind direction." He adds: "Before installing the slot intake condensation was almost always present. After installation, condensation was not once observed in areas under the slot."

Turner (105) makes a similar statement: "This intake system eliminates the wet spots on the walls and makes available tempered air from

the hay loft rather than the colder outside air. It is less expensive and easier to install. The fan operates continuously, and creates a partial vacuum that draws air down through the slots or holes into the house. This cool air flowing down along the outside walls keeps the warmed stable air from coming in contact with the cold outside wall. This prevents condensation and the formation of moisture on the walls." He adds further: "It should be emphasized that for the slot or bored-hole type of inlet at least one exhausting fan must operate at all times." This is to prevent flow of the moisture-laden air to the loft.

The same author (105) gives the following information regarding the construction details:

(1) Slot-type intake. Leave a one-inch slot in the ceiling at the junction of the ceiling and the sidewall. Place a four-inch board on edge by the slot to prevent hay and chaff from falling into the opening. Construct the slot all around the house, and within two feet of the fan.

(2) Bore-hole intakes. (The bore-hole intake is a modification of the slot-type of intake). Bore holes in the ceiling close to the wall all around the edge of the house except within two feet of the fan location. Twenty-five one-inch holes or fifteen 1 1/4-inch holes are needed for each 100 cfm of fan delivery. Place a four-inch board on edge, by the holes to keep the chaff away.

4-Natural cracks vs special air intakes

Some recent recommendations call for no inlets. One of the claims is that natural cracks and crevices provide sufficient draft free infiltration. According to Walton and Sprague (106) a second reason is due to the fact that field observation of inlet performance has added to the conclusion. "In a single installation it is probably that some inlets will be discharging, some inlets will be acting as inlets and some inlets may be

stagnant. In the next moment it is probable that their action may be entirely changed. It becomes evident that there is no real agreement as to what inlet requirements are."

Among those people recommending no inlets, Burress and Wakefield (17) are the strongest supporters: "If the fresh air experiences much difficulty entering the house, as might be the case with a tightly constructed barn the fan will have to work harder to induce the air to flow into the stable. This will cut the fan's capacity and will increase the amount of electrical energy required to move the air. On the other hand, the stable construction may be very loose. Then air can enter so easily that, if the wind blows against the loose walls, the stable is apt to be over ventilated. Judging from these facts, the structural tightness of the house is a very important item to consider when a ventilation system is being installed." Their recommendations for animal buildings of different tightness characteristics will be presented here.

(1) Very loosely constructed houses. These houses were probably tighter when built. They become looser with age as window and door cracks become larger and as walls and ceilings become more pervious. According to Burress and Wakefield (17), in such houses: "There is more than adequate area for air to enter without the addition of inlets. Temperature controlled fans usually greatly improve conditions in these very loose houses, but wind, not fans, may cause over-ventilation. Tightening the wall near the fan and closing large cracks in other areas will cause fresh air to enter more uniformly over the house and will help prevent drafty areas."

(2) Moderately loose house. The houses in this category especially include older houses. According to Burress and Wakefield (17): "Older houses tend to be moderately loose and usually have more than adequate openings for air to enter. In these older houses it is frequently nec-

necessary to tighten the wall near the fan to reduce "short-circuiting" of fresh air. Cracks about doorways and windows should be closed and any chute openings should be covered."

(3) Tight houses. Recently built or remodeled houses usually make up the "tight" type of houses. Tight new masonry walls, tighter doors and windows, and relatively good ceilings are common to the group. According to Burress and Wakefield (17): "A considerable portion of the required air will infiltrate through the walls. In the tighter houses of this class, however, slot inlets may be needed to supply one-fourth of the air when air is changed at a particularly high rate."

(4) Very tightly constructed houses. Such houses are rare and usually of very recent construction. Side walls frequently are made of well-mortared block or extremely tight frame construction such as plywood. Ceilings are either plasterboard, plywood or similar material, and doors and windows fit almost like those in dwellings. According to Burress and Wakefield (17): "When they are encountered such houses pose quite an engineering problem. Inlet area must be provided to admit about one-half or more of the fresh air needed."

Field observations indicate that most stable and poultry houses fall into the moderately loose to tight construction group. The general conclusion of Burress and Wakefield (17) is therefore: "... installation of inlets will seldom be required."

Saia (89) concluded from his tests on dairy barns: "Operating with the intake system open and closed, it was found that although the total fan delivery was slightly less with intakes closed, satisfactory temperature and relative humidity conditions were maintained. Cracks and air leaks provide sufficient infiltration to afford adequate ventilation in structures considered relatively tight. Distribution of fresh air cannot



be controlled, and drafts are more prevalent when the intake ducts are closed. Condensation will occur on the walls at a higher outside air temperature with the intake closed, showing the need for supplying intake air to the wall surfaces."

Whatever be the claims in favor of no special air intakes, the writer feels that the theory of admitting air through leaks certainly does not hold with the theory of making the walls of the poultry house tight and warm. Further than that, the leaks, in most cases are of insufficient area and very rarely uniformly distributed. For this reason the writer believes that some special intakes are always required. Poultry houses with so many leaks as to require no special intakes are generally not well enough insulated to profit from a mechanical ventilation system. If insulation is provided first, the house will become more tight and some special air intakes will be required.

By the use of the air change method, the natural infiltration rate through cracks is usually calculated as being about once or twice each hour. This is far from the air flow requirements, especially in relatively mild winter weather, since in such weather the rate of ventilation of 4 cfm per hen or 240 cfm per hen corresponds to an air change of $240 \div (35 \text{ sq.t. of floor area} \times 6' \text{ high}) = 240 \div 21 = \text{more than } 10 \text{ air changes per hour}$. It seems as if the use of cracks alone in combination with fans of 4 cfm capacity per hen might result in air velocities dangerous with respect to drafts.

Ryan and Pile (88) state that: "a good ventilation system provides a definite intake for air to enter the shelter and a definite outtake through which the moisture-laden air may escape. Leaks and cracks should be eliminated to make the system function efficiently." Shier (92) is also of

the same opinion: "The use of building paper in the walls, and the use of storm sash and storm doors on the openings is essential in controlling air leakage. No ventilation system will work satisfactorily as long as crack ventilation is permitted."

Figure 46 represents the findings of Walton and Sprague (196) from their laboratory investigation, and shows the characteristics of air infiltration through cracks in a frame wall. The wall panel tested was four feet wide and eight feet long and contained five cracks eight feet long. They concluded: "It can be seen that cracks admit a considerable quantity of air. One hundred feet of tight cracks (cracks resulting from nailing rough pine boards firmly against each other) will admit as much air as a standard 6- x 10-inch L-type inlet."

They also add in summarizing their results: "It is evident through all of these tests that none of the common inlets (any kind of special inlet excluding natural cracks) admits a very large quantity of air. Based upon present recommendations for size and number of inlets required for animal shelters, a very large percentage of the fresh air entering an animal shelter must of necessity come through cracks."

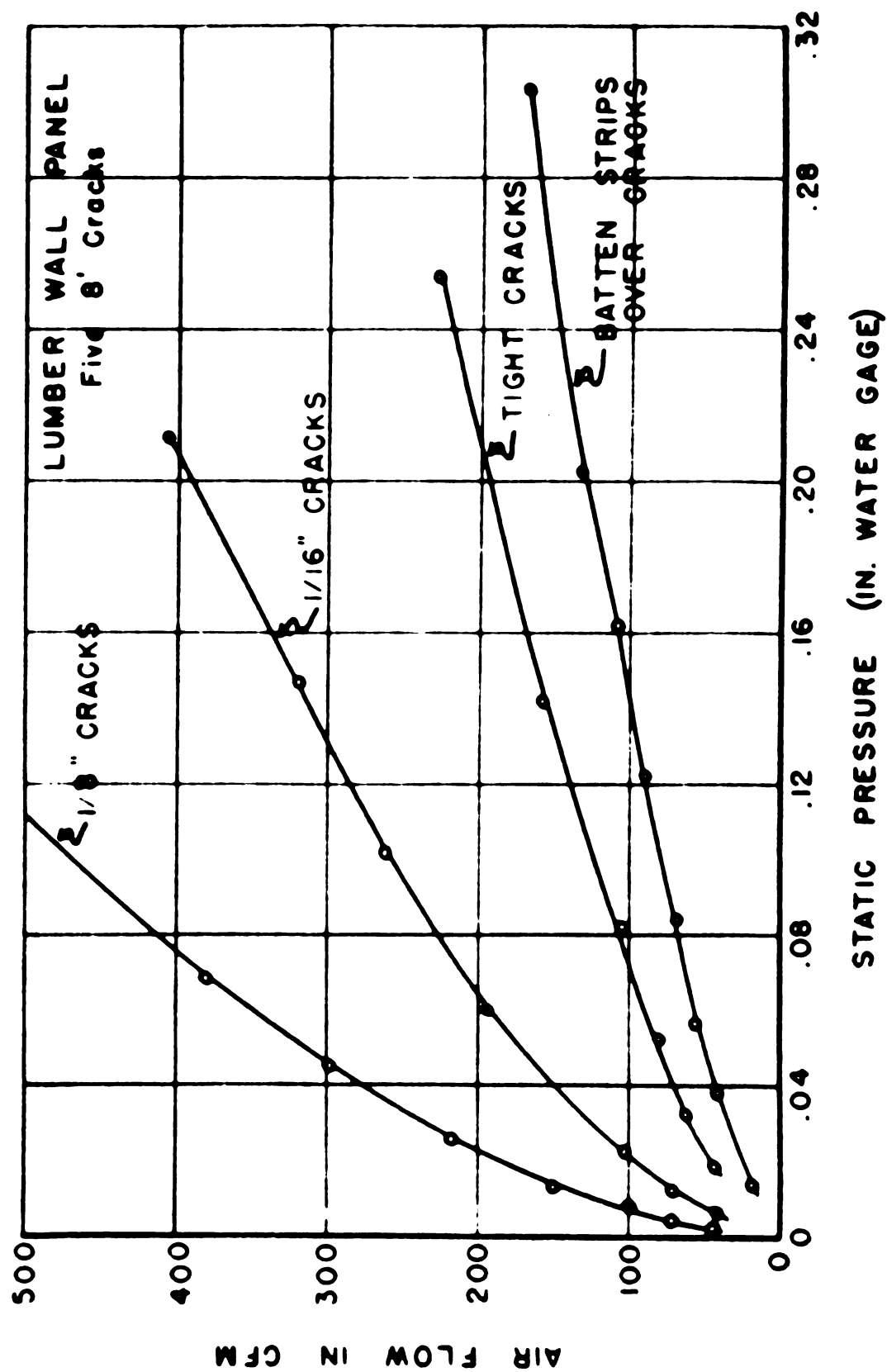


FIGURE 46. Air flow through cracks in a rough lumber wall panel
(Walton and Sprague) (106)

10-12-5

. Design of the Outtake System

1. Two debated theories: Continuous vs intermittent ventilation

It has been and is still a most debated question whether during winter, ventilating fans should be operated continuously at a restricted rate, or under thermostatic control to cut off the fan operation completely, at a predetermined low temperature.

Barre and Sarnet (11) state that: "it is essential that moisture produced by the birds be removed continuously. Otherwise wet litter will contribute to serious losses from disease and, in cold weather, severe damage to the structure will result because of condensation on the building surfaces." If the outtake system consists of several fans, Barre and Sarnet's (11, p.243) recommendation is that: "at least one fan should operate continuously."

Shier (92) states: "A size of fan should be selected so that it can run continuously rather than switching on and off."

According to Strahan (100) it is safe to say that: "unless enough insulation is used to cause the value of AC to be less than 1.0, it would be better to revert to the uncontrolled system of management and permit the house temperatures to fluctuate just about parallel to outside temperatures most of the season."

Cristel (25) in the province of Quebec (Canada) considers intermittent ventilation, with its resulting higher inside temperatures, preferable to continuous ventilation with respect to moisture removal and the maintenance of dry litter. The writer has visited many poultry houses using Cristel's recommendation (one thermostatically operated fan) and found the litter to be remarkably dry. Such results may be partially explained by the fact that infiltration is probably the cheapest and the simplest way to obtain continuous ventilation when the fan is not running. One other reason for

the dry conditions obtained in houses with intermittent ventilation lies in the fact that in well insulated houses, moisture and temperature increase about at the same rate when the fan is off.

From his own tests and observations (to be described later), it is the writer's opinion that in a well insulated house intermittent ventilation is satisfactory. While during the cold parts of the day the fan will be turned to low speed or off and moisture will accumulate, when the sun shines or during the warmer part of the day the fan will work at full capacity and more moisture will be removed than accumulated. Thus over a 24-hour period or a period of several days, equilibrium will be reached, provided sufficient air flow is maintained while the temperature is warm enough to remove the excess moisture.

2. The use of automatic controls

a. Manual vs automatic control. Another question which has been debated in the past is whether ventilating fans should be operated manually or under automatic control. Today the trend is toward complete automatic control.

Kable (55) had some doubt about the use of automatic controls: "Automatic control aids uniformity of temperature. Solenoid control of adjustable louvers or a damper in the outtake flue would be theoretically desirable. Practically, however, the additional cost and the added mechanism to get out of order might affect the advantage. Simplicity and ruggedness in farm equipment usually give better results than complicated equipment."

Otis and White (80) reported that: "although excellent results were obtained under conditions of manual ventilation control, it might not be possible to obtain such satisfactory results with this type of control where less attention is given to the care of the flock. Thermostatic control doubtless has possibilities for saving labor when keeping uniform temperatures."

Tribble (100) states that a manually operated fan system of ventilation is not satisfactory for the following reasons:

- 1-It requires personal attention by a semi-skilled operator
- 2-There must be some reliable source of weather information on which to base decisions
- 3-It will be necessary to buy about the same amount of equipment as for automatic control
- 4-Probably the most important reason is that automatic controls gives more satisfactory results.

He also adds: "Automatic controls were developed, primarily, to compensate for sudden changes in temperature. An average farmer will not arise at two o'clock in the morning and make the necessary adjustments to his ventilation system. This chore is unnecessary with automatic controls."

b. Thermostat vs humidistat control. Tribble (102) concluded from his experiment on controls for dairy barns fan ventilation that: "if the volume of air required to remove the moisture is greater than the volume of air required to remove the heat, the thermostat should be used to control ventilation. On the other hand, if the volume of air required to remove the heat is greater than that required to remove the moisture, the humidistat may satisfy the condition more readily." He also adds that a humidistat does not readily control temperature and a great temperature fluctuation results.

H. Mitchell (76) state: "Thermostats are more reliable than humidistats. Thermostats are more rugged, and if rust-resistant ones with enclosed contact points are used they work quite satisfactorily. On the other hand, a humidistat is rather delicate, especially the element. When used in dairy barn or poultry house, they work fairly well for a time; however, before long the moist dust in the barn collects on the element causing it to become insensitive to moisture changes in the air. The elements are so delicate that it is practically impossible to clean them without damage."

If humidistats may be satisfactory in some dairy barns, especially well insulated, the writer has some doubt however, concerning the success of their use in poultry houses. The main reasons for this opinion are based on observations of the fact that poultry houses are so dusty and the birds so nervous that the element of the humidistats would become covered with dust and defective much sooner than in dairy barns.

c. Type of thermostat suitable for animal shelters. Reverse action thermostats are used to control fans in animal shelters. When these thermostats are set at a temperature such as 40°F, their differential can be adjusted to three degrees so that the fans will start when the house temperature rises to 41.5°F and will stop when the temperature falls to 38.5°F. Thermostats used for starting oil burners on furnaces have a positive action and cannot be used for animal shelters.

d. Thermostat setting. The thermostat setting of single fans is generally based on the insulating value of the poultry house. In well insulated poultry houses, the setting point may be as high as 50°F while in moderately insulated poultry houses, the setting point is usually around 35°F.

Kalbfleisch and White (57) (Canada) state: "In commercial poultry houses where the temperature is controlled, thermostats are usually set between 40 and 50°F. Bell and Moore (14), in Michigan make the following statement for poultry houses: "If the thermostat is set to start the fan at 50°F. - and stop at 48°F. - it will do a good job of controlling both temperature and moisture. In some cases (writer's note = Uninsulated or very poorly insulated houses) it might be better to leave the fan running continuously during the winter months."

When several fans are used, it is sometimes recommended to have a different thermostat setting for the different fans, in order to have a more

flexible ventilation rate in relation to the outside temperature. Some of the recommendations given for thermostat setting in dairy barns with two fans are given here:

Soia (29) makes the following thermostat setting recommendations in dairy barns with two fans: "One fan should be operated by a thermostat turning it on at 51°F and off at 48°F. The other fan should be operated by a thermostat turning it on at 38°F and off at 35°F." The I.L.I. Electric Ventilation Company, Chicago, (48) recommend the following setting when two fans units are used: "They should be installed at opposite ends on the same side of the barn. One thermostat should be set to stop fan at 40°F and start it at 50°F and the other thermostat should be set to shut off at 35°F and snap on at 45°F.

The writer believes that such different thermostat settings could apply in poultry houses, especially when the fans are located side by side. If the fans are spaced along the wall, a uniform thermostat setting is usually used and it is believed to give a more uniform temperature in the house.

2. Thermostat location. The best location for a thermostat is usually as near the center of the house as possible and at about five feet above the floor. It should not be located near the outside walls, windows, doors, chutes, or too near the animals. A mercury thermometer should be permanently available near every thermostat in order to check it.

3. Types of outtake systems

Seven types of systems may be considered:

- a- The single-fan system, one volume delivery
- b- The single-fan system with automatic operated shutter or damper for two volume delivery
- c- The single-fan system, dual-speed or two-speed

d-The single-fan system, belt driven

e-The two-fan or multi-fan system

f-Ceiling fans system

g-Recirculating fans system

a. Single-fan system, one volume delivery. The single-fan system is the simplest type that has given satisfactory results. A duct leading to within 15 inches of the floor is very often used. In such a case, Turner (195) recommends partially closing the exhaust duct with a damper during excessively cold weather. The duct should never be completely closed, however. A thermostat is generally used to cut off the fan operation completely, at a predetermined low temperature, resulting in an intermittent operation. Manual control is not recommended because it is less reliable.

Kalbfleisch and White (57) state: "Motors with a standard speed of 1725 rpm are suggested, as low speed motors in sizes below one-half horsepower are not satisfactory when delivering air against wind pressure."

b. Single fan system with automatic operated shutter or damper for two-volume delivery. The one-fan, two-volume system gives more even temperature control and uses a smaller exhaust duct than the single-fan system. A small duct is used that extends to within 15 inches of the floor. The volume is restricted to from 30 to 40% when drawing air through this duct. A shutter that is controlled by a thermostatically regulated motor behind the fan provide the two-volume delivery. Manual control can also be used although it is less reliable. The fan draws off the moisture-laden warmer air at the ceiling when the shutter is open and draws the restricted or reduced flow of cooler air up the duct from the floor when the temperature of the house drops below the thermostat setting point. This prevents lowering the house temperature below the desired minimum but still moves air through the house. The shutter is activated by a simple furnace damper motor.

The duct size for this system is considerably smaller than that for the other systems. Turner (105) states: "It is from four to six inches deep and as wide as the fan unit. For poorly insulated stables, a four-inch size is recommended." He adds: "A motor and fan that can be throttled is needed for this system because the fan is not working at full capacity when it draws air through the restricted exhaust duct."

Some commercial units using single-speed fans are designed with a thermostat set to break the circuit when there is danger of freezing in the house and, in addition have an automatically operated damper set in the duct behind the fan. This damper when closed, restricts the flow of air approximately one-half of what it is at free delivery.

Some people will perhaps object to this method of controlling the rate of air flow on the basis that it might overload the motor because of the increased load caused by the static pressure and the lack of ventilation provided for the motor by the fan. However, Strahan and Marsh (97) speaking of the Louven fan built on this principle state: "Ordinarily this would be a perfectly valid objection, but in this instance it is not because the characteristics of the fan we are using are such that subjecting the motor to 100% maximum static head will increase the actual consumption of power so small an amount as to be practically negligible. Of course increasing the head rapidly decreases delivery. This however, is a simple matter of design having to do with calibration of the damper."

c. Single-fan system, dual-speed or two-speed. Dual-speed motors or two-speed motors used with fans normally have a standard speed of 1,725 rpm and a low speed of 1,140 rpm. Kalbfleisch and White (57) state: "At full speed they have the same capacity as the equivalent size of standard fans, while at the low speed the air delivery is about 60% of the standard rating."

Multispeed fans, although being a little more complicated and costly to repair in case of failure offer the advantage of providing a more nearly continuous ventilation as compared to a single-speed fan of the same total capacity.

White and Schwantes (107) note that: "Two speed fan operation appeared to offer advantages over constant speed while requiring only a minimum of manual attention."

d. Single-fan system, belt driven. Belt-driven fans deliver air at one, two, three, or four different rates by the use of step pulleys on the motor and fan. To obtain these different air delivery rates it is necessary to manually move the belt to different pulleys for adjusting the air flow to barn requirements. Because of this factor their use is not so popular nowadays and two-speed fans or automatic shutter and damper are usually preferred.

e. The two-fan or multi-fan system. Poultry houses with large flocks will get best results from a two-fan or multi-fan system.

According to Fairbanks and Goodman (33) the fans can be placed side by side at any convenient place in houses less than 110 feet long. According to them, there seems to be some advantage in placing the fans close together in the same duct: "There is always danger that in very cold weather moisture on the bearings of the lower vanes will freeze and hold the vanes open when the controlled fan is stopped. If the two fans are close together in the same duct the air that enters through the "dead" fan will at once be blown out by the fan that is in continuous operation. This means that little air will be exhausted from the house. The fan that is operating will, however, prevent cold air from blowing into the house through the duct. Placing the fans close together, in the same duct, also simplifies wiring and reduces its cost."

Turner (115) also states: "Install both fans of the two-fan system together in the wall, so air movement will always be in the same direction."

In houses much longer than 110 feet, the fans should be uniformly spaced on one wall. In such a case each fan must be controlled by a separate thermostat and hand switch.

f. Ceiling fans system. Ceiling fans have been found useful by certain farmers in barns already having natural gravity flues. In such a case, the electric fan operates as a supplementary unit only. The gravity system is functioning at all times. The motor-driven fan is brought into use only when atmospheric conditions are such that gravity movement of air is retarded or stopped entirely. The advantages claimed by the manufacturers of Jamesway ceiling fans (51) are: "... savings in current and a constant movement of air in the building."

g. Recirculating fans system. In recent years some emphasis has been put on the usefulness of air recirculating fans. The advantages claimed for the use of such fans are:

- 1-A dryer litter
- 2-A more uniform temperature and relative humidity in the house
- 3-A warmer air temperature at the floor and roost level

Since the principle of recirculating fans is based mainly on increasing the rate of evaporation of moisture from the litter, this subject has been discussed in the section dealing with: "The wet litter problem" and will not be repeated here.

4. Capacities of fan or fans

There has been and there still is considerable variation in the recommendations of the size of fan or fans to use to adequately ventilate a poultry house. In fact, differences are so great that there is little wonder that so many farmers and extension men are puzzled and don't know what size to recommend or use.

Fairbanks (32) recommends one cfm per hen. Chior (22) states that the fan size is usually based on removing one cfm of air for each hen. Kable (55) also states the same but adds that as far as he knows, this is an empirical value. The I.L.G. Electric Ventilating Company, Chicago (48) also recommend one cfm per bird.

Ryan and File (20) mention that following the recommended average insulating values for animal shelters, the minimum allowable volume of air change under restricted ventilation for hens is $1/4$ cfm of air (based on relative humidity of 75% and inside temperature of 45°F , when it is -15°F outside, i.e. 60°F temperature difference). They add: "When outside temperature rises to 30°F it will be difficult to maintain a constant inside temperature of 45°F ., because the required air movement would cause drafts. It is possible however, to hold a temperature difference of 25 degrees and permit the inside temperature to rise to 55°F under such mild conditions. To increase ventilation when temperature rises above freezing, doors and windows may be opened." They therefore recommend a maximum fan capacity of one cfm of air per hen.

Reed (84) states that actually the minimum ventilation required to maintain the maximum inside relative humidity needed to prevent condensation within the building seems to have been established by practice as one cfm per four-pound hen. He adds: "This quantity of ventilation removes exhaled vapor at low temperatures, and at normal outside temperatures also removes much moisture vaporized from excretions."

The American Machine and Metals, Incorporated, makers of Fan-Pac (2), are recommending two cfm per hen.

Moyer and Blickle (68) state: "Experimental work on the amount of air required indicates that the movement of 1 to 2 cfm of air for an average mature bird is sufficient with forced ventilation." However, in their recommendations (Table 54), they used $2/3$ cfm per square foot of floor area, based on the findings of Turner (103).

Table 54. Fan capacity and air intake area recommendation for different sizes of poultry houses *

Floor area (sq.ft)	Fan delivery recommendation (cfm)	Approximate fan diameter (inches)	Total air intake area (sq.in)	Number of 60 sq.in. intakes
400	260	9 or 10	120	2
800	530	9 or 12	180	3
1200	800	12	300	5
1800	1200	12	420	7
2700	1800	14	600	11

* From Moyer and Blickle (68)

H. Mitchell (76) mentions that investigations conducted by the poultry husbandry and agricultural engineering departments of Cornell University have indicated that 2 cfm per bird is satisfactory for light breeds such as Leghorns and $2 \frac{2}{3}$ cfm per bird is satisfactory for heavy breeds such as Plymouth Rocks, in houses with walls having a heat resistance of about two. Using the general floor space occupied by these birds, this represents about $2/3$ cfm per square foot of floor space used by birds. Bairbanks and Goodman (31) also recommend $2/3$ cfm per square foot of floor area.

Lippincott and Card (66) state that such information as is available suggests maximum winter ventilation in the poultry house should be at the rate of eight changes of air per hour. In the average poultry house this would mean about $3 \frac{1}{3}$ cfm per hen.

Oliver (78) recommends a maximum of at least 3 cfm per hen with a minimum of 1 cfm per bird where 0°F or below is common and 1.5 where winter weather temperatures seldom go below 10 to 20°F.

Bell and Moore (14) in Michigan, recommend a fan capacity of 3/4 to 1 cubic foot of air for each square foot of floor area or from 2.4 to 3 cfm per hen according to the size of the house (Table 55):

Table 55. Fan capacity and total air intake area recommendation for different size of poultry flocks*

Size of flock	Fan capacity recommendation (cfm)		Number	Total air intake area (sq.in)
	Total	Per hen		
100 birds	300	3	1	240
150 birds	450	3	1	360
200 birds	500	2.5	1	360
300 birds	750	2.5	1	480
500 birds	1200	2.4	1	720
600 birds	1500	2.5	2 (750 cfm each)	960
1000 birds	2400	2.4	3 (800 cfm each)	1440

* From Bell and Moore (14)

Cristel (25), in the province of Quebec (Canada), recommends as much as 4 cfm per hen on the basis that fan ratings are usually given in cfm of free air which is a higher value than the actual performance of fans on windy days, or when ducts are used. The writer has visited many poultry houses using Cristel's recommendation and found the litter to be remarkably dry.

In the above review of literature, the writer did not intend to include all the recommendations which have been published. His main objective is to show the wide variety of opinions existing on this debated

subject. In the sections dealing with the actual performance of fan ventilation in poultry houses, the writer will present his own conclusions on this subject.

5. Mechanical protection of fans against wind, rain and snow

To safeguard the fans from weather (rain, snow) and to prevent back-drafting (wind), it is necessary that fan openings in the wall be protected by a hood. Such a hood is especially necessary where the fan is installed on the windward side of the building in areas having strong prevailing winds. The radius of the hood can be chosen generally as two inches more than the diameter of the fan (91).

Exhaust louvers (also called shutters) are also provided on all intermittent operation fans. These louvers will close automatically against incoming air pressure but will open readily when the fan goes in action. It is usually preferable to install them inside of the house, since those installed on the outside sometimes freeze open or shut so the air flow cannot be regulated.

Kalbfleisch and White (57) state: "Louvers normally consist of a number of small metal flaps that are connected by a rod so that all the flaps operate together. This connecting rod should be removed to allow each flap to operate independently because the lower flap is sometimes frozen closed." They add: "In regions that have relatively mild winter weather louvers may be placed on the outside of a barn. In colder regions (Zones C or D), the louvers should be placed on the inside of the barn wall while the fan is located towards the outside of the wall. Heat in the barn will keep these louvers from freezing closed."

6. Fan and motor requirements

Poultry house ventilation fans and motors (usually smaller than 1/4 hp) must meet many important requirements if they are to operate successfully

under the severe usage they receive, i.e. if they must operate twenty-four hours a day, day after day through a whole season without any attention, exhausting dust, moisture, and ammonia fumes all of which may cause damage to the motor-winding, movable parts and blades.

These motor requirements may be listed as follow:

- 1-Motors should be enclosed for protection against a moist dirty atmosphere
- 2-Motors should have bearings and a lubrication system which require only infrequent attention (Either ball or sleeve-bearing motors will perform satisfactorily providing the lubrication system is as near foolproof as possible. Sealed permanently lubricated bearings are preferable)
- 3-Motor type should be such as to insure immediate starting whenever the current is turned on, even after long periods of disuse under poultry house conditions. H. Mitchell (76) states that: "Because of their unstable speed characteristics and low starting torque, shaded pole motors are not suitable for this application. Split-phase or capacitor motors have proven to be satisfactory."
- 4-Thermal protection should be provided to preclude any possibility of overheating and fire in case of stalling. (Protection against motor burnouts)

Overload switches on electric lines feeding each motor are considered essential for the protection of the motor and the building. A switch with as low an amperage rating as possible should be used to obtain the greatest amount of protection. Kalbfleisch and White (57) recommend the following size of overload switch: (Table 56)

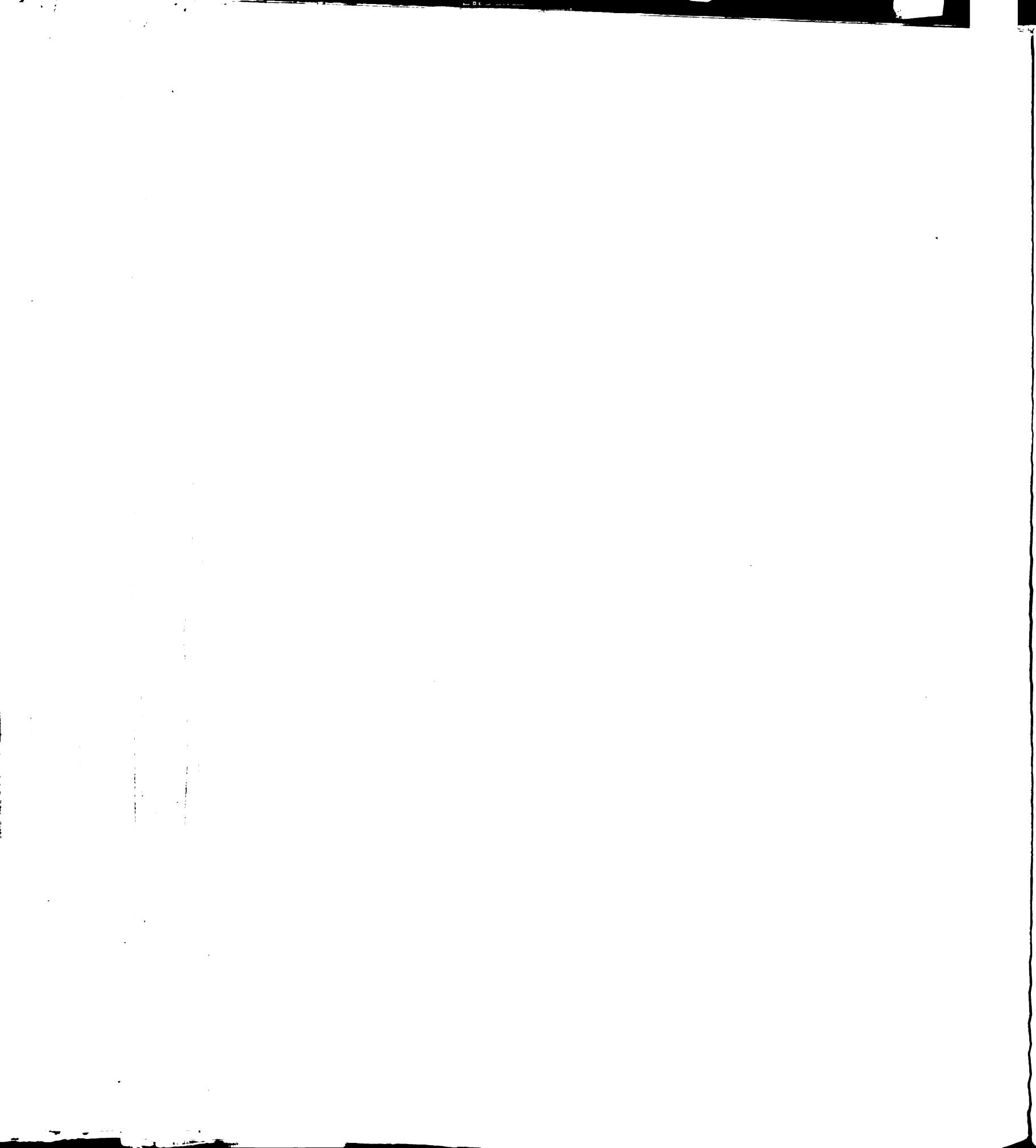


Table 56. Amperage of overload switch *

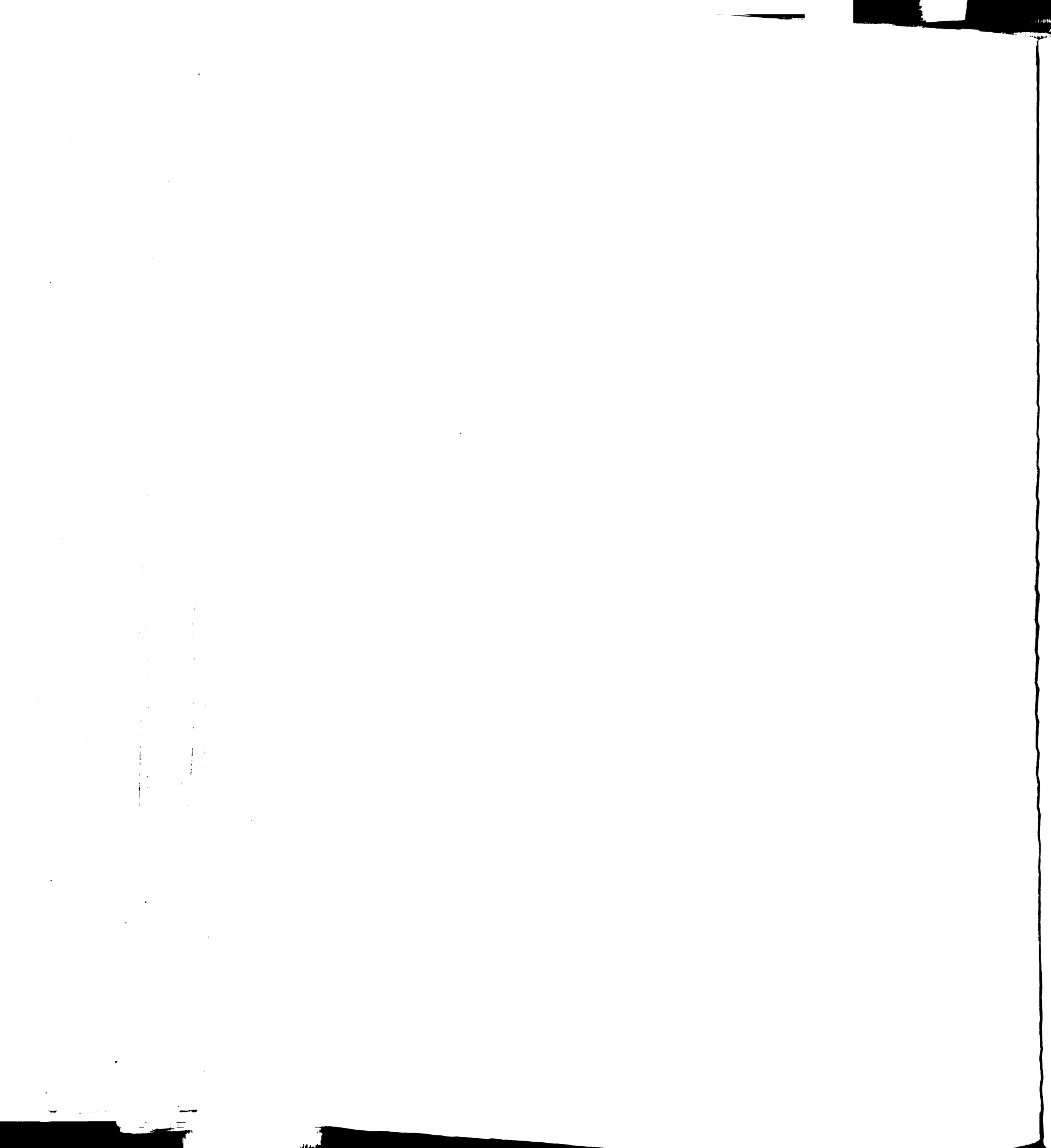
Diameter of fan (inches)	Horse power of motor	Motor speed rpm	Capacity of fan (cfm free air)	Voltage of supply line	Amperage of overload switch
8	1/20	1725	500	110	-
12	1/10	1725	1100	110	1/2 amp
12	1/8	1725	1500	110	1 amp
16	1/4	1725	2600	110	3 amp
16	1/3	1725	3000	110	5 amp
18	1/3	1725	3600	110	5 amp
				220	2 amp
	1/2	1725	4000	110	6 amp
				220	3 amp

* From Kolbfleisch and White (57)

All electrical wiring for motors, switches, and thermostats should be installed by a competent electrician and properly inspected. Separate fuses should be provided for the motors and all wiring should conform to local electrical standards.

As to the blades, their requirements are:

- 1-The blades should be of such design that the motor cannot be overloaded and excessively heated by changes in operating pressures.
- 2-The blades should be protected against rusting and corrosion.
- 3-The blades should be shaped so as to accumulate a minimum of dust (self-cleaning).
- 4-The fan wheel and motor should be dynamically balanced to assure long bearing life.



7. The fan duct

a. In the fan duct really helpful and necessary? The main purpose of winter ventilation is to remove moisture at the expense of the smallest amount of heat possible. Kalbfleisch and White (56) doing research on dairy stable ventilation came to the conclusion that the use of ducts extending to the floor is slightly disadvantageous with respect to moisture removal. In other words, the use of ducts results in a small increase in the heat loss through the ventilation system for the removal of given amount of moisture. The following table (Table 57) shows the saving of heat they found by the use of air removal at ceiling level as compared to floor level.

Table 57. Heat losses from floor and ceiling ventilators * (Btu loss per 100 grains of moisture removed at floor and ceiling outlets)

Outside temperature (°F)	Level of air removal	Inside conditions		Heat loss per 100 grains of water removed from the barn (Btu/100 grains)	Reduction or Saving in heat loss by air removal at ceiling	
		Temperature (°F)	R.H. (%)		(Btu/100 grains)	(%)
-9	Floor	47.5°	76	46.6		
	Ceiling	51.9°	73	44.8	+1.8	+4.00%
-11	Floor	49°	73	45.3		
	Ceiling	54°	76	43.7	+1.6	+3.66%
-15	Floor	50°	81	43.6		
	Ceiling	52°	78	43.0	+0.6	+1.40%
-15	Floor	48.0	75	46.2		
	Ceiling	51.5	80	42.7	+3.5	+8.20%
-15	Floor	48.5	83	42.8		
	Ceiling	52.5	75	43.8	-1.0	-2.2
-30	Floor	45.0	81	45.3		
	Ceiling	50.5	84	41.9	+3.4	+8.10%
-30	Floor	45.0	81	45.3		
	Ceiling	49.5	74	43.2	+2.1	+4.87%
-30	Floor	43.5	92	42.4		
	Ceiling	48.0	87	42.1	+0.3	+0.71%
-30	Floor	47.0	90	43.8		
	Ceiling	51.5	78	43.3	+0.5	+1.15%
-30	Floor	48.5	84	42.8		
	Ceiling	51.5	84	38.3	+4.5	+10.72%
-30	Floor	44.0	96	41.0		
	Ceiling	48.0	90	41.2	-0.2	-

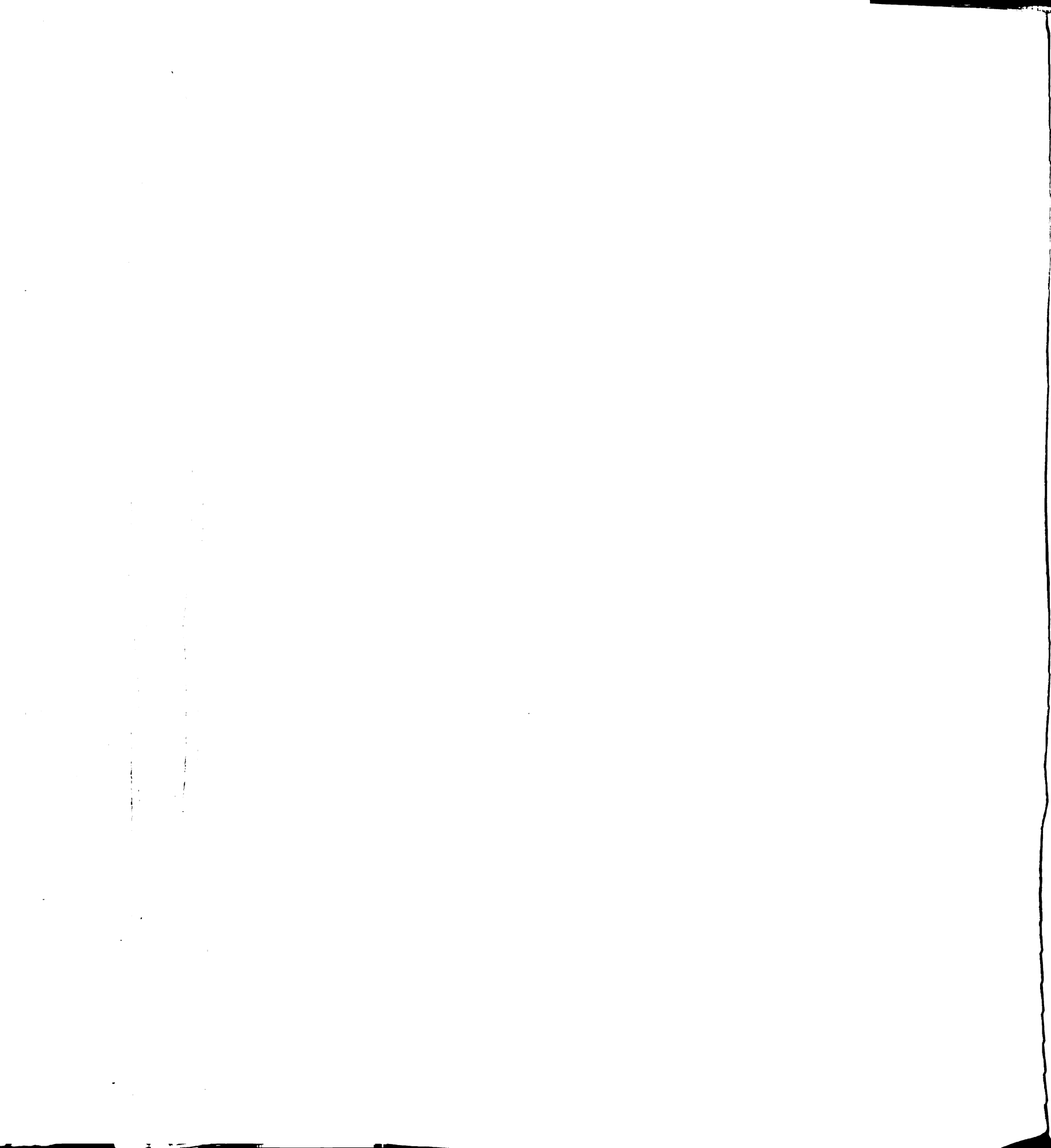
* From data by Kalbfleisch and White (56)

It seems logical to assume that the air of poultry houses would behave the same as the air in dairy barns, i.e., the air which collects at the ceiling is not only warmer than the air at the floor but contains also a relatively greater amount of moisture and ammonia gas (density relative to air = 0.587) than the colder air at the floor. Kalbfleisch and White (57) research would then mean that even in poultry houses, the construction of ventilation ducts could be neglected without any harm to the poultry house conditions. In addition to the slight heat saving, the installation of fans would be simplified and there would be a saving of both labor and material. In dairy barns one additional advantage is encountered because of the greater inconvenience of ducts and the difficulty of locating them out of the way.

The above assumption would have to be tested, since in dairy barns the fan is directly above a rather dry concrete alley instead of being located directly above the dropping pit as recommended in poultry houses. Bringing a duct very close to the wet litter or wet dejections in the pit may prove to be more advantageous as found by Stapleton and Cox (25) in the experiment described later in this section. The writer believes however that more tests are required on this matter, since not using a duct would make the use of fans still more appealing to poultrymen.

It should be noted that ducts may also be useful if it is desired to throttle down the ventilation air rate by the use of a shutter or damper.

b. Fan duct construction. Exhaust ducts for fans can be made of any tight building material (105), such as wood, cement asbestos, or insulation board, or of metal, such as those commercially available. A door is built in the duct back of the motor so it can be reached for cleaning and lubrication and opened for mild-weather ventilation. The bottom of the duct should be 15 to 18 inches from the top of the litter.



In a two-fan system with the fans side by side, two fans may be placed in the same duct (33) or else only one of the two can be equipped with a duct (105).

Exhaust duct cross sectional area and dimensions can vary considerably. The following dimensions suggested by Fairbanks and Goodman (31) are presented here as a guide. (Table 58)

Table 58. Area and dimensions of duct for fans of different capacity and diameter*

Capacity of fan (cfm)	Approximate diameter of fan (inches)	Area of duct (sq.in)	Dimensions of duct (inches)
270	9 to 10	114	12 x 12
400	9 to 10	114	12 x 12
540	9 to 10	168	12 x 14
665	10	210	14 x 15
800	12	240	15 x 16
1000	12	304	16 x 19
1200	12	360	18 x 20
1400	12	396	18 x 22
1600	14	437	19 x 23
1800	14	480	20 x 24
2000	16	528	22 x 24

* From Fairbanks and Goodman (31)

3. Location of fans

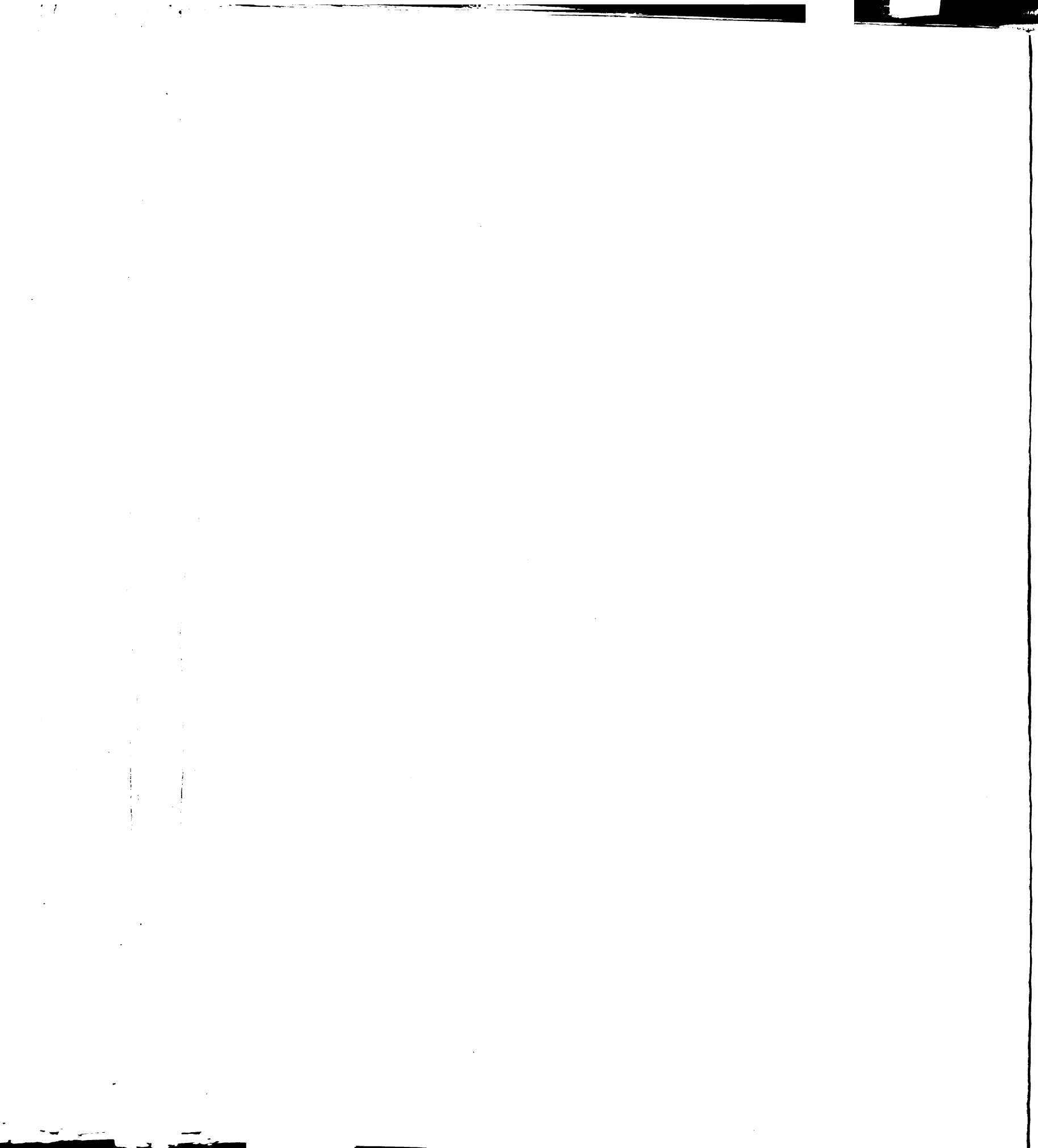
The general recommendation as to fan location is to install them on the side of the building away from the prevailing wind. Bell and Moore(14), Michigan, state: "If placed on the south or east side, it will not need to work against the prevailing winds."

As in the case of dairy barns, the intakes and fan (or fans) should be located in relation to each other so that fresh air will sweep through all parts of the house in traveling from the intakes to the exhaust duct without leaving any dead air pockets.

According to Oliver (78): "It seems that the best place from which to exhaust the air and moisture is from or near the dropping pit." The reason he gives is that: "this is probably the point of greatest moisture concentration and it is desirable to draw the dry air across the litter first to maintain dry litter and across the droppings last, since an excess of moisture in the droppings pit will not create the problem presented by wet litter on the floor."

N. Mitchell (76) makes a similar recommendation: "In poultry houses with the perches located along one side, the fan and exhaust duct should be located about the center of that side and the intakes distributed along the ends and opposite side." The reason he gives for this arrangement is that it prevents the cold air from the intakes spilling on the birds at night when they are on the perches."

Stapleton and Cox (95) also came to the same conclusion as a result of their tests. They made hygrothermograph recordings near the ceiling and near the floor of both the front and rear of the tested house. The recordings showed that: "... the fan location to give the greatest amount of water removal for a given amount of heat removal was near the floor in the droppings pit." They also add: "If no droppings pit is used, it is felt



that compromise must be made. The fan intake should be close to the wettest place in the house and also in the wall opposite the greatest glass exposure."

H. Mitchell (76) makes the following recommendation for dairy barn fan installation: "In barns up to 50 or 60 feet in length the fan and exhaust duct may be located in either side or end. In longer barns they should be located in the center of one side. In L-shaped barns the fan should be located near the angle." He also adds: "In cases where a two fan system is used, both fans should be located side by side. If they are separated each fan will have its sphere of influence. At the neutral point between these two spheres dead air space or eddying currents may develop."

According to Fairbanks and Goodman (33): "As with the outlet flue, for stables up to 110 feet in lengths, fans may, in general, be located in any convenient place. For longer stables, the fan may be near the middle of one side or, if the stable is in the form of an L, they are best placed near the angle of the building."

Turner (105) also states: "Install both fans of the two-fan system together in the wall, so air movement will always be in the same direction."

In the writer's opinion, for houses much longer than 100 feet, fans should probably best be uniformly spaced on one wall and each controlled by a separate thermostat and hand switch.

III. Theoretical Performance of Fan Ventilation in Poultry Laying Houses

If a composite chart, nomograph or slide rule for moisture removal is combined with a composite chart, nomograph or slide rule for temperature control, a heat and moisture balance composite chart, nomograph or slide rule is obtained which may be used for an analysis of the theoretical performance of fan ventilation in poultry laying houses, i.e. for a study of the theoretical inside relative humidity to be expected, the difference in house conditions resulting from the use of different thermostat settings, etc.

1. Goldrich's heat and moisture balance composite chart

Although the writer presented two Goldrich composite charts (one for moisture removal and one for temperature control), Goldrich (36) in his original paper presented only one heat and moisture balance composite chart grouping these two. This heat and moisture balance composite chart is presented on the next page. An illustration is presented in the right corner and is explained here to show how such a heat and moisture balance tool can be used to analyse theoretically the performance of a fan ventilation system.

Application

Given: Outside temperature = 25°F
 Outside relative humidity = 85%
 Inside temperature = 65°F
 Rate of moisture production in the house = 1.25 lb./hr, 50 birds
 (or 2.5 lb./hr, 100 birds)
 Weight of birds = 5 lb
 Total heat production (U_{AT}) = According to Mitchell and Velley's curve A
 Percentage of latent heat (L) = According to Mitchell and Velley's curve B

Net sensible heat from litter decomposition (H_L) = 0 Btu/hr, hen

Heat from the sun, lights, etc. (H_S) = 5 Btu/hr, hen

Heat loss characteristic of the house (AU) = 30 Btu/hr, °F, 10 hens

Wanted: The theoretical inside relative humidity to be expected when the above conditions remains sufficiently long enough for the house conditions to reach the equilibrium or balance point.

True mathematical solution:

$V_G = 1454.2 \text{ cfm}/50 \text{ birds}$ (from section on composite charts)

instead of 1800 cfm/50 birds as shown on the chart

(N.B. = The temperature part of this chart was found to be considerably in error.)

In terms of cfm per hen, $V_G = 0.42454 \text{ cfm/bird}$ as compared to 0.6066 cfm per bird as shown on the chart.

v_i (65°F, 80% assumed) = 13.44 (From Table 44)

$W_{o\text{sat.}}$ (25°F) = $28.26 \times 10^{-4} \text{ lb/lb dry air}^*$

W_o (25°F, 85%) = $28.26 \times 85\% = 24.02$

$$\Delta W = \frac{W_T \times v_i}{6000 V_L} = \frac{2.50 \times 13.44}{6000 \times 0.42454} = \frac{2.50 \times 13.44}{2567.27} = \frac{2.50 \times 13.44}{0.256727 \times 10^{-4}} = 113.23 \times 10^{-4}$$

$W_{i\text{sat.}}$ (65°F) = $132.6 \times 10^{-4} \text{ lb/lb dry air}^*$

$W_{i\text{act.}} = W_{o\text{act.}} + \Delta W = 24.02 + 113.23 = 137.25$

$$R.H._i = \frac{W_{i\text{act.}} \times 100}{W_{i\text{sat.}}} = \frac{137.25 \times 100}{132.6} = >100 = 100\%$$

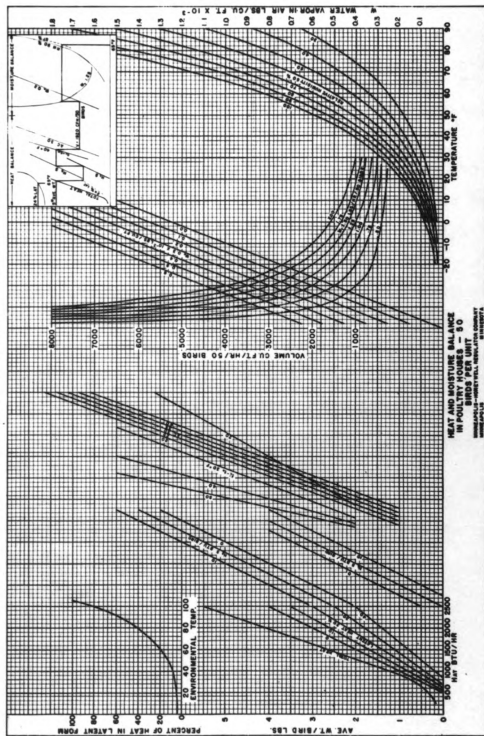
Solution by use of the heat and moisture balance composite chart

By referring to the composite chart, the answer is found to be 87%.

The reason for the big difference from the true mathematical solution (100%)

* From Zimmerman and Lavine (1945)

**GOLDRICH COMPOSITE CHART
FOR THE THEORETICAL ANALYSIS OF FAN VENTILATION
PERFORMANCE IN POULTRY LAYING HOUSES (36)**



is due to the error in the temperature side of Goldrich's chart, as shown in the section on composite charts.

2. Heat and moisture balance nomographs and slide rules

No heat and moisture, balance nomographs and slide rules will be presented here for lack of space, however, such tools can be obtain in the following manner.

(1) To obtain a heat and moisture balance nomograph: Use the writer's nomograph for moisture removal which was modified to have its V_L scale vertical and join it with the writer's nomograph for temperature control by putting the V_S scale on top of the V_L scale.

(2) To obtain a heat and moisture balance nomograph: Use the writer's exact slide rule for moisture removal and join it to the writer's corrected slide rule for temperature control by use of a middle support.

It should be noted also that it is possible to solve heat and moisture balance problems by using successively a temperature control nomograph or slide rule and a moisture removal nomograph or slide rule, without having them combined on the same sheet of paper.

Given: Same conditions than the one given for Goldrich's heat and moisture balance composite chart, i.e.:

$$Q_g (\text{Birds}) = 17.59 \div 50 = 35.18 \text{ Btu/Hr, bird}$$

$$Q_g (\text{Sun}) = 5.0 \text{ Btu/Hr, bird}$$

$$Q_g (\text{Litter}) = 5.0 \text{ Btu/Hr, bird}$$

$$Q_{g\text{Total}} = 35.18 + 5.0 + 5.0 = 45.18 \text{ Btu/Hr, bird}$$

$$AC = 30 \div 50 = 0.60$$

$$\Delta t = 65 - 25 = 40^\circ$$

$$t_o = 25^{\circ}\text{F}$$

$$\text{R.H.}_o = 85\%$$

$$W_T = 2.50 \text{ lb/hr, 100 birds}$$

Wanted: The theoretical inside relative humidity to be expected when the above conditions remain sufficiently long enough for the house conditions to reach the equilibrium or balance point.

True mathematical solution:

100% as shown above

Solution by use of the writer's heat and moisture balance nomograph:

1-Use of nomograph for temperature control

$$V_S = 0.46$$

2-Use of nomograph for moisture removal

$$\Delta W = 90 \times 10^{-4} \text{ lb/Cu.ft. dry air}$$

$$W_{iact} = 100 \times 10^{-4} \text{ lb/Cu.ft. dry air}$$

$$\text{R.H.}_i (65^{\circ}\text{F}) = >100 = 100\%$$

Solution by use of the writer's heat and moisture balance slide rule

1-Use of corrected slide rule for temperature control

$$V_S \text{ appr.} = 0.43$$

$$V_S \text{ corrected} = 0.46$$

2-Use of exact slide rule for moisture removal

$$W = 122 \times 10^{-4} \text{ lb/lb dry air}$$

$$W_{iact} = 145 \times 10^{-4} \text{ lb/lb dry air (approximately, by extending the scale)}$$

$$W_{isat.} = 132 \times 10^{-4} \text{ lb/lb dry air (approximately, by extending the scale)}$$

$$\text{Then R.H.}_i (65^{\circ}\text{F}) = >100 = 100\%$$

To conclude from the above discussion and applications, it may be said that heat and moisture balance composite charts, nomographs and slide rules will prove useful in the future only:

1-If they can be made the most accurate possible (The writer's efforts have been directed toward this goal)

2-If more reliable heat and moisture data are obtained both from calorimeter tests and tests in actual poultry houses

3-If more knowledge is acquired of the mechanism and rate of moisture evaporation from poultry house litters at different environmental temperatures and for different depths and management of the litter.

Until such data are available, any analysis of the theoretical performance of fan ventilation will be subjected to considerable error.

IV. ACTUAL PERFORMANCE OF FAN VENTILATION IN POULTRY LAYING HOUSES

A. Purpose of the Study

The purposes of the following tests on the actual performance of fan ventilation in poultry laying houses were as follow:

- 1-To determine the trend in the total heat available per hen at different environmental temperatures, under actual housing conditions, and to check them with values obtained in calorimeter tests.
- 2-To determine the trend in the net sensible heat coming from secondary sources.
- 3-To determine the theoretical moisture removal capacity of fans and, if possible, to relate it to the total amount of water produced in the house and to the change in the litter moisture content.
- 4-To determine whether or not ventilating rates as high as four cfm per hen offer a good recommendation for houses of different insulating value.
- 5-To determine whether or not intermittent ventilation may be applied successfully in houses of relatively high insulating value (AC value around 1.0 or lower).
- 6-To check the usefulness of ventilation slide rules in analyzing test data.
- 7-To indicate the need for further research.

Two poultry houses were studied by the writer for actual performance:
(a) Mr. Joseph Rohmwalter's poultry house, Okemos, Route I, Michigan, and
(b) Mr. Albert Levasseur's poultry house, St. Albert, P.Q., Canada. A description of the flocks, houses and ventilating systems will be presented here. The first house was studied for the period extending from February 16, to March 17, while the second house was studied only for the period from December 16, to December 20.

Each house will be described and analyzed separately then some general conclusions will be drawn from the two tests.

2. Test of Mr. Joseph Schmelter's Poultry House
(Okemos, Route 1, Michigan)

1. The birds

The bird population was composed of 1228 Leghorn laying hens and 100 males at the beginning of the experiment (February 16), making a total of 2028 birds. At the end of the experiment (March 17), the number of hens was reduced to 1898 making a total bird population of 1998 birds (March 17).

The average weight of the birds was computed on February the 26th, by weighing individually 200 birds chosen at random (approximately 10% of the bird population), and found to be 4.95 pounds per bird. The range of variation was from 3.7 to 5.9 pounds (this is a remarkably heavy line of Leghorns).

2. The house dimensions and arrangement

The house dimensions are 30' x 201'. Eighty-five windows are facing south (front) and eleven are facing north (back). The nests are on two tiers along the rear walls and the roosts and dropping pits are about three feet from this wall. A V-shaped metal waterer is fixed to the front wall and extends nearly all its length. Water is continuously flowing in order to maintain about 1/4 inch of water in the bottom of the V-shaped waterer. Feeders are placed perpendicular to the front wall at a distance of about four feet from this wall. The litter lies directly on a sandy soil and is made of baled straw. During the test period its depth varied from 6 to 10 inches all over the floor with an average of about 8 inches.



Fig. 47a. Front view of Mr. Joseph Rohmwalter's poultry house, Okemos, Route I, Michigan. Eighty-five windows face south. From left to right: first, second and third fan as referred to in the text. This house is 30' by 200' and contains more than 2000 Leghorn hens.



Fig. 47b. End view and rear view of Mr. Joseph Rohmwalter's poultry house. Eleven windows face north.

3. The house construction and insulating value

The house construction is as follows:

- 1-Lower part of walls = 8 inches blocks
- 2-Upper part of walls = 3/4 inch boards, building paper, 7/8 inch boards
- 3-Roof or ceiling = metal roofing, 1 inch Celotex
- 4-Windows = Single windows
- 5-Door near the third fan = 7/8 inch boards
- 6-Door near the first fan = Heavy paper
- 7-Foundation wall between the feed room and the pen = 13 inches concrete

The writer computed the area, heat loss and AC value of each of the construction units. The following tables (Tables 59 and 60) summarize these computations:

Table 59. Summary of the computations for the total heat loss of Mr. Joseph Rohrwalter's poultry house, Okemos, Route I, Michigan

Unit	A Radiation or exposed area (sq. ft)	R Insulation value	AU Heat loss AU = A ÷ R
1-Roof	7950.0	3.75	2120.0
2-Lower part of wall	1050.3 149.2	1.89 1.82	556.0 82.0
3-Upper part of wall	1061.1	2.26	469.0
4-Windows	428.0	0.90	476.0
5-Doors	100.0	1.73	57.8
Total	10,738.7 A total		3,260.8 AU total

$$C = U_{\text{aver.}} = \frac{\text{AU total}}{\text{A total}} = \frac{3,260.8}{10,738.7} = 0.303$$

Table 62. AC value of Mr. Joseph Rohmwalter's poultry house, Okenos, Route I, Michigan.

Period	A total Radiation area (sq.ft)	Number of birds	A Radiation or exposed area per bird (sq.ft)	C or U aver. value	AC value
Febr. 11-17	10,738.7	1928	5.57	0.303	1.686
Febr. 18-24	10,738.7	1919	5.60	0.303	1.697
Febr. 25-5	10,738.7	1912	5.62	0.303	1.714
March 6-12	10,738.7	1906	5.64	0.303	1.710
March 13-17	10,738.7	1898	5.66	0.303	1.714

For practical purposes the AC value of this poultry house can be said to be 1.70 for the full length of the test.

4. The fans, thermostats and air intakes

Three 19-inch dual-speed fans (1/8 hp, 1140/860 rpm) manufactured by James Manufacturing Company, Fort Atkinson, Wisconsin, are spaced 66 feet from each other in the south wall. These fans have a rated capacity of 2950 cfm (high volume) and 1966 cfm (low volume), free delivery. The thermostats are set at 35°F most of the time. No special air intakes are provided; some front and rear windows are opened 1/2 inch at the top to serve as intakes. However, especially during windy days, the 10' x 10' door in the west end of the house seems to be the main source of air intake because of the fact that the frame of this door is made of poultry mesh and only partially covered with heavy paper.

5. Instrumentation

a. Temperature and humidity instruments

Thermographs and hygromographs. Previous to the use of the 16-point Brown potentiometer recorder, the writer made numerous tests to determine the degree of accuracy and reproducibility of two thermographs and three hygromographs that were available. The results of this study are summarized here:

1-All instruments seem to give satisfactorily accurate results only in a rather small range (probably 30-35°F and 15-20% relative humidity) half being on each side of the adjusted temperature or humidity.

2-Instruments adjusted at inside conditions give results not at all satisfactory when subjected to outside conditions.

3-During a heavy rain (November 16-17), the maximum relative humidity indicated by the different instruments were:

C.I. 342 x = Maximum of 100% for a short time then 96% for a long time

Bact.T. = Maximum of 83% for a long time

A.B. 632-33 = Maximum of 20% for a long time

as compared to a maximum of 100% for a short time and 96% for a long time as given by the Airport Weather Bureau. (N.B. = The only instrument giving good performance during the rain had an unsatisfactory performance before the rain).

4-All adjustments were made with adjustment screws or knobs. There may be some possibility of better corrections by changing linkage lengths but such adjustments would require instruction books for each of these different instruments, and would require a room with controlled humidity and temperature.

5-It is also probable that proper cleaning of the hair elements might give better results, since dust collected on them may affect their performance.

But nevertheless, one should not expect higher accuracy than 5% under the best conditions.

6-Calibration curves seem impossible because of the lack of reproducibility and the need for constant checking and adjustment.

7-Temperature recorders, in general, have greater reproducibility than relative humidity recorders.

8-These instruments would have to be checked and adjusted more thoroughly before being used in research, and interpretation of the results should be very careful.

9-Thermocouples for dry bulb temperature should be preferred to these instruments because of greater accuracy and reproducibility.

10-Thermocouple arrangements for wet bulb determination should be preferred to these instruments because of greater accuracy and reproducibility.

As a result of this study, a recording potentiometer was used for both dry bulb and wet bulb recording in order to get a greater degree of accuracy.

Recording strip-chart potentiometer. A 16-point recording potentiometer was used starting February 11.

The thermocouples were made of twenty-six gage copper and constantan wires. They were installed by first wrapping them in masking tape and then fastening the masking tape to the wall, roof or posts by the use of a staple gun. This method was thought preferable to the use of a staple gun directly over the thermocouples, since this might possibly damage the insulation and provide undesirable contact between adjacent thermocouples.

After their installation was complete, each thermocouple was checked in place by the use of an ice water solution in a thermos bottle. The potentiometer was first set to record a given point and the recording or printing mechanism was stopped. This procedure made it possible to observe the

changes in temperature at one point for a period of time of any length since when the recording mechanism is stopped, the same thermocouple remains in contact all the time and the indicating mechanism may be used to show the temperature changes for the point considered. The thermocouple junction was then placed in the ice water solution and the indicated temperature was checked over a period of approximately one and a half minutes after which the recording mechanism was set in operation to allow the printing of that point. The recording mechanism was stopped again and the procedure was repeated for each of the 15 remaining thermocouples. Two thermocouples were found out of order and replaced.

Three copper-constantan thermocouples were required for the time of operation instruments (as will be shown later), and two for the wet bulb instruments, eleven thermocouples being therefore left for measuring outside and inside air temperature. The use of each thermocouple is presented here.

Thermocouple	Use
Thermocouple No. 1	Time of operation of the first fan.
Thermocouple No. 2	Inside temperature 5' above the floor, near the first fan.
Thermocouple No. 3	Wet bulb temperature, 5' above the floor, near the first fan.
Thermocouple No. 4	Inside temperature, 1' above the floor, near the first fan.
Thermocouple No. 5	Outside temperature, 5' above the floor, near the first fan, front of the house.
Thermocouple No. 6	Time of operation of the second fan.
Thermocouple No. 7	Inside temperature, 5' above the floor, near the second fan.
Thermocouple No. 8	Time of operation of the third fan.

Thermocouple No. 9	Inside temperature, 5' above the floor, near the third fan.
Thermocouple No.10	Wet bulb temperature, 5' above the floor, near the third fan.
Thermocouple No.11	Storage room temperature, 5' above the floor.
Thermocouple No.12	Inside temperature, 5' above the floor, near the first fan, back of the house.
Thermocouple No.13	Outside temperature, 5' above the floor, near the first fan, back of the house.
Thermocouple No.14	Inside temperature, 1' above the floor, near the second fan.
Thermocouple No.15	Litter temperature, 3 inches below the top of the deep litter, near the third fan.
Thermocouple No.16	Inside temperature, 1' above the floor, near the third fan.

Since the instrument was printing 16 temperatures in a distance of only 1/2 inch, the thermocouples were connected in such a way to give readings varying as much as possible from the previous one. A switching box and a timing circuit would have permitted the use of a greater number of thermocouples, but because of the time of operation instruments (to be described later), the cycle had to be the shortest possible and the instrument was therefore geared in such a way to print 16 temperatures every 4 minutes. Daily visits were made to the poultry house in order to check the instruments and verify the house conditions.

FLOOR DIAGRAM SHOWING THE LOCATION OF THERMOCOUPLES
(Not to scale)

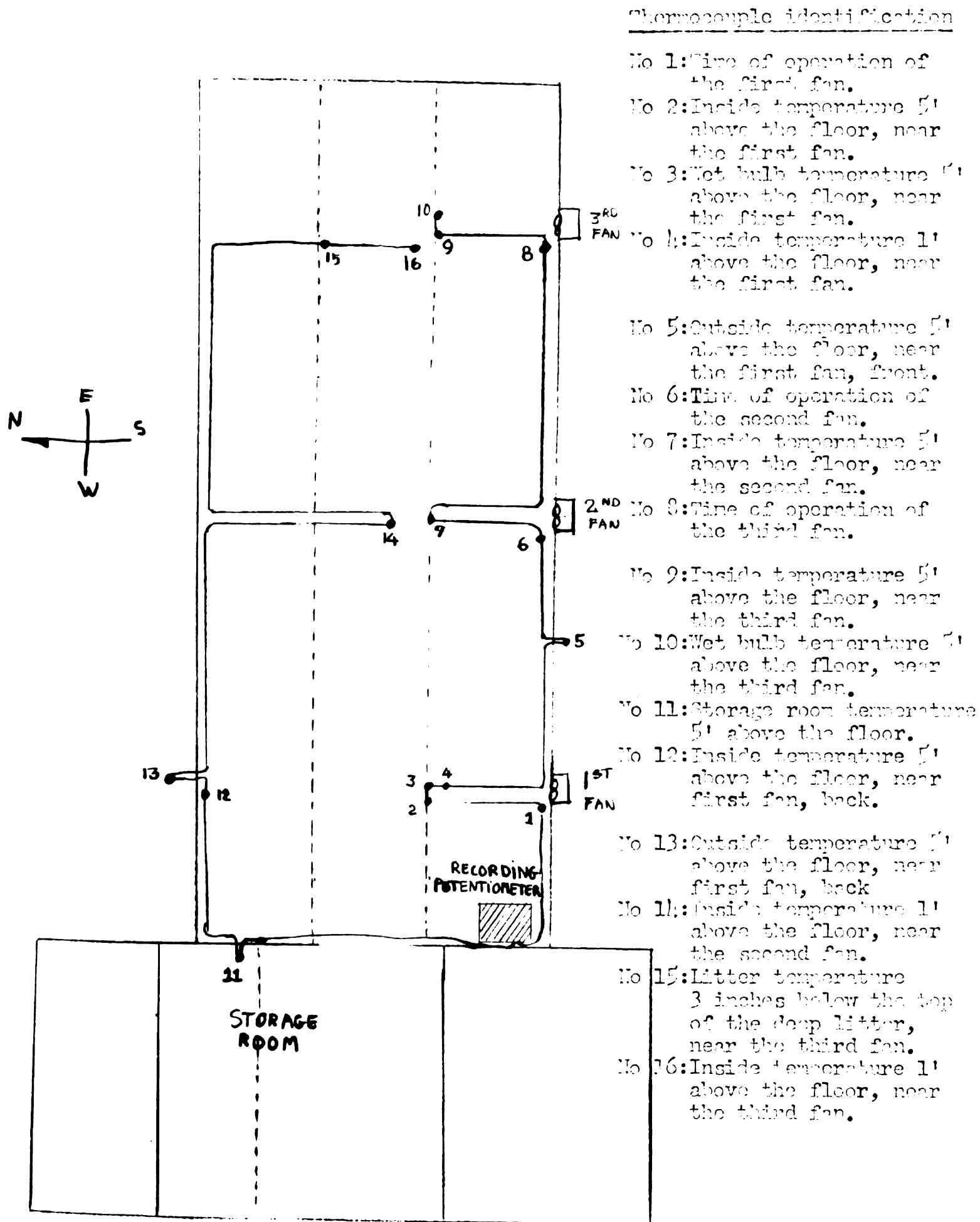




Fig. 49. The recording strip-chart potentiometer was located in a corner of the house and protected by poultry netting.

Continuous wet bulb instruments. Because of the unsatisfactory performance of hair hygrometers, two constant feed wet bulb units were built according to plans by Henderson (11) and one is shown in Fig. 50. The sensing element of these units includes a water reservoir made of a piece of 16-mm glass tubing 2 1/4 inches long, in which water surrounds a smaller glass tube containing the thermocouple. This glass tube is 5 mm in diameter and fits the standard wet bulb sock or wicking material. It is bent to an approximate right angle in order to minimize the required capillary rise. The length of the exposed horizontal wick is made approximately 1 3/4 inch and the thermocouple junction is located about 1/8 inch from the end. A 250 ml Erlenmeyer flask is used to maintain the water level in the sensing element. This quantity of water was found sufficient for a full week of operation.

Due to the nervous character of hens and the large amount of dust present in poultry houses, there is a tendency for the wet wick to become dusty very rapidly. It was therefore, found necessary to flush the wet bulbs every day (and never less than once every other day) in order to remove the dust collected on them. Tap water was used for this purpose. Distilled water was used however, in the Erlenmeyer and in the 16 ml tube. The wicks were changed at the end of each week of operation.

Each unit was installed five feet from the ground and one foot horizontally from the thermostats operating the first and the third fans. At this level and location the instruments were believed to give a better average of the poultry house humidity.

A fan was used with each of the instruments in order to increase the rate of evaporation. The location of these two fans with respect to the instruments was chosen in order to get a speed of approximately 500 feet

per minute, which is considered satisfactory to minimize the percentage of error (1).

The thermocouples for the relative humidity measurement were checked by the same method which was described earlier for temperature control recording thermocouples. In addition each unit was checked nearly every day by the use of a sling psychrometer. The difference was found to vary from 0.5 to 3% which is sufficiently accurate for the type of data being taken.



Fig. 50. A continuous wet bulb instrument built according to plans by Henderson. Both a dry and wet thermocouple are in the fan air stream.

b. Time of operation instruments. In order to make possible any heat and moisture balance determination one time of operation instrument was required for each of the three two-speed fans used in the poultry house. However, only one instrument was available for this purpose. Moreover, this instrument was of the on-and-off type (Magnet type) with the result that it was registering the same way whether the fan was on low or high speed.

Upon the suggestion of Professor D.E. Wient and graduate student Gerald Isaacs, the writer investigated in the laboratory the possibility of using light bulbs and thermocouples as a measure of the time of operation of the fans. The principle was to connect some small wattage bulbs in parallel with the motor windings, and of using thermocouples to measure the temperature change accompanying a change in the operation of the fans.

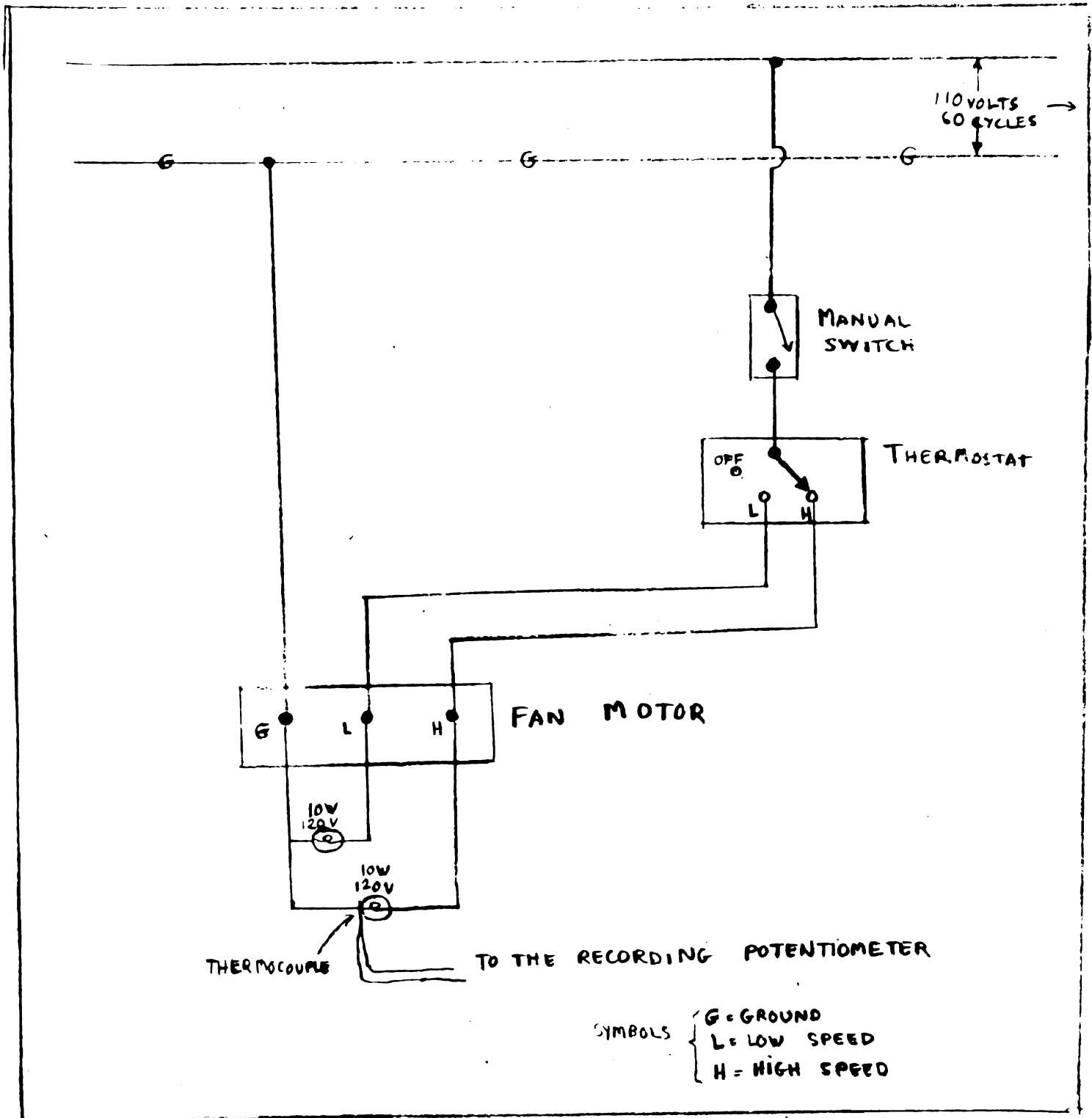
Three instruments were built on this principle, each unit consisting of two bulb sockets, two 10 watts - 120 volts bulbs, a holding panel, and a metal shield. The purpose of the metal shield was to avoid any possible disturbance of the birds at night. One of the two bulbs was connected in parallel with the low speed motor winding and the other with the high speed winding. The two bulbs were placed side by side and separated by a distance of one-half inch. A thermocouple was then fixed to the bulb in parallel with the high speed winding. This method gives a bigger temperature difference than attaching the thermocouple on the bulb in parallel with the low speed winding. With the fan running at high speed, the bulb with the thermocouple attached to was seen to indicate a temperature from 100 to 120°F (for house temperatures of 30 and 50°F respectively). With the fan running at low speed, the temperature recorded was from 70 to 80°F (for house temperatures of 30 and 50°F respectively). The big difference is due to the fact that in the first case the thermocouple receives heat by conduction because of its direct contact with the light bulb, while in the second case, the

thermocouple receives heat only by radiation and convection the bulb being one-half inch away from it. The temperature of the above thermocouple dropped to room temperature whenever the fan turned off. In order to obtain time of operation data of most accurate value, the Brown potentiometer was geared to its greatest speed in order to give the smallest time interval possible between each cycle, that is four minutes. The values obtained in this way are believed to be more reliable than those obtained with most of the commercial instruments because of the fact that the charts of such instruments must be changed every 24 hours, and also because the time given by these instruments has to be interpolated by the operator eye and no greater degree of accuracy is possible. Another advantage of the instrument developed here is that it permits grouping of the information concerning the time of operation of the fans on the same recording sheet used for dry bulb and wet bulb recording. This eliminates therefore, any possible error due to "too slow" or "too fast" movement of the clock mechanism usually used in commercial instruments.

When the thermostat mechanism changes the operation of the fan from one speed to another, for example from fast to slow, the temperature immediately starts to deviate from the vertical line of records and the cycle during which the change occurs can be readily determined.

FIGURE 51

CIRCUIT DIAGRAM OF A TEST OF OPERATION INSTRUMENT USING A
THERMOCOUPLE AS THE SENSING ELEMENT



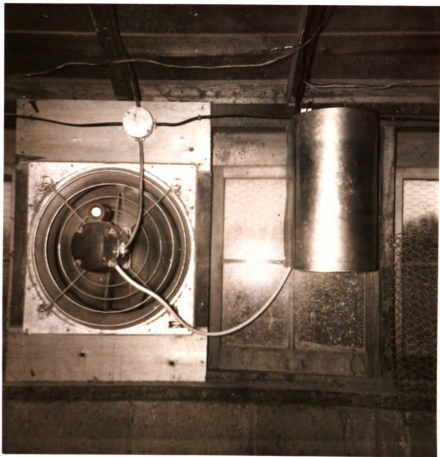


Fig. 52. View of one of the three fans with its time of operation instrument connected in parallel with the fan motor windings. The time of operation instruments were shielded to avoid any disturbance of the birds at night.

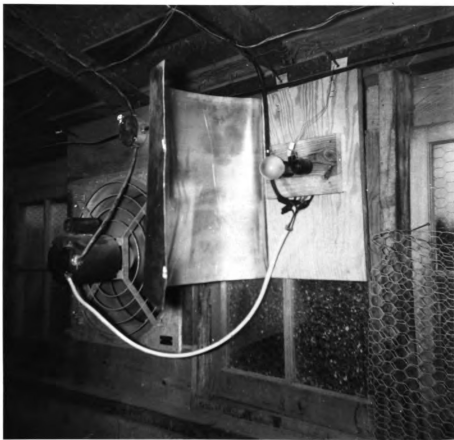


Fig. 53. View of a time of operation instrument with its aluminum shielding door opened. This fan motor had a characteristic different from the two other: one light bulb only was required. This bulb was shining and giving heat more or less according to whether the fan was on high or low speed. The thermocouple is attached to a piece of wood located one-half inch from the bulb.



Fig. 54. The two other fan motors used in the house required two bulbs in the time of operation instruments, one for the low speed and one for the high speed. The thermocouple is attached to the bulb which is in parallel with the high speed winding of the motor.

c. Litter sampling and drying instruments. To determine the moisture content of litters, samples were taken at three different times. Because of the zonal character of the deep litter which varying in depth from six to twelve inches with an average of about nine inches, separate samples were taken for approximately every three inches layer down to the ground surface below the litter. Samples of the sandy soil were also taken in order to determine its moisture content. Each sample was taken over an area of one square foot. The material collected was then mixed thoroughly in a tub, and a sample varying from 300 to 500 grams was obtained by the quartering method. This sample was immediately weighed and placed temporarily in a wax coated box with a cover.

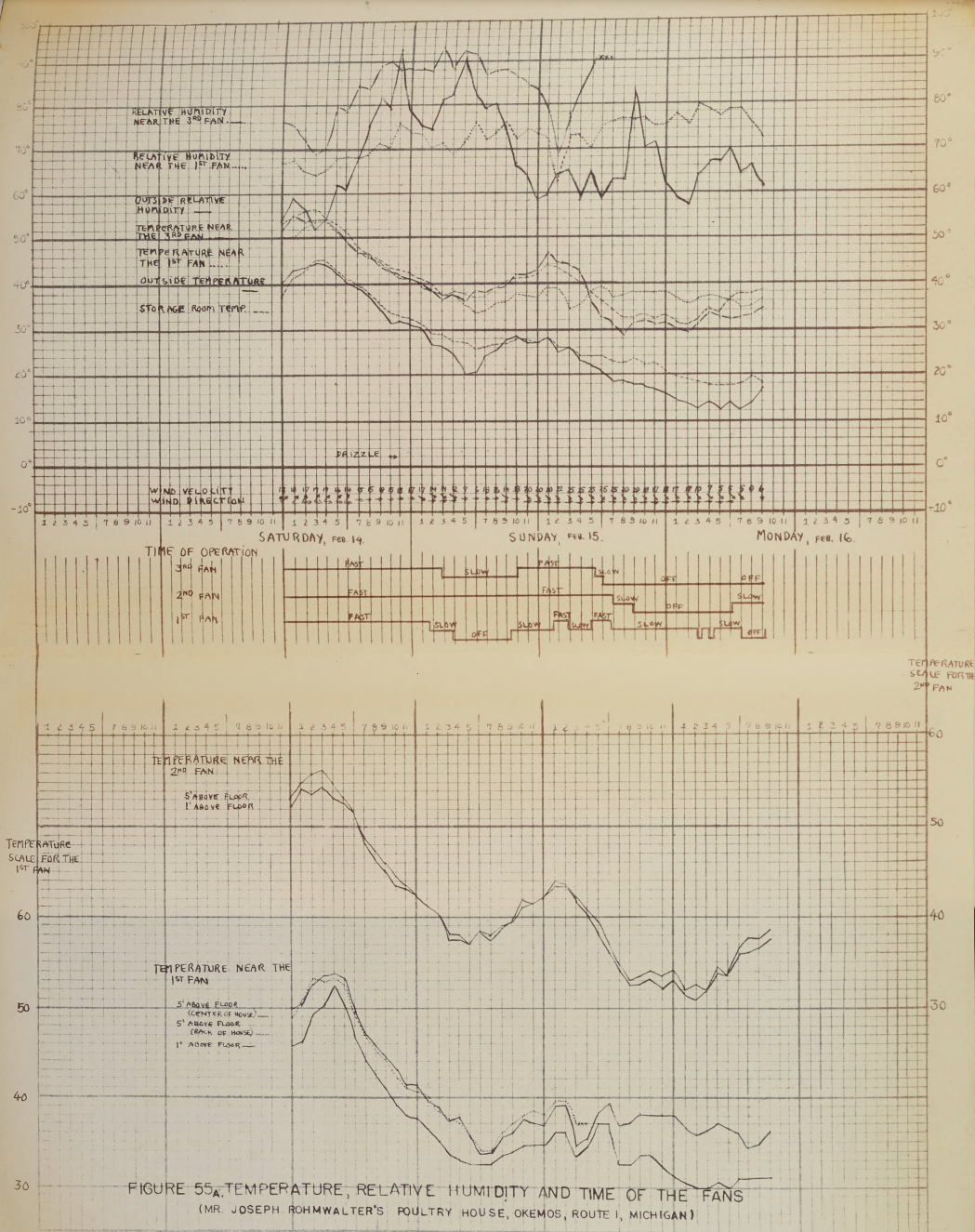
The drier was a large metal cabinet electrically heated and thermostatically controlled. The temperature of the cabinet interior was maintained between 190° and 200°F. At the end of the drying period (at least 48 hours were required) each litter sample was removed from its container and placed in an open pan to get its dry weight. The grams of moisture evaporated were found by subtracting the weight of the dry sample from that of the wet sample and the percentage of moisture content was determined on a wet basis.

6. Presentation of results

a. Temperature, relative humidity data and fan operation data. Because of the considerable amount of data collected, a graphical presentation of the results was believed to be the most desirable one (Figure 55).

In additions to the data collected by the writer, some other data from the Lansing Weather Bureau are included. These are: outside relative humidity, wind direction and wind velocity.

Because of the unreliable character of hygrothermographs when submitted to the wide range of relative humidity encountered outside, it was found more reliable to compute absolute humidity values from the readings of the Lansing Weather Bureau. The use of these values was based on a comparative study of the outside temperatures obtained with the recording potentiometer and those given by the Lansing Weather Bureau. The agreement was within one or two degrees for most of the readings. Very often the readings were the same and only in a few occasions was the difference higher than three degrees (readings twenty minutes after the hour). This similarity of temperature is due highly to the reasonably small distance between the location of the poultry house and that of the Weather Bureau. The writer assumed that for these reasons a similar resemblance would exist between the absolute relative humidity of the two locations and in his computations used the absolute humidity obtained from the Weather Bureau readings.





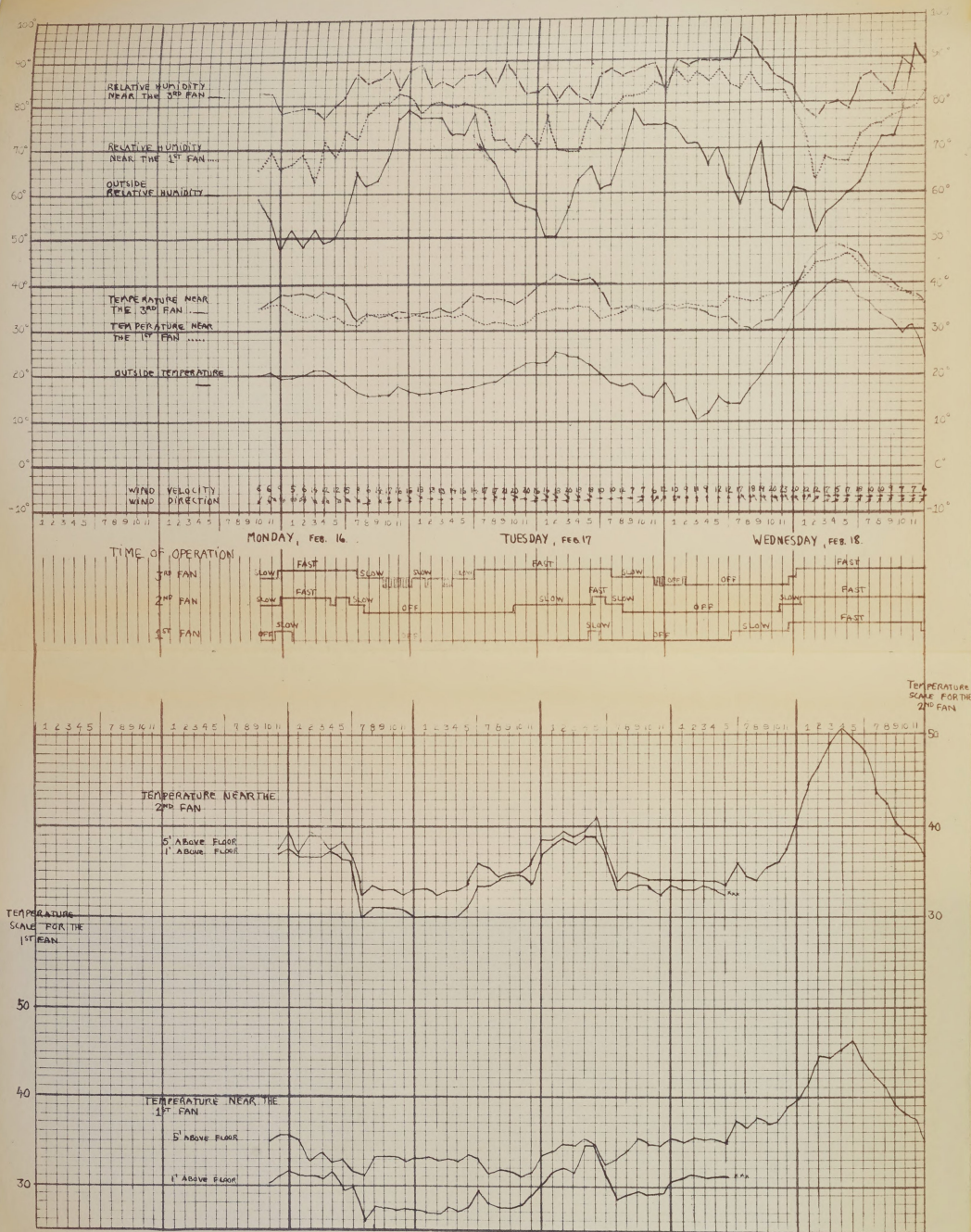
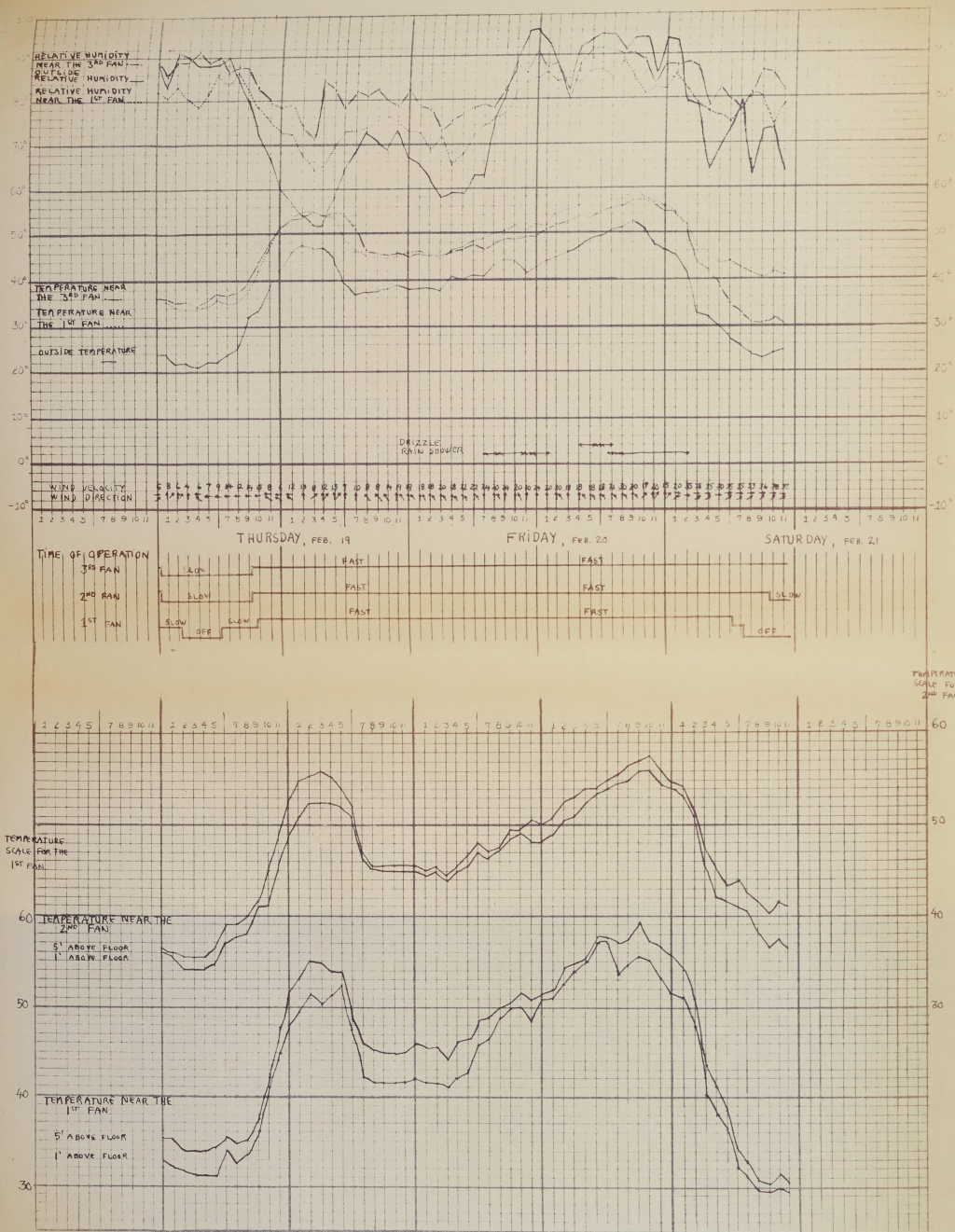
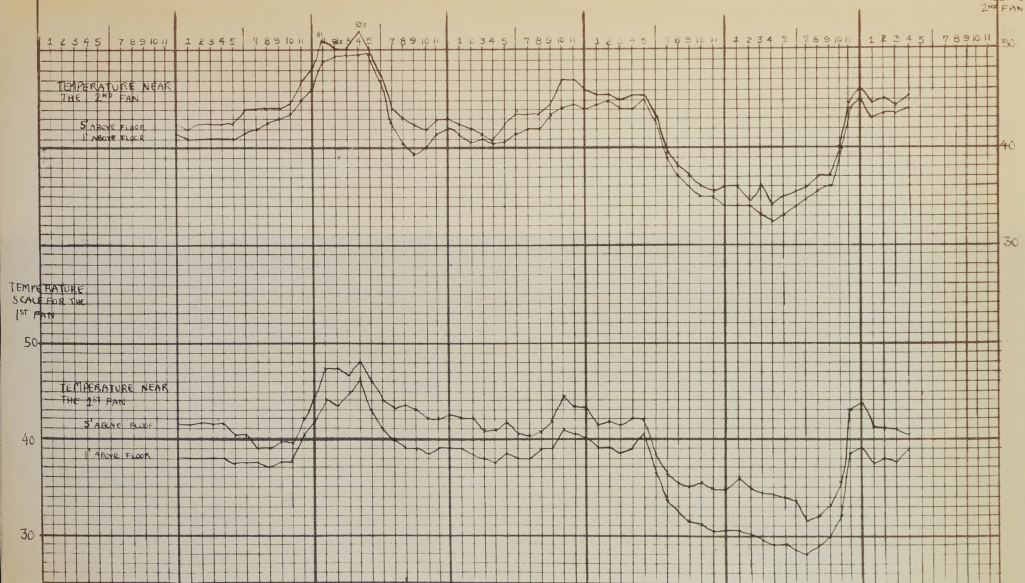
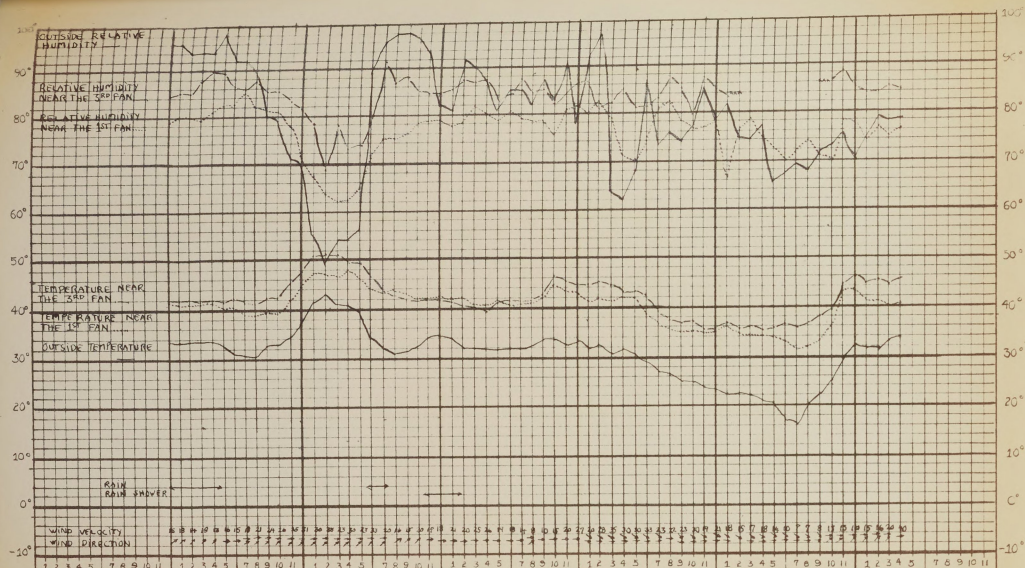


FIG. 55B











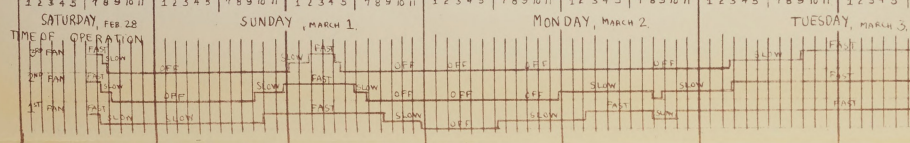
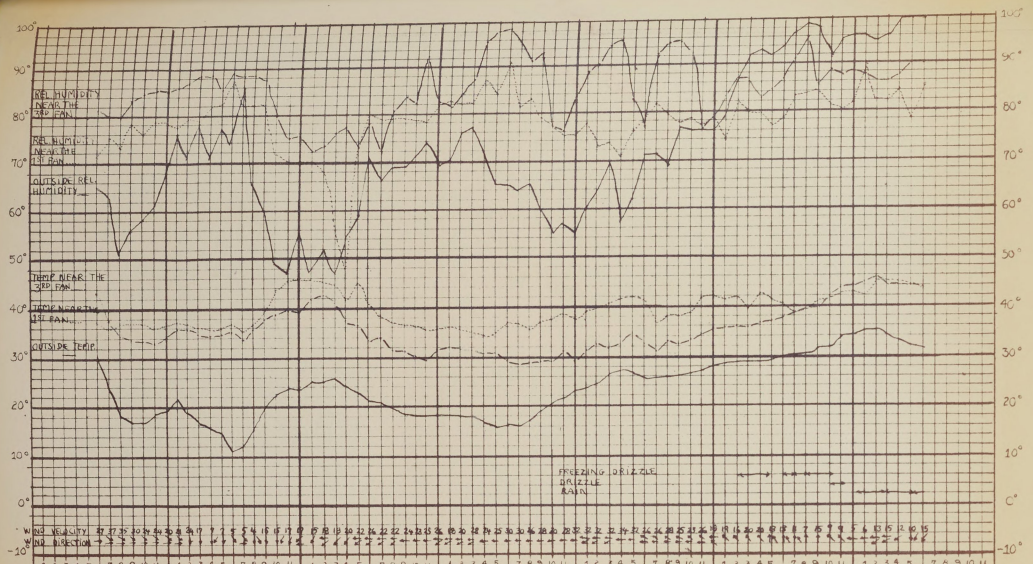
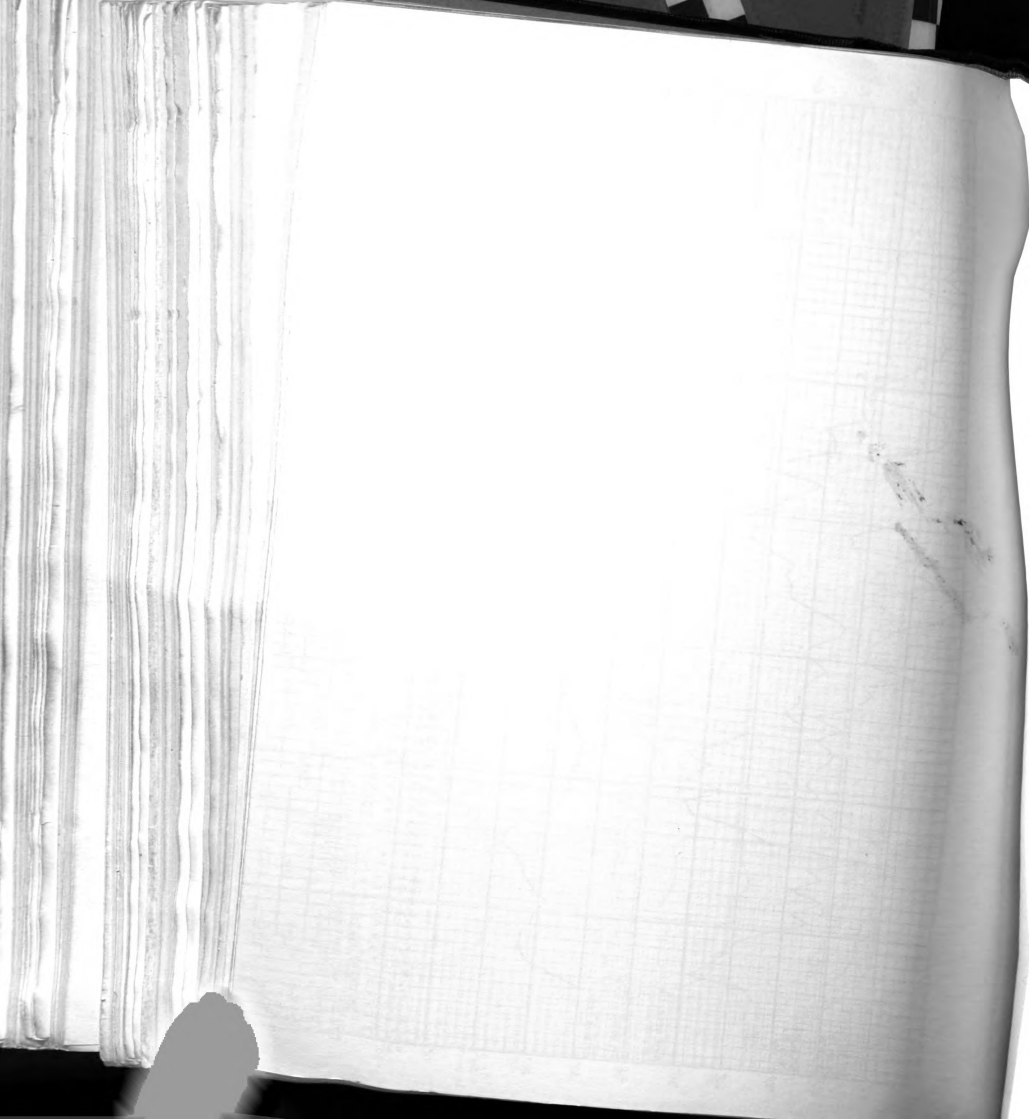
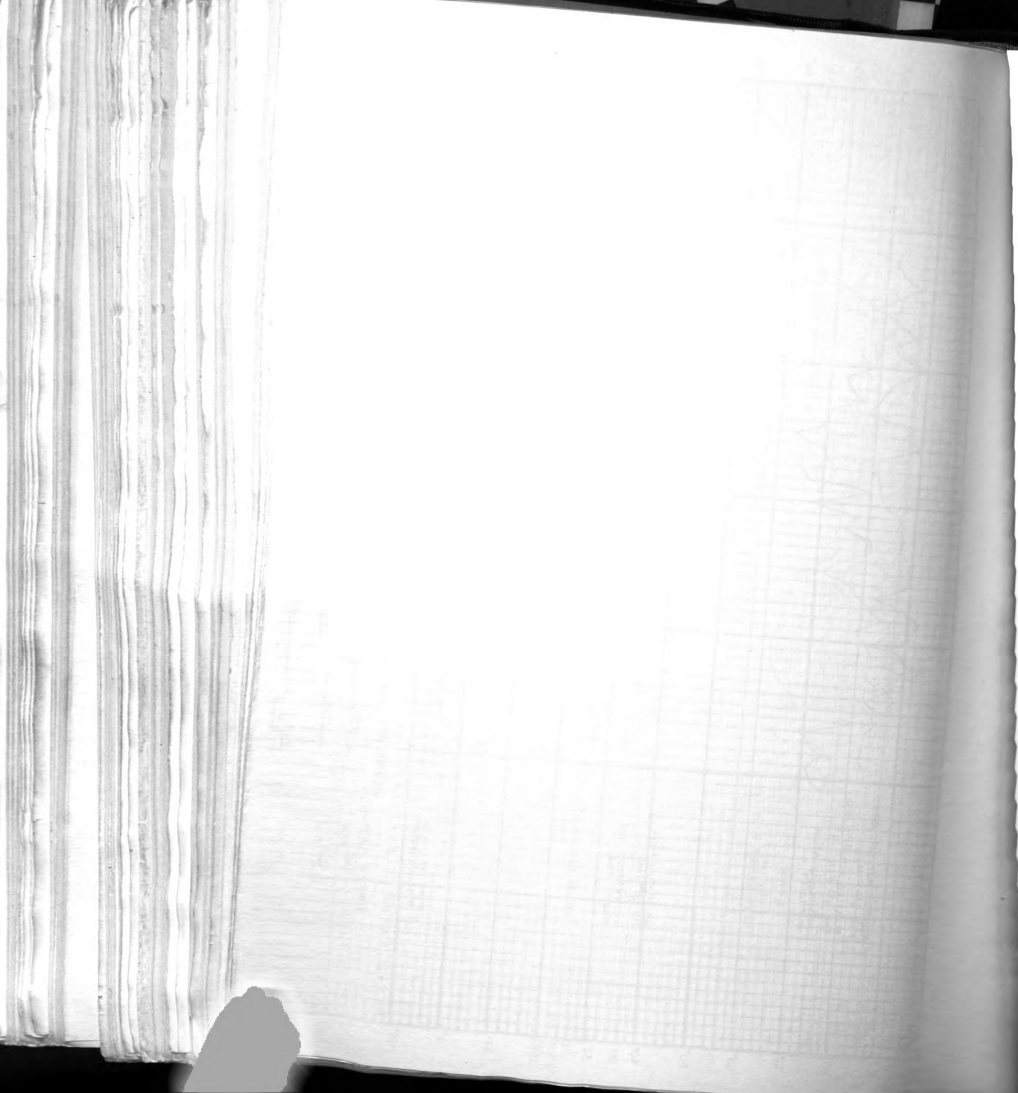


FIG 55c





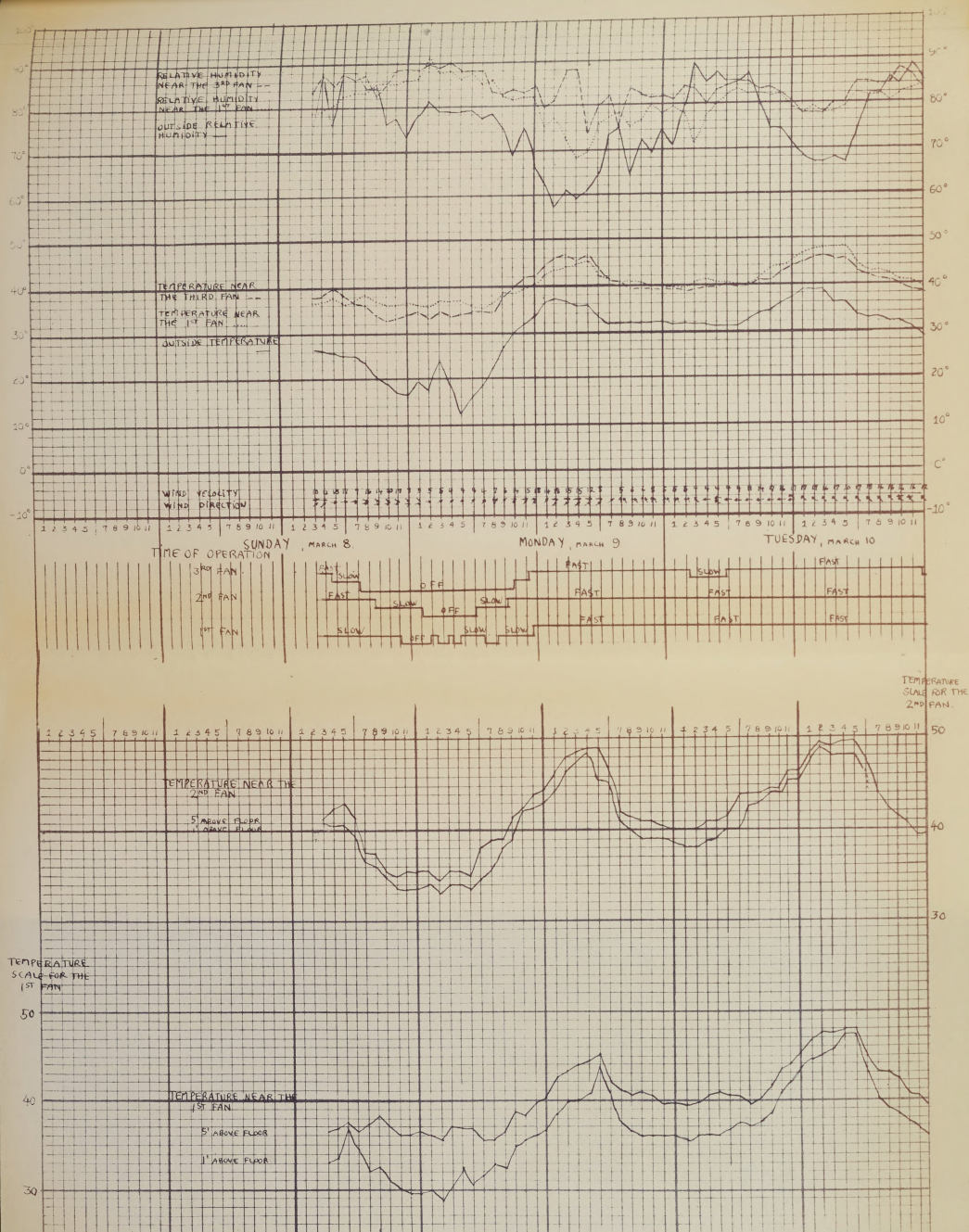
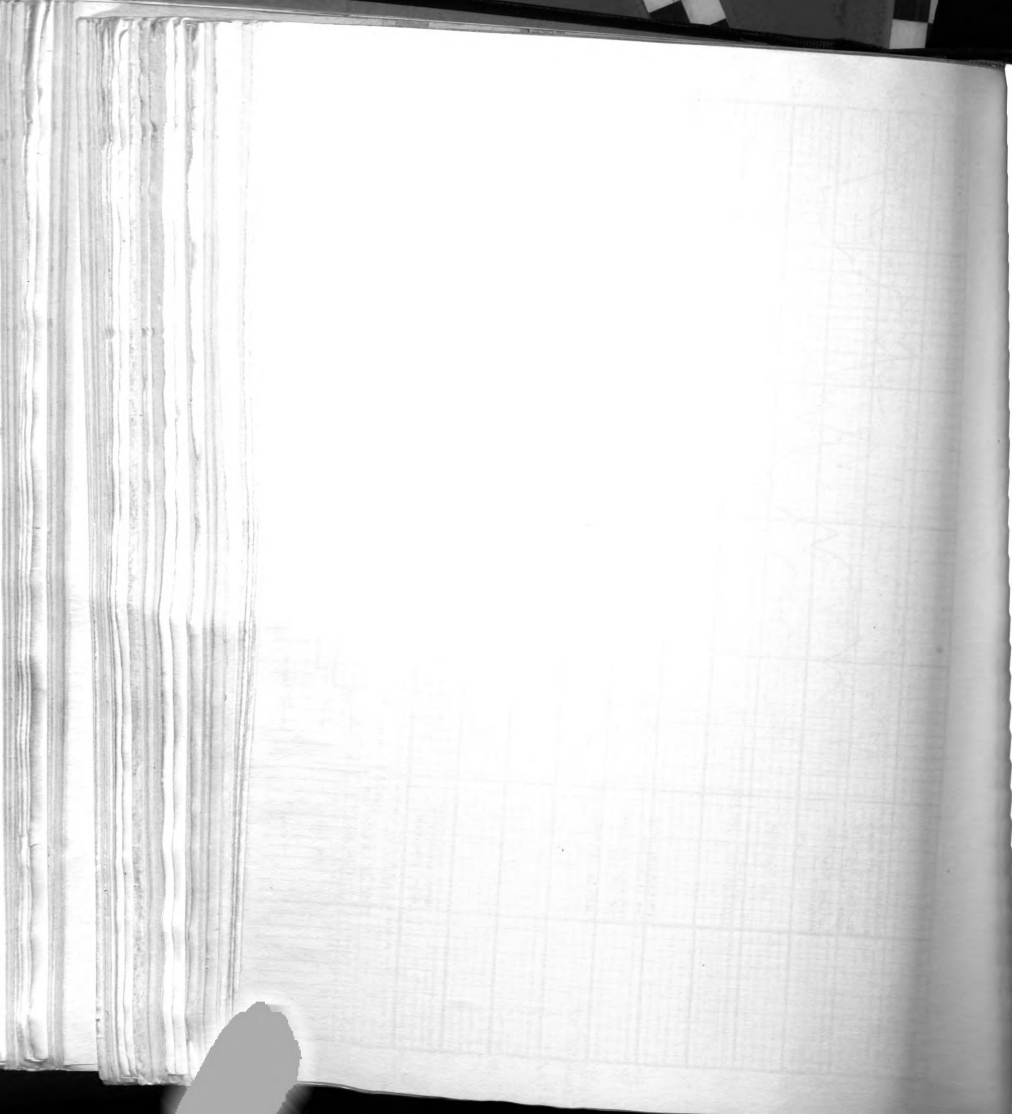


FIG. 55H



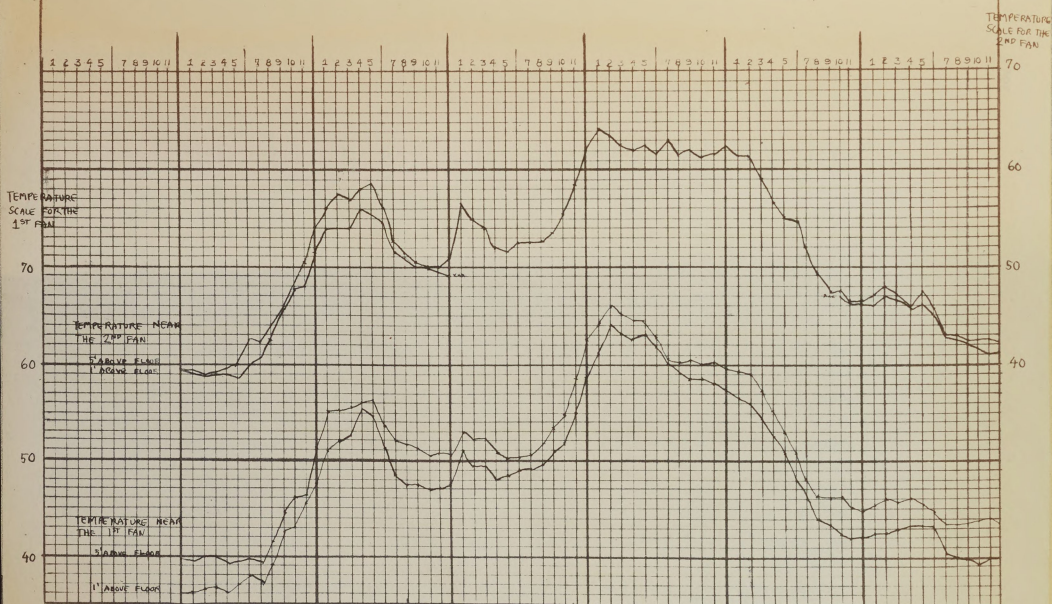


FIG. 55:



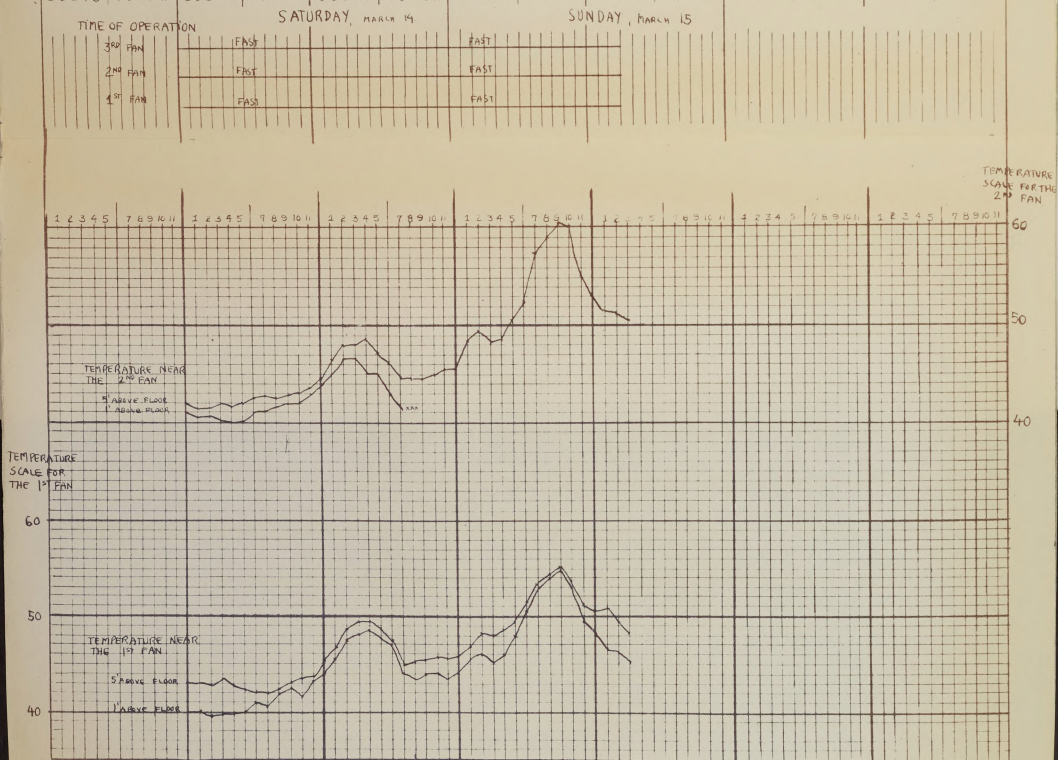
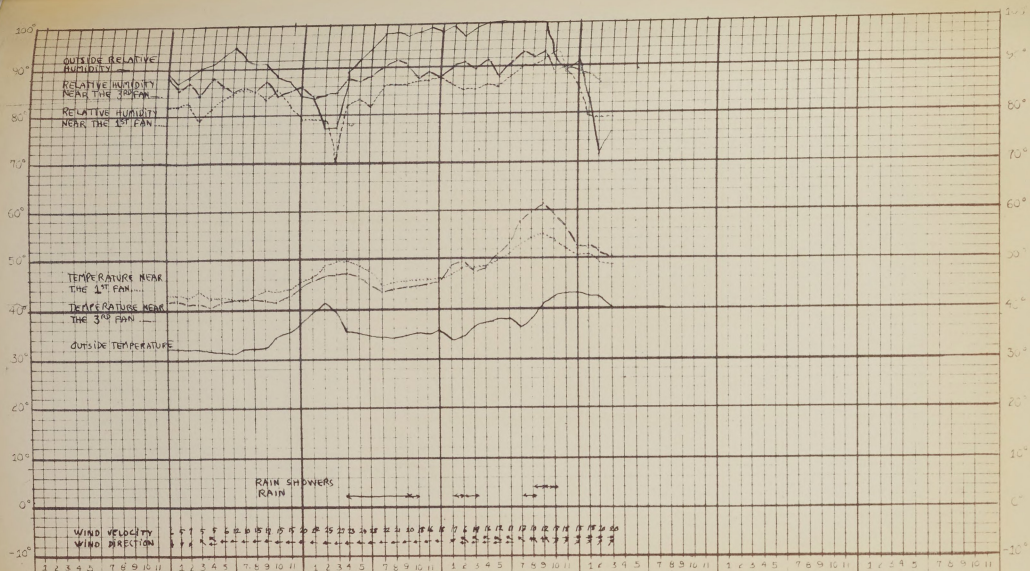


FIG. 55 J

4. Litter moisture content. Litter samples were taken on February the 14th and on February the 28th. The moisture contents obtained are presented on a wet basis, in Table (1).

Table (1). Moisture content of Mr. Joseph Rohrwalter's poultry house litter

Sample	Location	Layer thickness (inches)	Nature of the sample	Appearance (Feb. 14, 1953)	Moisture content (% wet basis)	
					Feb. 14 1953	Feb. 28 1953
No. 1	Center alley, in line with the first fan	3	Litter only	Fairly good condition	42.8	53.77>
		3	Litter only	Somewhat drier	29.2	33.47>
		2	Sandy soil	Quite dry	9.3	9.85>
No. 2	Center alley, in line with the second fan	3	Litter only	Quite wet	46.4	39.7<
		3	Litter only	Somewhat drier	27.6	17.1<
		2	Sandy soil	Quite dry	10.8	7.8<
No. 3	Center alley, in line with the third fan	3	Litter only	Quite wet	52.9	51.6*
		3	Litter only	Somewhat drier	27.6	27.1*
		2	Sandy soil	Quite dry	7.0	6.9
No. 4	Near the front wall (close to the feeders and waterers), in line with the first fan	3	Litter only	Very wet	58.3	55.6<
		3	Litter only	Somewhat drier	49.6	49.9*
		3	Litter only	Quite dry	46.9	46.5*

* Practically no change



Fig. 56. Location No. 1 in the center alley. The top of the litter was in fairly good condition and after analysis was found to contain 42.8% moisture, wet basis. (February 14, 1953).



Fig. 57. Location No. 4, along the front wall. The top of the litter was very wet, containing 58.3% of moisture, wet basis. (February 14, 1953).

c. Litter and soil temperature. A thermocouple (thermocouple No. 15) was placed in the litter in the center alley, in line with the second fan. This thermocouple was at a level three inches below the top of the litter and recorded from February 16 to March 5. The range of variation of the litter temperature at this depth was from 51 to 60°F and varied with the air temperature at floor level in the following manner.

Air temperature at floor level (°F)	Litter temperature 3 inches below the top of the litter (°F)
32-40	51-55
40-50	53-57
50-55	55-60

In addition to the use of this thermocouple a few readings were taken with mercury bulb thermometers, at different depths in the litter and in the soil. The following table (Table 62) shows the increase in litter temperature in relation to the levels at which the temperatures were taken.

Table 62. Temperature of the soil and deep litter of Mr. Joseph Rohrwalter's poultry house*

Depth of the litter or soil (inches)	Air temperature at floor level (°F)	
	43	37
	Temperature of the litter (°F)	
1 1/2	50	47
2	53	54
3	56.5	56
4	61	59
5	67.5	59
6	64	60
7	65	60
	Temperature of the sandy soil (°F)	
1	64	59
3	64	59
5	64	59

* These readings were taken in the center alley, near the second fan.

The temperature readings of the droppings inside the dropping pit were also taken. For an air temperature of 42°F at floor level, the temperature of the droppings was found equal to 49 , $51\frac{1}{2}$, 51 and 51°F at $1\frac{1}{2}$, 3 , $4\frac{1}{2}$ and 5 inches deep respectively.

4. Egg production. The egg production for the period studied (February 11 to March 17), varied from a maximum of 1140 eggs daily (60%) to a maximum of 1488 (77.5%), the average production being 69%. The period of study was too short and the house temperature too variable from one end to the other to permit any attempt of relating the egg production to the house temperature. Such an analysis is especially suited when houses of different conditions are compared.

7. Analysis of results

a. Estimate of the total sensible heat available for purpose of house temperature control. An analysis of the considerable inside and outside temperature data collected was done and the relationship is shown in Figures 48 and 49. Table 62 summarize these values and shows the average outside temperatures and temperature differences corresponding to inside temperatures of 34, 35, 40, 45, 50 and 55°F.

Table 63 . Relationship between inside and outside temperatures

Inside temperature (°F)	Average outside temperature for occurrence of the given inside temperature		Temperature difference (Δt)		Fan operation
	A-For any time of the day	B-For roosting period*	A-For any time of the day	B-For roosting period*	
55	47.5	49	7.5	6	Fans running continuously
50	42	44.5	8	6.5	
45	36	37.5	9	7.5	
42 and 41.25	32 ($t_i = 42$)	32 ($t_i = 41.25$)	10	9.25	
40	29	30	11	10	Fans running intermittently
35	19	20	16	15	
34	16	17	18	17	
34	13	14	21	20	Fans off

* 11:00 P.M. to 6:00 A.M.

An analysis of the fan ventilation rate and outside temperature data was next done and the relationship is shown in Figures 60 and 61. Table 64 summarize these values for inside temperatures of 34, 35, 40, 45, 50 and 55°F.

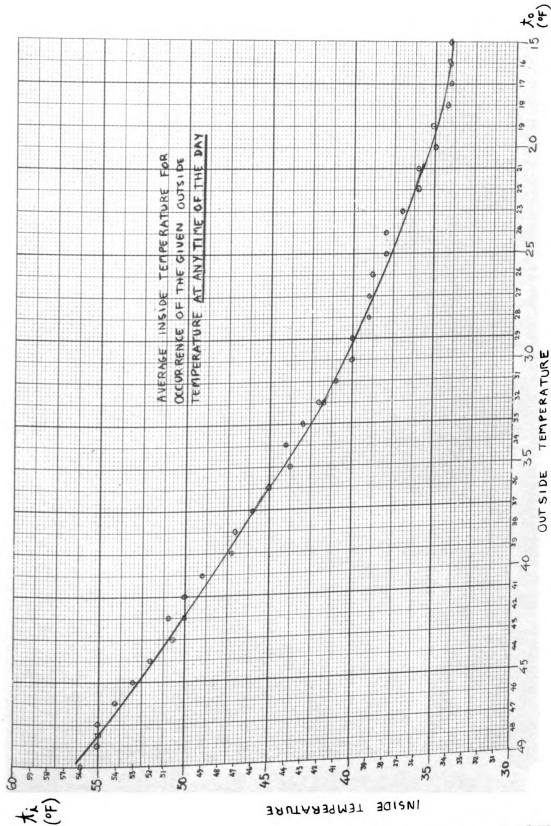


FIGURE 58. RELATION BETWEEN INSIDE AND OUTSIDE TEMPERATURES

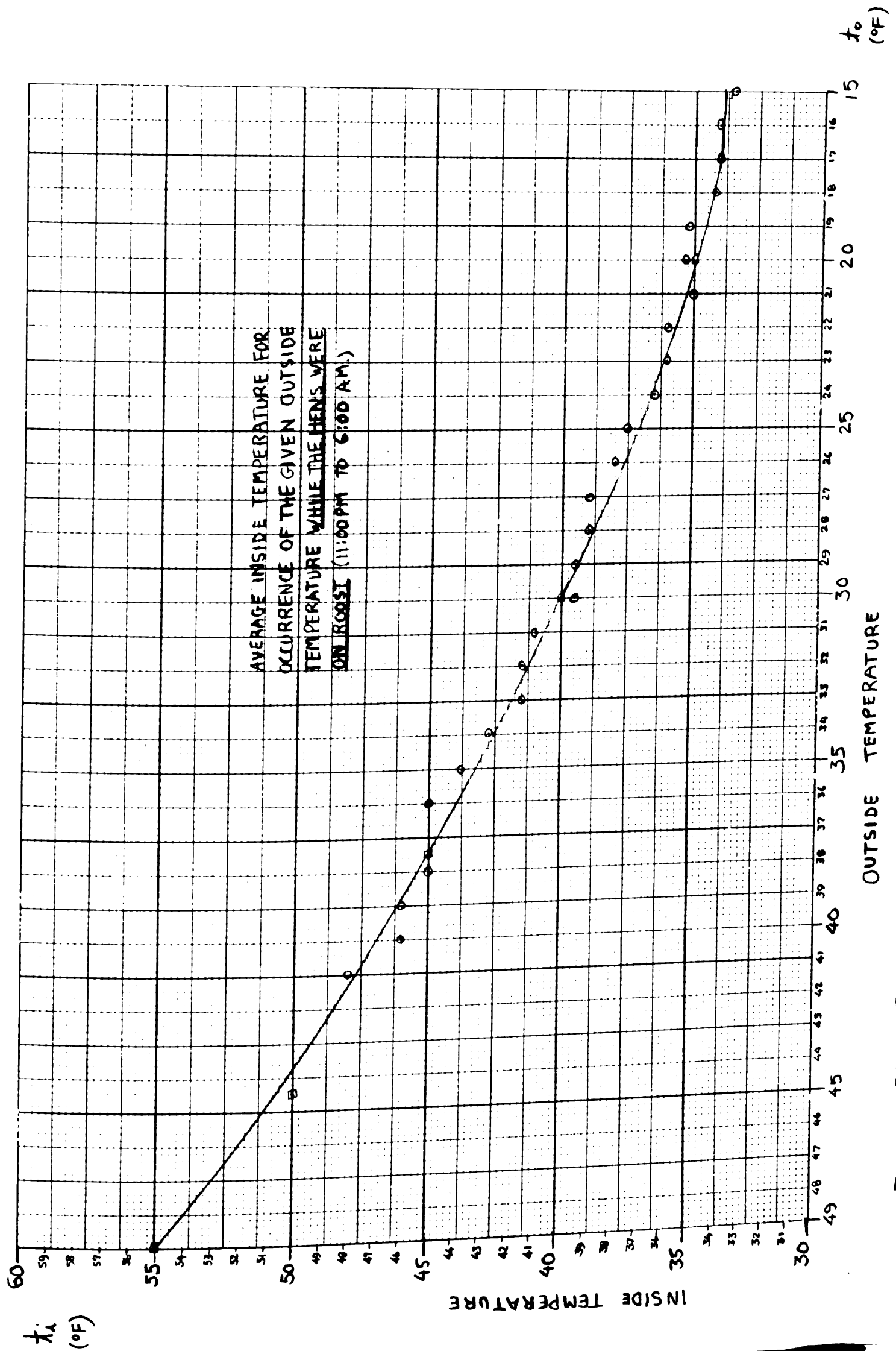


FIGURE 59. RELATION BETWEEN INSIDE AND OUTSIDE TEMPERATURES

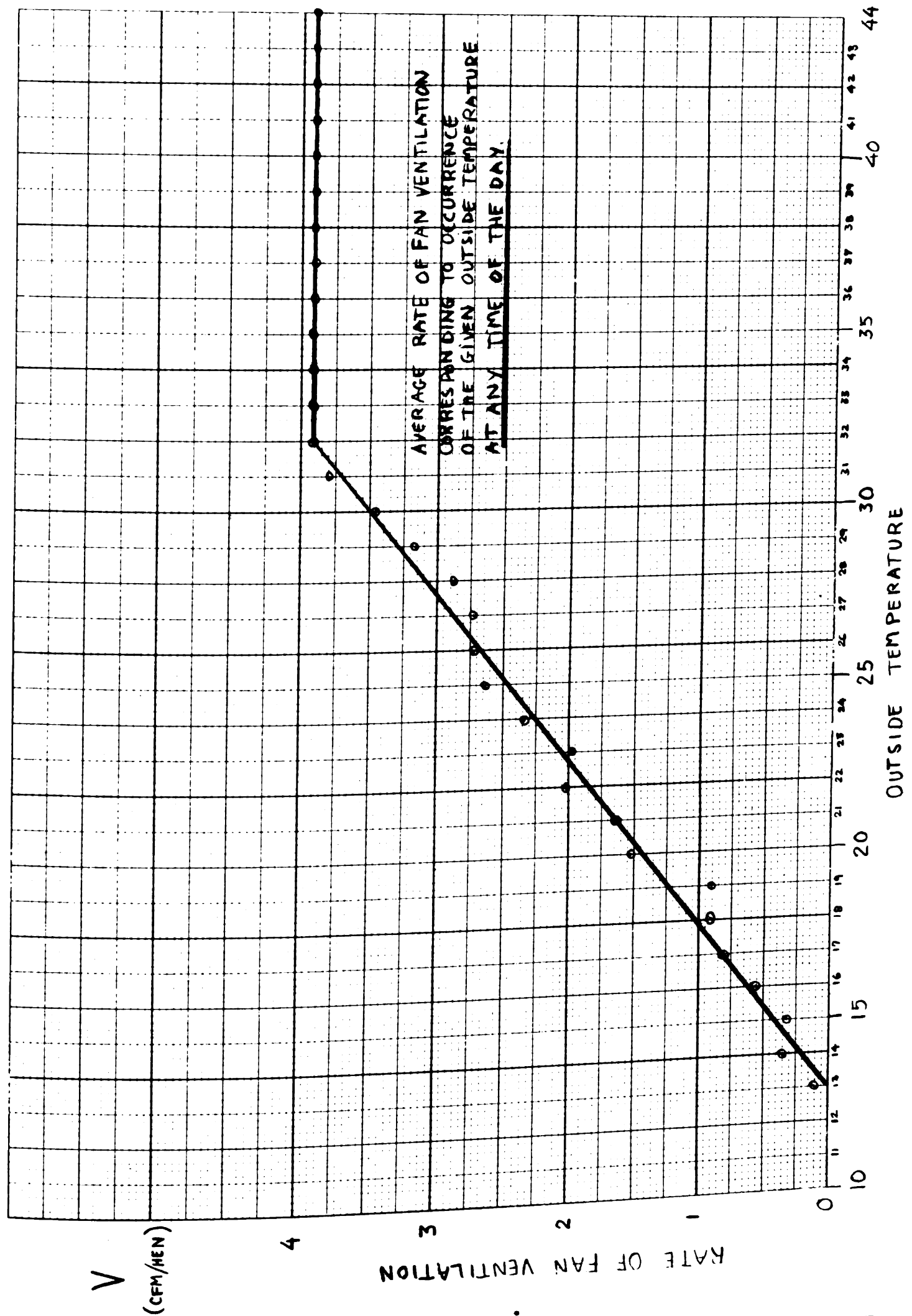


FIGURE 60 . RELATION BETWEEN OUTSIDE TEMPERATURE AND THE RATE OF VENTILATION

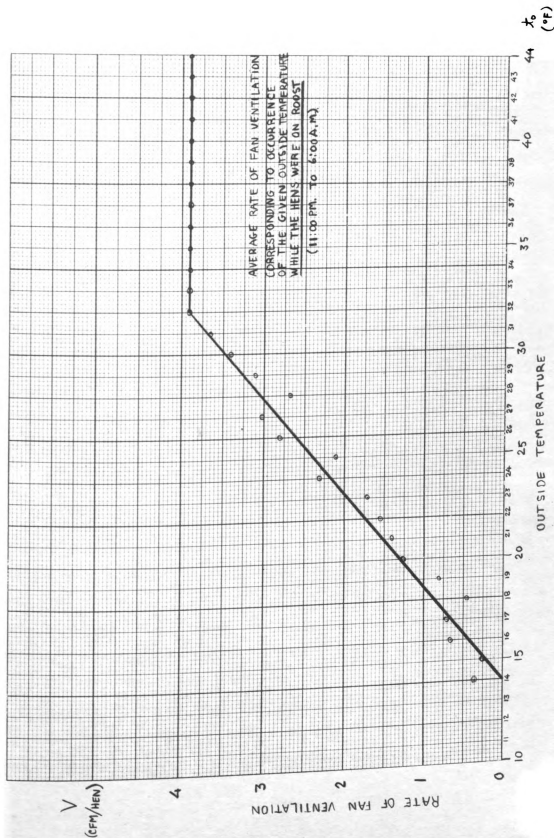


FIGURE 61 RELATION BETWEEN OUTSIDE TEMPERATURE AND THE RATE OF VENTILATION

Table 64. Relationship between inside temperature and rate of ventilation

Inside temperature (°F)	A-Any time of the day			B-Roosting time *			Fan operation
	t_o	Δt	Average rate of fan ventilation for occurrence of the given inside temperature (cfm/hen)	t_o	Δt	Average rate of fan ventilation for occurrence of the given inside temperature (cfm/hen)	
55	47.5	7.5	3.9	49	6	3.9	Fans running continuously
50	42	8	3.9	44.5	6.5	3.9	
45	36	9	3.9	37.5	7.5	3.9	
42	32	10	3.9	-	-	-	
41.25	-	-	-	32	9.5	3.9	
40	29	11	3.4	30	10	3.5	Fans running intermittently
35	19	16	1.2	20	15	1.3	
34	16	18	0.8	17	17	0.5	
34	13	21	0	14	20	0	Fans off

*11:00 P.M. to 6:00 A.M.

From the above table (Table 64) it is seen that as soon as the outside temperature reached 13 or 14°F ($\Delta t = 21$ and 20°F), the fans were shut off all the time and only natural infiltration was providing some air change. It is noted also that for an outside temperature below 32°F ($\Delta t = 10$ and 9.25°F), the operation of the fan was intermittent instead of being continuous.

In the particular poultry house studied, the writer arrived to the conclusion that most of the infiltrating air would be exhausted by the fan, when the fan would be running at high speed for continuous periods, and therefore only a small infiltration rate of 0.1 cfm per hen was considered. This value added to the fan capacity (average of 3.9 cfm per hen against a wind of 10 to 13 miles per hour) gives a maximum ventilation rate of 4.0 cfm per hen.

The temperature difference (Δt), the ventilation rate (V) and the heat loss characteristic of the house (AC) being all known for inside temperatures above 41.25° (roosting time) and 42°F (any time of the day), the total sensible heat available in the house was computed by using slide rules Nos. 9 and 10 and also by using the standard formula $V_s = \frac{v_1}{0.24} \frac{Q_s}{t} AC \frac{1}{60}$. The results are presented in Table 65.

Table 6. Trend in the total sensible heat available for purpose of house temperature control (Fans running continuously)

Time	Inside temper- ature	Outside Temper- ature	diff. (Δt)	V (Fan only)	V (Exfil- tration)	V _{Total}	Q _s (Btu/Hr, hen)	Appr. slide rule	Cor- rected slide rule	True math. solut.
	(°F)	(°F)	(°F)	(Cfm/Hen)	(Cfm/Hen)	(Cfm/Hen)	No. 9	No. 10		
A-Any time of the day	55 50 45 42	47.5 42 36 32	7.5 8 9 10	3.9 3.9 3.9 3.9	0.1 0.1 0.1 0.1	4.0 4.0 4.0 4.0	47.5 50.5 56.5 62.5	47.5 49.1 55.5 62.4	45.6 49.1 55.5 62.4	45.68 49.13 55.73 62.46
B- Roost- ing time	55 50 45 41.25	49 44.5 37.5 32	6 6.5 7.5 9.25	3.9 3.9 3.9 3.9	0.1 0.1 0.1 0.1	4.0 4.0 4.0 4.0	37.7 41.0 47.5 52.5	36.6 39.85 46.35 57.65	34.6 39.85 46.35 57.65	34.55 39.22 46.44 57.78

All answers obtained by use of slide rule No. 10 were found to be within 1% of the true mathematical solution and therefore very satisfactory for computations of this nature.

For temperatures of 40°F and below, computations of the total heat available were difficult since fan ventilation was intermittent at these temperatures and there was no known method to determine the amount of air infiltration. However, the following reasoning was used and a few assumptions made in order to arrive at some values.

Step No. 1 The first assumption made was as follows: "For an inside temperature of 40°F, the difference in total heat available is probably very small as compared to an inside temperature of 42°F (any time of the day) or 41.25°F (roosting time). Therefore, it is probably safe to assume the same amount of total heat available, i.e. 62.46 and 57.78 Btu per hour per hen, respectively."

Step No. 2 The second assumption used may be stated as follows: "Since the total sensible heat available at 40°F is 62.46 Btu per hour per hen (any time of the day) and 57.78 Btu per hour per hen (roosting time), the total sensible heat available at 34°F is probably somewhat higher since the trend is toward higher rate of heat production by the birds and the litter, at the lower temperatures."

Step No. 3 The third assumption consisted in determining the average of air infiltration for periods during which the fans were off continuously. In the particular house studied, the writer arrived to the conclusion that two air changes per hour would seem a very reasonable value. This number of air change corresponds to an exfiltration rate of 1.15 cfm per hen for the house concerned here. Using this value, the total amount of sensible heat production at 34°F was found by use of slide rules Nos. 9 and 10 and checked mathematically by use of the standard formula

$$V_S = \frac{V_i}{0.24} \left(\frac{Q_s - AC}{\Delta t} \right) \frac{1}{60}$$

The mathematical results were respectively 63.43 Btu per hour, per hen (any time of the day) and 60.25 Btu per hour, per hen (roosting time). These values seem reasonable since they indicate that the total sensible heat available to increase somewhat with a decrease in the environmental temperature, as assumed in step No. 2.

Step No. 4 A fourth assumption was finally made by considering the total sensible heat available at 35°F to be the same as the heat total sensible heat at 34°F, i.e. 63.43 Btu per hour, per hen (any time of the day) and 60.25 Btu per hour, per hen (roosting time).

Step No. 5 With the total sensible heat (Q_s), temperature difference (Δt) and heat loss characteristic of the house known (AC), it was simple to find the corresponding total rate of ventilation (V_{Total}) for inside temperatures of 35 and 34°F with intermittent fan ventilation. The average rate of exfiltration was obtained in each case by subtracting the fan ventilation rate (as given by the time of operation instruments) from the total rate of ventilation. The results of all these computations are presented in the following table (Table 66):

Table 66. Trend in total sensible heat available for purpose of house temperature control (Btu/Hr, hen)

t _i	t _o	Δt	V	V	V _{Total}	Q _{sTotal}			Fans operation
			(Fan only)	(Exfil-tration)		Appr. slide rule	Cor-rected s.r.	True math. solut.	
			(Cfm/Hen)	(Cfm/Hen)	(Cfm/Hen)	No 9	No 10		
A-Any time of the day									
55	47.5	7.5	3.9	0.1	4.0	47.5	45.8	45.68	Fans
50	42	8	3.9	0.1	4.0	50.5	49.1	49.0	running
45	36	9	3.9	0.1	4.0	56.5	55.5	55.73	contin-
42	32	10	3.9	0.1	4.0	62.5	62.4	62.46	uously
40	29	11	3.40	0.10	3.50	62.5	62.5	62.46	Fans
35	19	16	1.2	0.80	2.00	63.8	63.8	63.43	running
34	16	18	0.8	0.80	1.60	63.8	63.8	63.43	inter-
									mitten-
34	13	21	0	1.15	1.15	63.8	63.8	63.43	Fans off
B-Roosting time									
55	49	6	3.9	0.1	4.0	37.7	36.6	36.55	Fans
50	44.5	6.5	3.9	0.1	4.0	41.0	39.95	39.92	running
45	37.5	7.5	3.9	0.1	4.0	47.5	46.35	46.44	contin-
41.25	32	9.25	3.9	0.1	4.0	50.5	57.65	57.78	uously
40	30	10	3.50	0.10	3.6	50.5	57.65	57.78	Fans
35	20	15	1.30	0.75	2.05	61.0	61.0	60.25	running
34	16.5	17.5	0.50	1.05	1.55	61.0	61.0	60.25	inter-
									mitten-
34	14	20	0	1.15	1.15	61.0	61.0	60.25	Fans off

The next table (Table 67) was prepared in order to further summarize the total sensible heat available and compare these values.

Table 67. Trend in total sensible heat available for purpose of house temperature control (Btu/Hr, hen)

Environmental temperature (°F)	Total sensible heat available for house temperature control		Difference
	A-Any time of the day	B-Roosting time	
55	45.68	36.55	9.13
50	49.13	39.92	9.21
45	55.73	46.44	9.29
40	62.46*	57.78	4.68
35	63.43*	60.25	3.18

* The writer believes that these two values are typical of roosting time and would have been greater if more samples had been collected of occurrence of the given inside temperature during daytime.

b. Estimate of the amount of net sensible heat coming from secondary sources. The above sensible heat production include not only the sensible heat given off by the birds but also the sensible heat from secondary sources (heat from litter decomposition, solar heat, ground heat, etc.). The next table (Table 6) presents an estimate of the sensible heat coming from secondary sources:

Table 6. Estimate of the amount of sensible heat coming from secondary sources. (Any time of the day, 5-pound hens)

Environmental temperature (°F)	Total sensible heat available under housing conditions (Btu/Hr, hen)**	Net sensible heat found in calorimeter tests * (Btu/Hr, hen)	Net sensible heat from secondary sources (Litter, sun, etc.) (Btu/Hr, hen)
55	15.48	26.0	12.63
50	42.13	29.5	20.63
45	55.73	31.0	24.73
40	62.46	32.5	29.96
35	63.43	-	-

* From Ota, Garver and Ashby calorimeter tests (79)

** From the writer's test

According to these findings, the so-called "secondary sources of heat" seem to be much more important than their name imply and then previously thought.

It is interesting to note that Stapleton and Cox (25) computed the total secondary sources of heat at 48.5°F to be 21.15 Btu per hour per hen, which agree quite closely with the writer's finding of 20.63 Btu per hour per hen at 50°F.

Further research is required in smaller houses with better controlled conditions, in order to verify these findings. The writer believes that the experimental houses built here for solar heat analysis would fit

very well and experiments. Calorimeter tests might also be done with different depths and management of the litter, although the effect of solar heat and ground heat could not be included in such tests.

c. Estimate of the theoretical moisture removal capacity of the fans.

The moisture removal capacity of the fans was analyzed for a total of seventeen days. The days not considered in the analysis were discarded because they did not include complete and reliable data for the full twenty-four hours.

The procedure followed in the analysis consisted of the following steps:

- 1-Use of Carrier's charts AG-246 and AG-198 to find the moisture content of the outside air and that of the inside air near each of the three fans.
- 2-Computation of the moisture carrying capacity of the outside air for each of the three locations (Simple subtraction)
- 3-Computation of the rate of ventilation of each of the three fans for the hour considered (From the time of operation records)
- 4-Use of the writer's slide rule No.8 to find the moisture removal capacity of each fan.

It should be noted that step 1 could be changed by using the writer's slide rule No.7 instead of the Carrier charts. But at the time this study was done, this slide rule was not prepared and, therefore, was not available.

Table 69 shows a comparison of the moisture removal capacity of the three fans considered individually. From this table it is seen that for the seventeen days analyzed the average capacity of the three fans considered as a whole was 17.22 pounds per day per 100 hens while, considered individually, the first, second and third fans were removing 30, 34 and 36% respectively.

Table 69. Comparison of the moisture removal capacity of the three fans considered individually

Date	Outside temperature (°F)		Outside relative humidity (%)		Water removal capacity of the fans (100% efficiency assumed) (lb./Day, 100 hens)			
	Average, daily	Range, daily	Average, daily	Range, daily	First fan	Second fan	Third fan	Average
Febr. 15	24	24-35	71	52-91	32.27	74.00	60.31	55.76
17	19	17-25	66.6	50-79	1.33	24.25	65.10	30.23*
18	26	17-41	65.8	50-93	38.50	33.30	42.20	38.00*
19	34	21-48	73.5	52-91	47.10	59.75	68.35	58.40
20	46	38-53	81	58-95	46.15	42.25	44.45	44.28*
24	34	26-42	78	61-89	38.28	48.05	61.40	49.24
25	31	20-40	79	54-94	40.92	53.30	62.10	52.11
26	35	31-43	84	49-96	29.62	47.40	65.20	47.41
27	31	24-35	81.5	62-97	34.30	48.80	62.18	48.43
March 1	20	11-26	66	47-86	71.90	36.10	15.52	41.17*
2	22	16-27	66.6	55-77	41.10	32.60	0	24.57*
5	21	16-26	74.8	63-84	17.73	40.80	43.35	34.03*
9	23	15-37	71	57-81	42.30	50.20	42.10	44.87*
10	31	30-39	77	67-89	60.19	55.43	52.70	56.11
11	40	29-52	70	46-89	65.30	62.80	60.90	63.00
12	48	40-60	91	73-99	61.20	62.60	65.80	63.20
14	35	31-42	90	77-99	53.70	51.90	50.80	51.90
Average of the 17 days analyzed		lb./Day, 100 hens % of the total			42.51 30%	48.33 34%	50.74 36%	47.22

* Water removal capacity of the fans below 45 lb./Day, 100 hens

The lowest moisture removal capacity recorded was equal to 30.23 pounds per day per 100 hens. The moisture removal capacity was less than 45 for seven days out of seventeen. It was less than 50 for eleven days. The maximum recorded was equal to 63.20 pounds per day per 100 hens and occurred at an average daily outside temperature of 46°F.

The average amount of total water removed by the fans for average daily outside temperatures from 20 to 30, 30 to 40, and 40 to 50°F are shown in Table 70. For the range of 20 to 30°F, the average amount of moisture removed for the seven days included in this range was equal to 38.31 pounds per day per one hundred hens. The amount of water to be removed from a poultry house housing 5-pound birds is around 45 pounds per day (from Table 30). When the fans remove less than this quantity, the

litter had to absorb the remaining water until the environmental conditions bring an increase in the moisture removal capacity of the fans.

Table 70. Average amount of total water removed by the fans, for different ranges of average daily outside temperatures

Range of average daily outside temperatures	Average amount of water removed by the fans (100% efficiency assumed) (lb/day, 100 hens)	Number of days analyzed
20-30	36.31	7
31-40	51.95	7
40-50	58.83	3

Table 71 shows a comparison of the total moisture removal capacity of the fans during morning and afternoon periods. For the seventeen days analyzed, 36.5% of the moisture was removed from midnight to noon and 63.5% was removed from noon to midnight.

The above figures are for the fans only and do not include the moisture carried out by the exfiltrating air at low ventilating rates or when the fans are off. The addition of this moisture removed by the exfiltrating air would have increased the figures shown in Tables 69, 70 and 71. However there was some evidence of fan inefficiency, of air short-circuiting and of relatively still or dead spots in the houses as a result of the poor distribution of the air intakes (the air was coming in nearly exclusively through the windows located on the front wall and through the big west end door). If the air intakes had been located in the back of the house the writer believes that the efficiency of the fans would have been increased and the house conditions would have been drier. Because of the partial efficiency of the ventilation system, the amount of water carried out by the exfiltrating air was not added.

Table 71. Comparison of the total moisture removal capacity of the fans during morning and afternoon periods

Date	Outside temperature (°F)		Outside relative humidity (%)		Water removal capacity of the fans (100% efficiency assumed) (Lb/Day, 100 hens)		
	Average daily	Range, daily	Average, daily	Range, daily	Morning period (Noon to midnight)	Afternoon period (Noon to midnight)	Full day
Febr. 15	24	16-31	71	58-91	23.32	32.44	55.76
17	19	11-25	66.6	50-79	13.83	14.40	28.23*
18	26	11-41	66.8	50-93	5.82	32.18	38.00*
19	21	21-42	73.5	52-91	14.15	42.25	56.40
20	46	38-53	81	58-95	25.56	12.72	38.28*
24	34	20-42	78	61-89	12.40	30.24	42.64
25	31	22-40	79	54-94	12.43	33.68	46.11
26	35	31-43	84	49-96	12.78	28.63	41.41
27	31	24-35	81.5	62-97	20.60	27.83	48.43
March 1	20	11-26	66	47-86	11.82	22.35	34.17
2	22	14-27	66.6	55-77	4.38	20.19	24.57*
5	21	14-26	71.8	63-84	13.23	20.80	34.03*
9	23	15-37	71	57-81	11.58	33.29	44.87*
10	21	30-39	77	67-89	22.48	33.63	56.11
11	40	29-52	70	46-89	21.26	41.74	63.00
12	42	40-60	91	73-99	23.50	39.70	63.20
14	35	31-42	90	77-99	25.10	26.80	51.90
Average of the 17 days analyzed		Lb/Day, 100 hens % of the total			17.26 36.5%	29.96 63.5%	47.22

* Water removal capacity of the fans below 45 lb/Day, 100 hens

Reference to Table 61 shows an increase in the moisture content of the litter near the first fan toward the end of February, while there was a decrease in the moisture content near the second fan and practically no change near the third fan. It should be noted that the area near the first fan was the coldest one and the cold air coming through the big door at this end of the house probably caused some condensation on the litter. With a good distribution of well-built air intakes, it is believe that the conditions in this area would be considerably improved.

reference to Figure 5f on which the temperature and relative humidity data were plotted shows the relative humidity near the third fan to be in general from 7 to 11% higher than that near the first fan, in spite of the fact that the temperature at this location was generally from 2 to 3°F warmer than near the first fan. This is another proof of the needs for a better distribution of the incoming air. Under the actual arrangement, it appears that the air infiltrating through the big door located near the first fan pushes the moist air of this location toward the other end of the building where it accumulates. The lack of air intakes in the back of the house causes a good part of the air to remain still while the air coming in through the front windows is exhausted by the third fan before having been fully effective in its moisture removal capacity.

If the writer were to repeat similar experiments, he would place the wet and dry bulb thermocouples near the fans instead of placing them in the center alley. Such a location would perhaps not be so representative of the condition in the house but would permit a better estimation of the amount of water removed by the fans.

The presence of some condensation was noticed on the walls and ceilings when the temperature of the house dropped below 34°F, especially near the third fan where the relative humidity was the highest. However, it was impossible to take any data on this subject since these low temperatures usually occurred late at night or early in the morning, i.e. at times when the writer was not around. The writer observed the presence of condensation but never in excessive amount. It is very probable that if the winter had been much colder the amount of condensation might have been objectionable and for best results the walls and ceilings would have to be insulated more heavily. A higher insulating value and a better distribution of the incoming air are the two improvements this particular poultry

house require. These improvements should be done in accordance with the recommendations given in the sections dealing with the problem of temperature control and the design of the intake system.

Rates of fan ventilation smaller than the one used in this poultry house (3.9 cfm per hen) would have resulted in a somewhat higher house temperature and a correspondingly higher moisture carrying capacity of the outside air. However, the writer believes that this increase in moisture carrying capacity of the outside air would have been offset by the reduction in the fan capacity especially, with maximum rates as low as 1 or 2 cfm per hen. It is true that the fans would have run a longer time but still the total moisture removed per day would have been smaller and the house conditions resulting from these low rates would not have been so satisfactory. A further argument for the use of ventilating rates from 3 to 4 cfm per hen lies in the fact that when the house temperature raises after a cold spell, it is possible to more rapidly remove any moisture previously accumulated. Fans with ventilation rates of around 4 cfm per hen are also more satisfactory to keep the house cooled, without any manual attention, during the rather warm days encountered in early spring, in late fall and even during some of the winter months.

8. Conclusions

The conclusions derived from this experiment may be summarized as follow:

- 1-The so-called "secondary sources of heat" (heat from litter decomposition, sun, lights, etc) seem to be much more important than their name implies. The net sensible heat coming from these sources, at an environmental temperature of 50°F, was found equal to 22.63 Btu per hour per hen. This agrees very well with the 21.15 Btu found by Stapleton and Cox (95), at 42.5°F.

2-Rates of ventilation from three to four cfm per hen are desirable for moisture removal in houses of moderate insulating value ($AC = 1.70$). With a rate of 3.2 cfm per hen, the average moisture ^{CARRYING} capacity of the fans for the seventeen days of moderate weather analyzed was equal to 47.22 pounds per day per 100 hens, of which 36.56 was removed from midnight to noon and 10.66 from noon to midnight.

3-In adequate design of the air intake system seems to be one of the most important factor in the satisfactory performance of a fan system. It should be remembered that the air exhausted by the fans has done its job by the time it reaches the exhaust point and the incoming air is the one which must be primarily considered if the system is to do the job we are asking from it.

4-The proper insulation of houses is another primary factor to consider if condensation is to be kept to a minimum, if the house temperature is to be maintained above freezing even at design or critical outside temperature, and if the inside air and the litter are to be maintained as dry as possible (air relative humidity below 60% and litter moisture content below 40%, wet basis).

5-The writer's slide rules proved to be very satisfactory in the analysis of experimental data of the kind analyzed here.

6-Intermittent ventilation seems reasonably satisfactory in the house studied (moderate insulation) and would probably have proven more satisfactory if the air intake system had been properly designed.

7-The time of operation instruments devised for this experiment were found to be entirely satisfactory. They permit the grouping of this type of data on the potentiometer chart, next to the dry bulb and wet bulb temperature data.

8-Further research is required in smaller and better controlled houses and in calorimeters of the type used at Beltsville, Maryland, in order to

verify some of the findings obtained in this experiment and in order to determine any of the unknowns related to the problem of poultry house ventilation.

2. Limitations of the test

Period of tests. Longer periods of valid data should have been obtained. With longer periods some insight might have been gained on the influence of outside air temperature below 14°F or above 50°F upon mean poultry house temperatures.

Air intakes. The absence of special box-type air intakes and the fact that a high percentage of the incoming air was through the big 10' x 10' mesh door in the west side of the house made this poultry house unsatisfactory with respect to the proper distribution of air. The above conditions resulted in a large temperature and relative humidity variation between the two ends of the house. Such a lack of uniformity renders all computations subject to error. A poultry house of smaller size and with adequate air intakes should have been chosen. However, there was no such house available in a reasonable distance from the campus. If such a house had been located, it would have been possible to reach more satisfactorily the objectives desired.

Outside relative humidity data. Because of the unreliable character of hygrothermographs when submitted to the wide range of relative humidity encountered outside, it was found more reliable to compute absolute humidity values from the readings of the Lansing Weather Bureau. The basis for the use of these values is explained in the presentation of the results. The writer would like to mention however, that the use of the Lansing Weather Bureau data as empirical and reliable instruments for outside relative humidity should be developed or acquired by the Department if any further study is to be done which involves the use of outside relative humidity.

Air delivery. Air delivery was based on the performance curves published by the Jones Manufacturing Company. It is not easy however, to evaluate the wind effect when making computations of the type required. Gaia (22) states: "There are no known methods of including the effects of wind direction, velocity, and amounts in the analysis of fan ventilation performance." To make computations easier and more reliable, the experimental poultry house should have been as small as practical, cracks should have been eliminated and a type of fan with constant air delivery should have been used.

Location of the house. The poultry house not being on the campus and not belonging to the college, it was not possible for the writer to go as often as desired to make observations on certain factors such as condensation on walls during the coldest periods (which usually occur late at night or early in the morning).

Mild winter. Finally, it should be said that the 1952-1953 was a particularly mild winter.

3. Test of Mr. Albert Lavasseur's Poultry House
(St. Albert, P.C. Canada)

1. The birds

The birds population was composed of 204 Light Sussex hens with an average weight of 5.5 pounds.

2. The house dimensions and arrangement

The pen considered is 32' x 24' and is located on the first story of a two story house 48' x 24'. It has five windows 3' x 4' facing south-east (front) and two small windows facing north-west (back). The nests are on three tiers along the east side wall and dropping pits are along the back wall. Two circular waterers with a constant level valve are installed at the middle of the pen. The floor is made of 2 to 2 1/2 inches of concrete over a sandy soil. The ceiling height is seven feet. The litter is made of wood shavings about a foot deep. Pictures of the house are shown in Figures 42, 43 and 44.



Fig. 62. Mr. Albert Levasseur's poultry house, St. Albert, Province of Quebec, Canada. (Photo furnished by the Shawinigan Water and Power Co., Montreal).

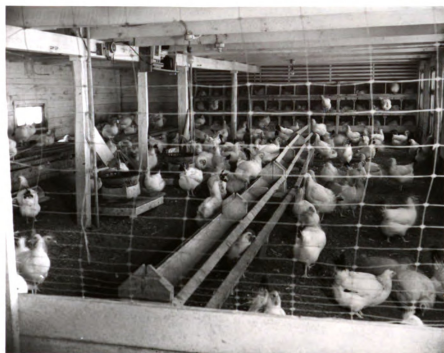


Fig. 63. Inside view of the pen showing nests, dropping pits, waterers, feeders and location of instruments. (Photo furnished by the Shawinigan Water and Power Co., Montreal).

3. The house construction and insulating value

The house construction is as follow:

- 1-Foundation wall = 8 inches concrete, earth bank outside.
- 2-All above foundation = 7/8 inch board, building paper, 4 inches dry clarkins, paper, 7/8 inch board.
- 3-Ceiling = 7/8 inch board, 12 inches deep litter on the second story.
- 4-Windows = single windows.
- 5-Door = 7/8 inch board, paper, 7/8 inch board.

The writer computed the area, heat loss and AU value of each of the construction units. The following table (Table 72) summarize these computations.

Table 72. Summary of the computation for the total heat loss of Mr. Albert Levasseur's poultry house, St. Albert, P.Q. Canada.

Unit	A Radiation area or exposed area (sq. ft)	R Insulation value	AU Heat loss $AU = A \div R$
1-Ceiling	1152	-	0
2-Foundation	144	-	0
3-Wall	744.6	11.5	64.8
4-Windows	104.5	0.90	116.5
5-Doors	14.9	2.7	5.5
Total	864 A total		186.8 AU total

$$C = U \text{ aver.} = \frac{AU \text{ total}}{A \text{ total}} = \frac{186.8}{864} = 0.216$$

$$A = 864 \div 204 \text{ hens} = 4.23 \text{ sq.ft radiation or exposed area per bird}$$

$$AC = 0.216 \times 4.23 = 0.915$$

4. The fan, thermostat and air intakes

A 12 inch single speed fan (1/2 hp, 1725 rpm) manufactured by L.L. Marquette Ltd., Montreal, Canada, is located in the center of the front wall. This fan has a rated capacity of 1072 cfm of air, free delivery. The thermostat is set at 50°F all winter. Four air intakes of 12 inches x 5 inches cross-sectional area are used, which have a distance of three feet between the upper and lower end.

5. Instrumentation

Hygrometers, thermographs, a maximum-minimum thermometer and a time of operation instrument were previously furnished by the Dominion Water and Power Survey (Montreal) and the Quebec Meteorological Station, for use in this test. These instruments are shown in Figures 65 and 66 .



Fig. 64. The 12-inch fan shown delivers more than four cfm of air per hen and the thermostat is set at 50°F most of the time. The deep litter is made of wood shavings and was kept dry and friable all the winter in all parts of the house. (Photo furnished by the Shawinigan Water and Power Co., Montreal).



Fig. 65. The hygrothermograph and the time of operation instrument were placed approximately six feet from the floor or one foot from the ceiling in order to be out of the farmer's way and to avoid roosting of the birds on the shelf where they were placed. (Photo furnished by the Shawinigan Water and Power Co., Montreal).

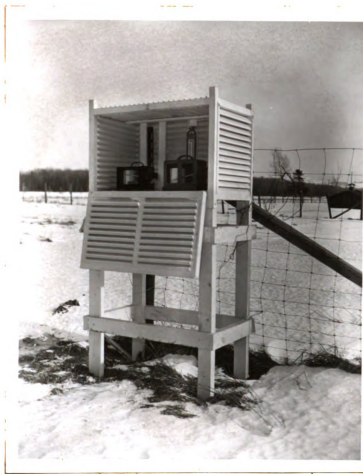


Fig. 66. Outside meteorological shelter with an hygograph, a thermograph and a maximum and minimum thermometer. (Photo furnished by the Shawinigan Water and Power Co., Montreal).

4. Description and discussion of results

The temperature, relative humidity and fan operation data for the few days analyzed are presented in Figure 7.

The relative humidity at the beginning of the experiment (December 16th) was 70% and the inside air temperature close to 50°F, for an outside temperature of 30°F ($\Delta t = 20^\circ$). Although the temperature dropped to -8°F on December the 19th and to -11°F on December the 20th, the temperature remained close to 50°F ($\Delta t = 60^\circ$) and the inside relative humidity increased only to 75 and 80% during these very cold periods. This can be explained by the rather tight and warm construction of the house and the large amount of heat given off by the deep litter.

The litter made of wood shavings was approximately one foot deep. Although the moisture content of the litter was not measured, it remained remarkably dry and friable during the entire experiment (and during all the winter season). The writer took some handfuls of this litter and squeezed them without wetting his hands which shows that the litter moisture content was much below the critical point (40%, wet basis).

No analysis of the amount of sensible heat available was possible because of the scarcity of the available data; the same is true of the moisture removal capacity. However, there is evidence that the system was entirely satisfactory. It is the writer's intention to do more extensive tests in this house and similar ones, next year, in order to find which combination of factors make the litter and inside air so dry and the fan operation so satisfactory.

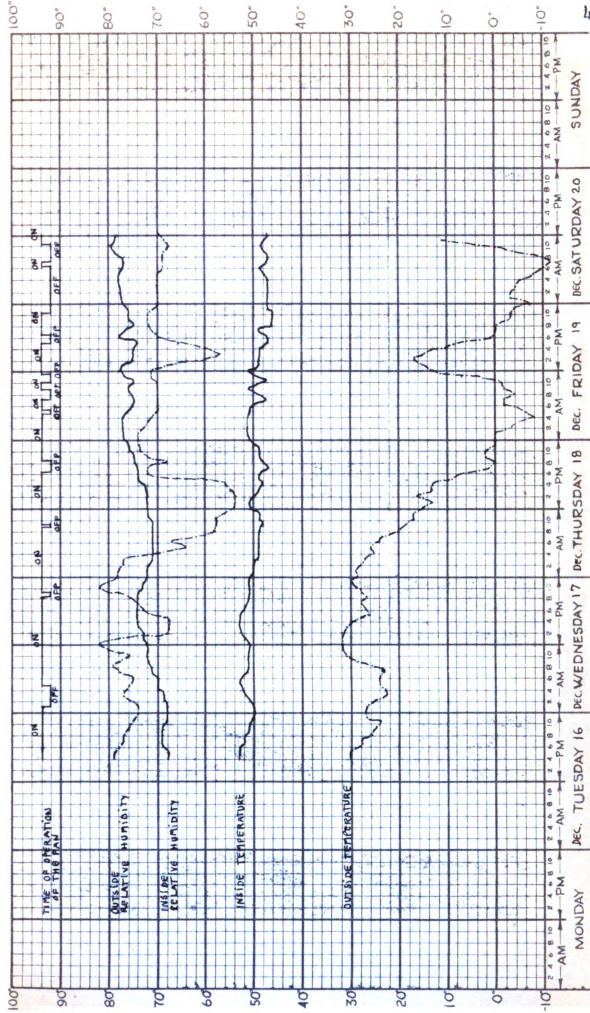


FIGURE 67. TEMPERATURE, RELATIVE HUMIDITY AND TIME OF OPERATION OF THE FAN
(MR. ALBERT LEWASSEUR'S POULTRY HOUSE, ST. ALBERT, P.Q., CANADA)

7. Conclusions

From the little amount of data available on this poultry house, it can be concluded:

1-That the heat given off by deep litter seems to be very important since, added to the heat given off by the birds, it permitted a temperature difference of as much as 60 degrees in a house of AC value equal to 0.92.

2-That dry shavings constitute an economical and very satisfactory insulating material for the walls ($R = 11.5$ in this poultry house) and the saving in heat loss resulting from their use probably accounted for much in the dry conditions observed.

3-That rates of ventilation around four cfm per hen results in extremely good performance in well insulated houses ($AC = 0.92$).

4-That house tightness is an important factor under very cold climate, if the houses must maintain temperatures differences above 50 degrees.

5-That the proper construction and distribution of the air intakes account for much in the obtaining of good fan performance.

6-That intermittent ventilation gives a very good performance in well insulated houses.

8. Limitations of the test

It should be mentioned that this test offered many limitations among which can be listed: the short duration of the test, the nature of the instruments used, etc. However, it gave some valuable preliminary information which the writer intends to use in future tests of the same house. The conclusions of the previous page are qualitatively true even though no quantitative analysis was possible.

B. General conclusions

The general conclusions to the two experiments done in actual poultry houses are:

- 1-The so-called "secondary sources of heat" (heat from litter decomposition, sun, lights, etc) seem to be much more important than their name implies.
- 2-Because of the importance of the heat from litter decomposition at low temperatures (even during roasting time), it looks like if calorimeter heat data might be too low for use in design computations and higher total amount of available heat might be used for this purpose.
- 3-Rates of ventilation around four cfm per hen are desirable in both moderately insulated and well insulated houses.
- 4-In adequate design of the air intake system seem to be one of the most important factor in the satisfactory performance of a fan system.
- 5-The proper insulation and tightness of houses is another primary factor for avoidance of excessive condensation, conservation of heat and maintenance of high temperature differences.
- 6-Dry shavings constitute an economical and very satisfactory insulating material for the walls of poultry houses, allowing high insulating value ($R = 11$ or 12) to be reached at a minimum cost.
- 7-Intermittent ventilation gives a very good performance in both moderately insulated and well insulated houses.
- 8-The writer's slide rule proved to be very satisfactory in the analysis of experimental data involving temperature control or water removal.
- 9-The time of operation instruments devised and used were found to offer great advantages over the commercial types.

12-Further research is required in smaller and better controlled houses and in colonizers of the type used at Beltsville, Maryland, in order to verify some of the findings obtained here and in order to determine many of the unknowns related to the problem of poultry house ventilation.

SUGGESTIONS FOR FURTHER RESEARCH

The writer believes that the following points are worthy of further research.

A. Heat and moisture production in poultry laying houses

- 1-What is the total heat production of laying hens under different environmental temperature (single-hen calorimeters)?
- 2-What is the percentage of latent heat produced by laying birds under different environmental temperatures (single-hen calorimeters)?
- 3-What is the total heat production in poultry houses under actual housing conditions (small houses for solar heat study)?
- 4-What is the total net sensible heat coming from secondary source under actual housing conditions (small houses for solar heat study)?
- 5-What is the relationship between environmental temperatures and the percentage of moisture content of the fresh droppings of laying hens (single-hen calorimeters)?
- 6-What is the relationship between environmental temperatures and the total water consumed and eliminated by laying hens (ten-hen calorimeters and small houses for solar heat study)?
- 7-What is the relationship between environmental temperatures and the feed consumption of laying hens (ten-hen calorimeters and small houses for solar heat study)?
- 8-What is the relationship between the weight of laying hens and their total heat production (single-hen calorimeters)?
- 9-What is the relationship between environmental temperatures and the heat produced by deep litter of different depth and different management (ten-hen calorimeters and small houses for solar heat study)?
- 10-What is the relationship between environmental temperatures and the temperature of deep litter of different depth and different management (ten-hen calorimeters and small houses for solar heat study)?
- 11-What is the relationship between environmental temperatures and the temperature of concrete floor and earth floor in poultry houses (small poultry houses directly on the ground)?
- 12-What is the relationship between environmental temperatures and the heat gain or heat loss from the earth under the poultry house litter or floor (small poultry houses directly on the ground)?

D. The problem of temperature control in laying houses

- 1-How do heat exchangers perform in Michigan poultry laying houses?
- 2-How do heat exchangers compare with insulation, in Michigan laying houses, with respect to economy?
- 3-What are the minimum insulating values required in Michigan for houses of different bird population and different exposed (or radiating) areas?
- 4-What are the economical advantages of insulation (amount of saving in feed consumption, amount of increase in egg production, amount of saving in labor required for egg cleaning and litter changing) (ten-hen calorimeters and small houses for solar heat study)?
- 5-What are the environmental temperatures desired by Michigan poultrymen (survey)?
- 6-What is the most economical and the safest outside design temperature for Michigan animal shelters?
- 7-How does fan ventilation compare with the different types of natural ventilation, with respect to temperature control (small houses for solar heat study)?

C. The problem of condensation on walls and ceiling surfaces

- 1-What is the effect of recirculating fans on the critical relative humidity for condensation?
- 2-What is the effect of a curtain of cold inlet air on the critical relative humidity for condensation?
- 3-How harmful is condensation on walls and ceiling surfaces with respect to the life and maintenance of poultry laying houses?

D. The problem of condensation within walls and ceilings

- 1-How do poultry houses without vapor barriers compare with poultry houses having vapor barriers, with respect to temperature control, moisture removal, condensation inside the walls, duration of the building, etc?
- 2-How much does condensation within walls affect the heat loss characteristics of the insulating material and the amount of heat loss through the walls of laying houses?

E. The wet litter problem

- 1a-What is the relative importance of the different factors producing dry litters (insulation, ventilation, type of litter, depth of litter, amount of stirring, use of recirculating fans, use of heat exchanger, increase in bird population, frequent cleaning of the dropping boards, addition of new litter, addition of lime, etc.)?
- 1b-What are the most important single factors for the maintenance of dry litters?

1c-What are the single factors absolutely required for the maintenance of dry litters?

2-How much beneficial to litter drying is the feeding of grains directly in the litter in order to encourage stirring of the litter by the birds?

3-Should regular cleaning (bimonthly or monthly) of the dropping pits be preferable to the actual practice of letting the droppings piled through all the season?

4-How efficient is the use of mechanical litter stirrers for the maintenance of dry litters?

5-How efficient is the use of ground limestone for the maintenance of dry litters?

F. Design of the intake system

1-What is the effect of increasing the total air intake area on the temperature control characteristics of the house and the chances of having some intakes act as outtakes?

2-How much infiltration occurs through L-type air intakes and the many other types of air intakes when the fans are off and winds of different speeds are blowing on the walls of the building?

3-Are hinged or sliding door really necessary on the top of air intakes? Are they really effective in decreasing air flow during cold spells?

4-How do special air intakes compare with cracks with respect to temperature control, moisture removal, condensation, etc?

5-How much exfiltration occurs in a poultry house with rates of fan ventilation from four to zero cfm?

G. Design of the exhaust system

1-Is the fan duct really helpful and necessary in poultry laying houses?

2-What is the best location for fans in poultry laying house?

3-How does continuous ventilation compare with intermittent ventilation with respect to moisture removal, avoidance of condensation on walls and ceilings and maintenance of dry litter, in houses of different insulating values?

4-How does intermittent one volume delivery fans compare with intermittent two volume delivery fans (dual-speed motor or automatic operated shutter in the duct) with respect to moisture removal and temperature control?

5-How does ceiling fan system (combined with gravity system flue) compare with wall fan system with respect to cost of operation?

6-Are recirculating fans really helpful and worth their price?

7-What are the best thermostat setting for houses of different insulating values?

8-How does maximum fan capacities of 1, 2, 3 and 4 cfm compare with respect to moisture removal and temperature control?

9-How does the performance of the second floor and first floor of a two-story poultry house compare, when using one fan only with the duct extending to the first floor and one opening provided in the duct at each floor?

10-How does the size of intake area (manual control on intakes' opening) affect the performance of the fans?

H. Instrumentation

1-Is it possible to develop a simple cheap and reliable wet bulb instrument for outside conditions, utilizing a low temperature of freezing liquid instead of water as used in the Henderson wet bulb instrument?

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