AN INVESTIGATION OF

THE SUITABILITY OF STANDARD MECHANICAL REFRIGERATING APPARATUS

FOR MODERN OFFICE AIR CONDITIONING

by

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Table of Contents

I. Introduction

Modern business has attained its present high degree of efficiency only though man's constant efforts to introduce improved methods and machines which have increased our productive ability and made possible the attainment of success with a minimum of effort and lost time. To the office man, competent management has profitably increased his efficiency through the introduction of adding machines, accounting machines, cross-index files and numerous other labor saving devices. But what has business done to stop the enormous losses caused by improper indoor atmospheric conditions which fatigue the individual to a state where his mental and physical efficiency is lowered? It has been estimated that a single heat wave has cost New York as much as five million dollars in one day through only personal efficiency losses of from 15 to 30 percent. The advent of colds and sickness produce similar results in the winter. Therefore, our present treatment is such that instead of the air furnishing health and vitality that nature intended, it actually produces fatigue.

Comfort and health go hand in hand. Statistics show that only a proper combination of humidity, cooling, clean and constantly moving air is productive of health. Therefore, to have a comfortable office in an effort to promote the highest efficiency, these elements must be produced artificially. The efforts of this investigation have been centered around these four elements throughout the design and manufacture of the Humidraft machine, upon which this thesis is based.

Now that we have become familiar with the reason for this investigation, let's consider a few fundamentals essential to the thorough understanding of the problem. Air possesses many interesting properties, some of which are shown in Appendix II, Table 6. These properties must be accepted as facts and not the result of superb intelligence or reason on the part of man.

It is important in a study of this nature to remember that one B.t.u. will raise 55 cubic feet of air one degree fahrenheit. Also that the weight of air per unit volume decreases with increasing temperature. Frequently, however, we say that for approximate results 13.5 cubic feet of air weighs one pound.

The ordinary thermometer ignores all moisture in the air and measures only dry heat. It is referred to as the Dry Bulb thermometer.

The Wet Bulb thermometer is a Dry Bulb thermometer with a moist cloth on it. The evaporation of moisture from this cloth gives a temperature corresponding to the Wet Bulb temperature. This represents the Total Heat in the air or Wet Heat plus Dry Heat.

Relative Humidity indicates the moisture content of the air. The air will hold a certain definite amount of moisture. If it contains only half that amount, the Relative Humidity is fifty percent.

Dew Point and Saturation temperature are synonymous. When air is filled with water vapor it is at its Dew Point. If you introduce more moisture, the air won't take it. Saturated air indicates a Relative Humidity of one hundred percent, or that the air is at its Dew Point.

Table 6, Appendix II, gives the relative proportions of the above four elements for various temperatures. It is imperative, therefore, that intelligent use be made of this table. Similar results may be obtained, however, from a psychrometric chart.

II. Review of Literature

The possibility of using a refrigerator to cool offices was first suggested by Lord Kelvin in a paper entitled "Economy of Heating and Cooling Buildings by Means of Air Currents," presented before the Royal Society in 1852.

A. R. Stevenson, Jr., F. H. Faust, and E. W. Roessler of the General Electric Company have recently written a noteworthy article on the "Application of Refrigeration to Heating and Cooling of Homes." Of particular importance is their explanation of the theoretical justification of the phenomenon of a refrigerator absorbing an amount of heat greater than the heat equivalent of the applied power. Explanation of this fact being established on the basis that a refrigerator may be regarded as a reversed Carnot engine. This article also discusses many reasons why the method of heating as suggested by Lord Kelvin has not been in commercial use ever since it has been known and recognized in the days of Lord Kelvin. Some reasons worthy of special attention are:

1. The cost of electricity in most localities was too high for this method to compare anywheres favorable with existing methods.

2. Very little was known of the actual operating costs of a refrigerating heating system, because of the special nature of the refrigeration and auxiliary equipment needed, and the careful study of climatic conditions required to predict their performance.

3. Of utmost importance is the fact that, people were not educated to the needs of air conditioning and indicative of the

supply is always the demand.

For the past year this country has experienced a revolution in the art of air conditioning, for new ideas and improvements have developed which will far outclass the development of the streamlined automobile. Typical of this parodoxical statement is the results of a survey just completed by the air conditioning department of the commonwealth Edison Company, in Chicago proper. This survey showed a total capacity of nearly 19,000 tons installed for air conditioning a variety of commercial buildings.

Such installations as that recently installed in the new R. C. A. Building at Rockefeller Center, New York, containing over 2,000,000 square feet of rental area which has been completely air conditioned, exemplifies not only the present day demand, but achievement.

Mr. F. F. Groseclose in his recent article, "Air Conditioning and You," published in the Virginia Tech Engineer, March 1934, states that it is by no means inconceivable to believe that the automobile of 1935 will possess air conditioning equipment.

Last February the Norfolk and Western Railroad Company spent approximately a million dollars for 43 new type cars to be completely air conditioned. The date of delivery having been set for August 1934.

Thus we see the idea of air conditioning being a luxury is quickly fading out of existance. Present achievement demands that we be modern and think of air conditioning not as a luxury but as one of the necessities of life.

III. Investigation

A. Object of Investigation

The object of this investigation is to endeavor to show that by combining modern mechanical refrigerating apparatus with parts of the present type vapor radiation heating system there will result a practical and efficient means of air conditioning.

The factors comprising the principles of air conditioning that have been considered in this investigation are:

1. Humidity control
2. Cooling
3. Air Motion and Cleanliness

Air absorbs moisture very easily and quickly. To make it moist all that is necessary is to put a fine spray of water in the air.

Extracting moisture from the air is also a very simple operation, but usually a much more expensive one. When moisture laden air is cooled by a refrigerating machine the moisture condenses and if passed over eliminators having an angle of deflection, it collects on the eliminators and can be drawn off.

The above procedure produces a scrubbing effect and will also wash the air as well as cool and dry it.

The air may be cooled by a combination of two different methods. First, when the same water is recirculated, the air is cooled by evaporation; that is, the heat necessary to evaporate the water is extracted from the air. Obviously this produces the greatest cooling on the hottest days, giving a temperature drop to the air ranging from ten to twenty degrees which depends upon the humidity of the air entering. Second, the use of refrigeration or cold water in sufficient quantities will allow the air to be cooled and moisture condensed from it, thus lowering the humidity.

The spray of water through which the air must pass absorbs all foreign dust particles thereby cleansing and purifying it.

It is necessary to have a fan or blower circulating the air through the machine at varying speeds to adequately take care of the fluctuating demand in load (B.t.u. per hour).

In an effort to combine all of the above principles into as simple and efficient an operation as possible the author has completely designed and built a machine in which he takes considerable pride since it has proven successful from many standpoints.

B. Method of Procedure

Upon completion of the humidifier and overhauling the compressor the two machines were combined into a single unit, after which the compressor was charged with approximately three pounds of methyl chloride, the refrigerant used in this investigation. Tests were then made in an effort to determine the efficiency or degree of satisfaction of the unit. A general outline of which was as follows:

1. Several preliminary test runs were made by first observing initial room conditions and then allowing the unit to run for a short while, after which observations were made to determine improvements, if any. The object of this being to determine the efficiency of the fan, eliminators, washer and cooling coils.

2. The required cooling load for an average summer day in Blacksburg, Virginia, was determined.

3. The cooling load was converted to cubic feet per minute.

4. The unit was then adjusted to take care of such conditions.

The wet and dry bulb temperature readings indicating the conditions of the office were observed and from the "air properties table" given in Appendix II, Table 6, the percent relative humidity was found to be 71.6%. This value was obviously too high, for experience indicates that a humidity of approximately 50% is ideal for human comfort. This fact being established, it was desired to determine the effect of using the fan to pass the air over the baffles or eliminators only. The cooling coils were not used for this test.

After a run of approximately two hours the improved room conditions were again observed. This time the relative humidity was 62.6%, thus indicating a fan and eliminator capacity of 9%(1).

The compressor was then started and the expansion valve so regulated as to give the desired temperature in the expansion coils for lowering the humidity to as near 50% as possible. For those days having a relative humidity below 50% the water pump was started and a spray of water necessary to compensate for this differential in humidity was obtained.

The required cooling load was next determined for the following conditions:

	Dry Bulb Temp. °F	Wet Bulb Temp. °F	Dew Point Temp. °F	Relative Humidity %
Outside	85	67	57.5	38.6
Inside	76	64	57	51

In selecting the desired inside conditions consideration was given to the fact that for average conditions a differential of from 9 to 12 degrees is enough for offices. General practice also indicates 69 grains(2) of moisture per pound of dry air for best conditions. A table giving these values is shown in Appendix II, Table 1.

(1) This value does not represent the maximum capacity of the eliminators.
(2) 1 grain = 0.002286 ounces. 69 grains = 0.157734 ounces.

Using such coefficients of heat transmission as prescribed by The Trane Company, La Crosse, Wisconsin, an entire heat load was calculated (Appendix II, Table 3). Air entered the room only through infiltration. The amount of heat given off was separated into a sensible heat load and a latent heat load. To compute the sensible load the following formula was used:

$$\frac{\text{Cu. ft. per hr.}}{13.5} \times (DB_1 - DB_2) = \text{B.t.u. per hour}$$

where:

DB_1 = the heat in the dry air at dry bulb temperature on the outside

DB_2 = the heat in the dry air at dry bulb temperature on the inside.

13.5 cu. ft. of air = 1 pound (approximately)

The latent heat load was figured in exactly the same way except that the B.t.u. content at the outside dew point (LH_1) was subtracted from the B.t.u. content at the inside dew point (LH_2). This formula thus becoming:

$$\frac{\text{cu. ft. per hour}}{13.5} \times (LH_1 - LH_2) = \text{B.t.u. per hour}$$

The procedure for determining the remaining factors comprising the necessary heat load (Appendix II, Table 5) is self explanatory.

Now knowing the required load the next step was to convert this load (B.t.u. per hour) into cu. ft. per minute to be circulated by the fan. The following simple formula will indicate this step very clearly.

$$\frac{\text{B.t.u. per hour x 55}}{\text{60 x air temp. change}} = \text{cu. ft. per minute}$$

where:

55 = cubic feet that 1 B.t.u. will raise 1^{o} fahrenheit.

60 = minutes in one hour.

The unit was then adjusted in an effort to take care of this load. The condition of the refrigerant entering the expansion valve was as follows:

A Liquid at

Pressure = 30 pounds per sq. in. Abs.

Temperature = 20^{o} fahrenheit

The total heat corresponding to the above conditions was minus 4 B.t.u. per pound(3).

The refrigerant upon leaving the expansion coils was a superheated gas with

Pressure = 30 pounds per sq. in. Abs.

Temperature = 32^{o} fahrenheit

Such conditions indicated a total heat content of 175 B.t.u. per pound. The heat absorbed by the cooling unit was therefore 175 B.t.u. per pound. (The difference of heat content between initial and final conditions.) The capacity of the compressor being 350 pounds in 24 hours or 14.6 pounds per hour, the total heat absorbed by the cooling unit was necessarily 2613.4 B.t.u. per hour. This value obviously being too low to take care of the required load, the initial conditions

(3) This value was obtained from P - I (Pressure - total heat) chart for methyl chloride shown in "Handbook of Mechanical Refrigeration" by Macintire. Similar charts may be found elsewhere.

of the refrigerant entering the cooling coil were altered in an effort to increase the capacity of the unit and better adapt it to the required conditions.

C. Description of Apparatus

All experimental data and results were obtained through the Humidraft machine designed and built solely by the author.

A Servel Model 18-A electric refrigerating machine was used to cool the cooling coil over which the air was forced to pass. This machine consisted of a compressor with a rated capacity of 350# in 24 hours, a condenser, float valve and pressure control switch. The compressor was driven by a General Electric A.C. Motor model No. 30044, type RSA, 110 volts, 5.9 amps., 1/3 H.P., 1725 r.p.m., 60-cycle, and single phase.

The coils were manufactured by Shultz and James Company, Incorporated, Richmond, Virginia, and were originally intended to be used as a vapor radiation heating unit with an exposed surface of approximately ten square feet.

Methyl Chloride gas was the refrigerant used in this test.

A rotary oil pump was somewhat modified to warrant its use as a water pump. Constant circulation and an even spray of water being its important qualifications. This pump is run by a 1/4 H.P. General Electric Alternating Current motor, with a speed reduction of 45 to 1. The speed of the pump is 38.3 R.P.M. with a head of approximately 25 feet of water.

The water is discharged from 3/8" copper tubing through 1/64" holes, evenly spaced.

After passing over the eliminators the water is deflected to a trough through which it is drained back to the reservoir or settling tank.

The settling tank which is placed under the entire washer and eliminator is 4" high, 5" wide and 6" long. There is a brass wire cloth strainer through which the water passes before entering the suction of the pump. This eliminates clogging of the pump and pipes and insures a discharge of pure and clean water. A one-half inch fresh water supply main is connected to the suction pipe of the pump. A drain pipe is connected to the reservoir.

The washing surface consists of six eliminators or strips of number 40 galvanized tin 1-1/8 inches wide and 30 inches long. These eliminators have an approximate angle of deflection of 60° with spacing of plates only 1/2 an inch apart.

Constant circulation of air is maintained through a 5.85 inch electric fan. At the maximum speed of 1750 R.P.M. 1758.3 cubic feet of air are circulated through the cooling coils and over the eliminators per minute. This is a variable speed fan making it adaptable to take care of the fluctuating demand in load. Another striking feature of this fan is the ease with which the direction of air flow over the cooling coils may be changed. The lever used in raising or lowering the fan may be clearly seen in Appendix III, View No. 2.

The two cylinder compressor is of the vertical reciprocating type and is lubricated by the splash system from oil in the compressor base. The inlet valves in the piston heads are automatically operated. The exhaust valves are of the flat spring type, also operated automatically.

In order to maintain a pressure high enough to condense or liquefy the gas in the condenser and at the same time maintain a pressure low enough to evaporate the liquid in the expansion coil there must be some gate, or valve between these two extreme conditions. This is the function of the float valve, for this valve not only holds back the pressure of the compressed gas, but automatically opens at the proper time to let the condensed methyl chloride into the expansion coils, in order that refrigeration will not be interrupted.

The pressure control switch is mounted on the base of the machine behind the compressor. This pressure control is connected into the tubing which carries the gas from the expansion coil to the compressor and the switch is wired in series with the electric motor. The opening and closing of the switch is controlled by a plunger attached to a bellows type sylphon. This sylphon being connected in the suction line from the chilling unit to compressor must always be under the same pressure as the chilling unit. By having this switch set to open and close between two definite pressure limits a fairly uniform temperature in the chilling unit may be maintained. The pressure limits used in this investigation were 7 pounds per square inch and 18 pounds per square inch (gage).

The condenser consists of 105 feet of 3/8" copper tubing and is air cooled. The air is kept in circulation by a fan operated by the compressor belt. Shut-off valves are located at each end of the condenser to permit all of the methyl chloride to be pumped into the condenser and held there while removing or working on any other part of the machine.

D. Materials

Methyl chloride (CH_3Cl), the refrigerant used in the Servel compressor to cool the coils over which the air was forced by means of a fan resembles Ethyl Chloride. However, the unit pressure are greater and the piston displacement per ton of refrigeration is less. The gas is inflammable and slightly anaesthetic when more than 10% by volume is in the air, but not unpleasant to inhale.

The latent heat of vaporization is about 120 B.t.u. per pound. It is not corrosive to copper or alloys of copper, nor to iron or steel. The pressures generally encountered are in the neighborhood of 95 pounds per square inch Abs. and the evaporating pressures are greater than one atmosphere if the boiling temperature is higher than minus ten degrees fahrenheit.

Methyl chloride is about as harmless as carbon dioxide. It is very stable at the usual temperature of household refrigeration and is now being used as a refrigerant in a number of household refrigerating systems.

A sling psychrometer was used in addition to the wet and dry bulb thermometers which was mounted on a panel, in an effort to obtain more accurate results.

Air entered the room through infiltration, thus necessitating the division of the cooling load into a sensible load and a latent load. An advanced theory that the necessary amount of air will enter a room through the opening and closing of doors, being accepted as true.

E. Results

The dehumidifying capacity of the eliminators as indicated in Table 2, Appendix I, varies from 3.4 percent to 16 percent. Indicative of the high percentage in most cases is the respectively high initial humidity.

Table 4 indicates the amount of dehumidification obtainable through the combined action of the cooling coils and eliminator. The maximum drop in relative humidity was from 71 percent to 50 percent, a dehumidification of 21 percent. Attention is called to the fact that most of these cases represent maximum obtainable results. Therefore, we are justified in believing that below 48 percent humidity the machine is of little or no value. It is important, however, to remember that relative humidities below 50 percent are beyond the scope of this investigation.

Curve 1, Appendix III, is a psychrometric chart showing the approximate winter and summer temperature zones observed in this investigation. The winter wet bulb range is shown to be between 60 and 75 degrees fahrenheit while for summer conditions the range is from 65 to 80 degrees fahrenheit. It is interesting to note that the winter and summer wet bulb temperature ranges overlap each other by 10 degrees fahrenheit.

Table 6 gives in condensed form "The Properties of Mixtures of Air and Saturated Water Vapors," which was used in determining the relative humidities at different dry and wet bulb temperatures.

The required cooling load of 16,119.8 B.t.u. per hour shown in Table 5, extends outside the operating range of the Humidraft, which was 2613.4 B.t.u. per hour. This fact alone indicates the insufficient capacity of the machine for the required job.[4]

(4) Interest in the subject matter, explains the efforts of this investigation, regardless of this predetermined error. It was necessary to use the office and equipment already available.

IV. Discussion of Results

The relatively high dehumidifying capacity of the eliminator is a direct result of the large deflection angle and small spaces between eliminators. Approximate spacings of one-half an inch were used as compared to many present day machines with a distance of one-inch or more between eliminators.

The low unit dehumidifying capacity of the machine was a result of the small size compressor used. Illustrative of this is the fact that for a theoretically designed heat absorption unit of one hundred percent, the machine would have still been approximately one-half the required size. The running time of the unit varied from one and one-half to two and one-half hours. Attention should be called to the fact that due to the smallness of the unit all observations were made at the machine which introduced a slight unavoidable error. However, these readings compensated in part for the changing initial conditions, resulting from a constant varying of outdoor conditions.

"The Properties of Mixtures of Air and Saturated Water Vapors" table was taken from Section 1, page 3, of the Air Conditioning correspondence course presented by The Trane Company, La Crosse, Wisconsin. This table was used in finding all relative humidities in this investigation.

The psychrometric chart given in Appendix II is similar to the one published by Carrier Engineering Corporation, Newark, New Jersey. On this chart dry bulb temperatures are represented by vertical lines and wet bulb temperatures by oblique lines. In reading this chart it should be noted that in the case of saturated air, the dry bulb, wet bulb and dew point temperatures are identical.

The vapor radiation heating coils although efficient could have been improved by increasing the fin area. The scrubbing effect of the air over these coils was increased by the position of the fan and its variable direction of flow (see View 4, Appendix III).

Worthy of special attention is the sensible heat transmission through glass exposed to the sun as shown in Table 5; a major factor in determining the required heat load. It is important to observe that if this glass had been shaded with awnings the heat-transmission factor of 160 would have dropped to 30 thereby reducing the required load by as much as 80 percent. The Humidraft machine would then in all probability have been adequate for more pleasing results.

V. Summary

Favorable results indicated unlimited possibilities in the refrigeration method of Dehumidifying Air. Its use having become universal in practically all major units of any importance.

The water spray bath is probably the only successful means of humidifying the air.

Vapor radiation heating coils if properly designed to successfully encounter the enormous pressures often involved in refrigeration, frequently produce efficiencies equal that of the especially designed cooling coils.

Methyl Chloride has a very low heat absorbing capacity, thereby limiting its practicability as a refrigerant.

The Humidraft machine used in this investigation was inadequate to take care of the required load due to the low B.t.u. absorption of Methyl Chloride and to the low fan capacity.

VI. Acknowledgments

It is with sincere appreciation that the author gratefully acknowledges the many invaluable suggestions offered by Mr. F. F. Groseclose, Instructor in the Mechanical Engineering Laboratory.

To Professor J. B. Jones of the Mechanical Engineering Department for the many timely and helpful suggestions that he made.

Also to Mr. H. B. Groseclose of the Mechanical Engineering Laboratory for his able assistance in setting up the apparatus.

VII. Appendices

Appendix I: Bibliography

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Appendix II: Tabular and Graphical Results

Table 1

Conditions Recommended for Office Cooling

Outside Temp. Deg. F	Dry Bulb Deg. F	Wet Bulb Deg. F	Effective Temp. Deg. F	Relative Humidity Percent	Grains of Moisture per lb.Dry Air
95	80	65.2	73.4	45	69
90	78	64.5	72.2	47.5	69
85	76.5	64	71.1	50	69
80	75	63.5	70.2	52.5	69
75	73.5	63	69.3	56	69
70	72	62.5	68.2	60	69

Table 2

Combined Fan and Eliminator Capacity

Initial Room Conditions			Improved Room Conditions		
Dry Bulb Temp. Deg. F	Wet Bulb Temp. Deg. F	Relative Humidity Percent	Dry Bulb Temp. Deg. F	Wet Bulb Temp. Deg. F	Relative Humidity Percent
68	62	71.6	69.5	61.5	62.6
70.5	64	71.6	74	62	50.65
90	81.5	71	87	74.5	55
68	61	67.2	69	61	63.4
77	69	67	77	66	56
80	70	60.9	80	67.5	52
69	63	72	71	62	60.4
72	61	54	74	62	50.6

Table 3

Capacity of Spray Water System

Initial Room Conditions			Improved Room Conditions		
Dry Bulb Temp. Deg. F	Wet Bulb Temp. Deg. F	Relative Humidity Percent	Dry Bulb Temp. Deg. F	Wet Bulb Temp. Deg. F	Relative Humidity Percent
74	67	69.9	72	65.5	71.05
69.5	61.5	63.3	70	64	72.43
73	64	61.4	72	65.5	71.05
85	67	38.6	81	70.50	60
71.5	62	58.7	69	61.5	65.5
68	61	67.2	64	59	74.7

Table 4

Combined Fan, Eliminator and Cooling Coil Capacity(1)

Initial Room Conditions			Improved Room Conditions		
Dry Bulb Temp. Deg. F	Wet Bulb Temp. Deg. F	Relative Humidity Percent	Dry Bulb Temp. Deg. F	Wet Bulb Temp. Deg. F	Relative Humidity Percent
68	62	71.6	66	55.5	52
90	81.5	71	84	70	50
72	61	54	71	59	49
80	70	60.9	78	64.5	48
74					
74	68	73.7	72	61.5	55

(1) The accuracy of this data is explained in discussion of results.

Table 5

Required Cooling Load For Conditions as Shown

CONDITIONS	D.B.Temp.	W.B.Temp.	D.P.Temp.	Percent R.H.
OUTSIDE	85	67	57.5	38.6
INSIDE	76	64	57	51

HEAT SOURCES(1)	Units Figured	Factors	Sensible Heat	Latent Heat
Outside Glass East (Exposure) Unshaded	75.00 sq. ft.	160	12,017.6	
Outside Wall	45	3	135	
Inside Glass	13.5	20	270	
Inside Wall	642	1.5	963	
Ceiling Unsprayed	167.6	1.5	251.4	
Floor	167.6	1.5	251.4	
Air Contents	2320		624	34.4
Lights - Watts(3)				
Occupants	4	200 Lat. 200 Sen.	800	800
Total Sensible Load			15,285.4	
Total Latent Load				834.4
Total Load				16,119.8

(1) Those sources having no relation to the problem are omitted.

(2) Heat transmission constants for differential of 10 degrees.

(3) Where there is sufficient glass area so that artificial lighting is not required during the daytime, the load due to lights may be omitted.

Table 6

The Properties of Moisture of Air and Saturated Water Vapor

Temp. °F	Vol. in Cu. ft. of 1 lb. Dry Air plus Vap. to Saturate	Heat in Dry Air - B.t.u. Per lb.	Heat in Moisture - B.t.u. Per lb.	Total Heat (Columns 2 & 3) - B.t.u. Per lb.
50	13.00	12.07	8.12	20.19
51	13.03	12.31	8.43	20.74
52	13.07	12.55	8.75	21.30
53	13.10	12.79	9.08	21.87
54	13.13	13.03	9.41	22.45
55	13.16	13.28	9.76	23.04
56	13.20	13.52	10.13	23.64
57	13.23	13.76	10.50	24.25
58	13.26	14.00	10.89	24.88
59	13.30	14.24	11.28	25.52
60	13.33	14.48	11.69	26.18
61	13.36	14.72	12.12	26.84
62	13.40	14.97	12.56	27.52
63	13.43	15.21	13.01	28.22
64	13.47	15.45	13.48	28.93
65	13.50	15.69	13.96	29.65
66	13.54	15.93	14.46	30.39
67	13.58	16.18	14.97	31.15
68	13.61	16.42	15.50	31.92
69	13.65	16.66	16.05	32.71
70	13.69	16.90	16.61	33.51
71	13.73	17.14	17.19	34.33
72	13.76	17.38	17.79	35.17
73	13.80	17.63	18.41	36.03
74	13.84	17.87	19.05	36.91
75	13.88	18.11	19.71	37.81
76	13.92	18.35	20.38	38.73
77	13.96	18.59	21.08	39.67
78	14.00	18.84	21.80	40.64
79	14.05	19.08	22.55	41.63
80	14.09	19.32	23.31	42.64
81	14.13	19.56	24.11	43.67
82	14.17	19.80	24.92	44.72
83	14.22	20.04	25.76	45.80

Table 6 (Continued)

Temp. °F	Vol. in Cu. ft. of 1 lb. Dry Air plus Vap. to Saturate	Heat in Dry Air - B.t.u. Per lb.	Heat in Moisture - B.t.u. Per lb.	Total Heat (Columns 2 & 3) - B.t.u. per lb.
84	14.26	20.29	26.62	46.91
85	14.31	20.53	27.51	48.04
86	14.35	20.77	28.43	49.20
87	14.40	21.01	29.38	50.39
88	14.45	21.25	30.35	51.61
89	14.50	21.50	31.36	52.86
90	14.55	21.74	32.39	54.13
91	14.60	21.98	33.46	55.44
92	14.65	22.22	34.59	56.78
93	14.70	22.46	35.69	58.15
94	14.75	22.71	36.86	59.56
95	14.80	22.95	38.06	61.01
96	14.86	23.19	39.30	62.48
97	14.91	23.43	40.57	64.00
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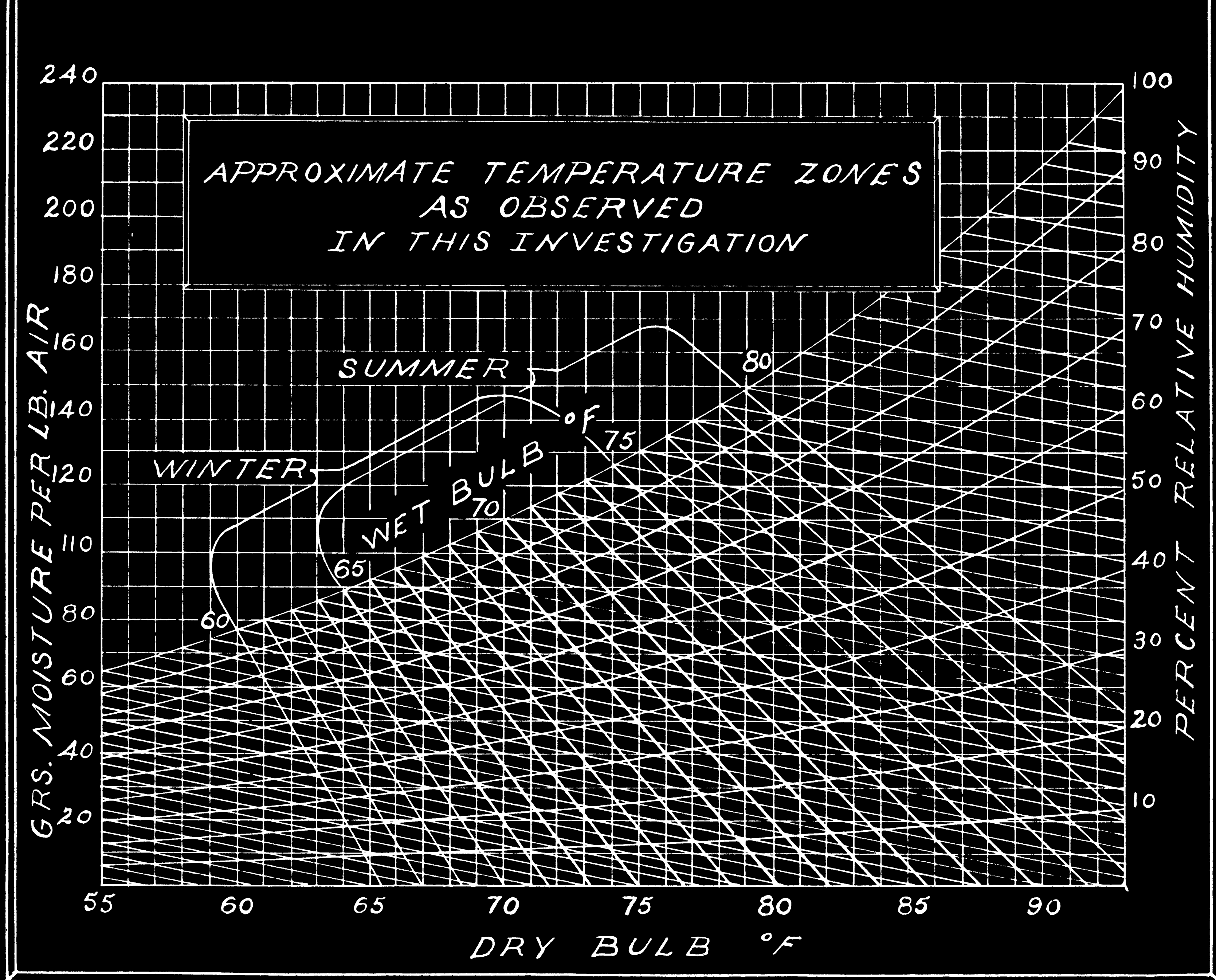
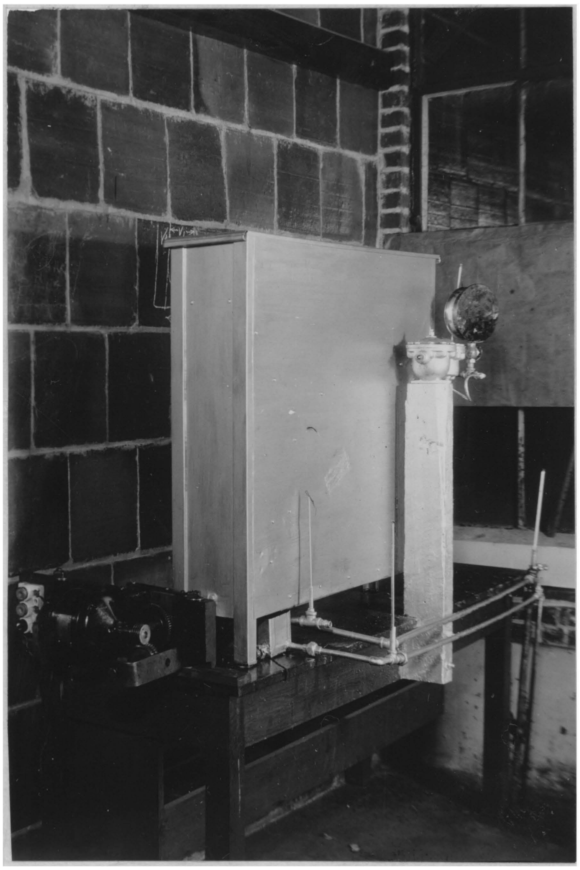
99	15.02	23.91	43.24	67.15
100	15.08	24.16	44.63	68.79

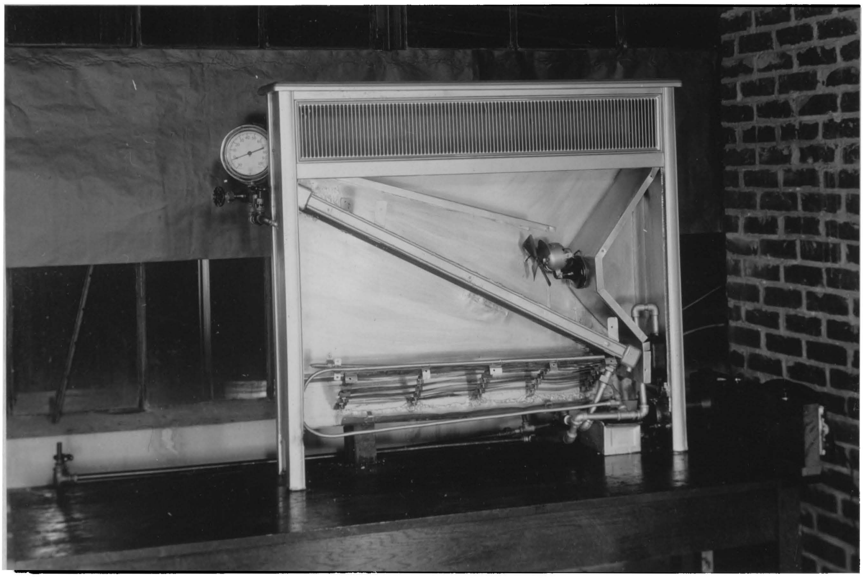

APPROXIMATE TEMPERATURE ZONES
AS OBSERVED
IN THIS INVESTIGATION
SUMMER
WINTER
WET BULB °F
60
65
70
75
80
GRS. MOISTURE PER LB. AIR
240
220
200
180
160
140
120
110
80
60
40
20
PERCENT RELATIVE HUMIDITY
100
90
80
70
60
50
40
30
20
10
55
60
65
70
75
80
85
90
DRY BULB °F

Appendix III: Views of Apparatus

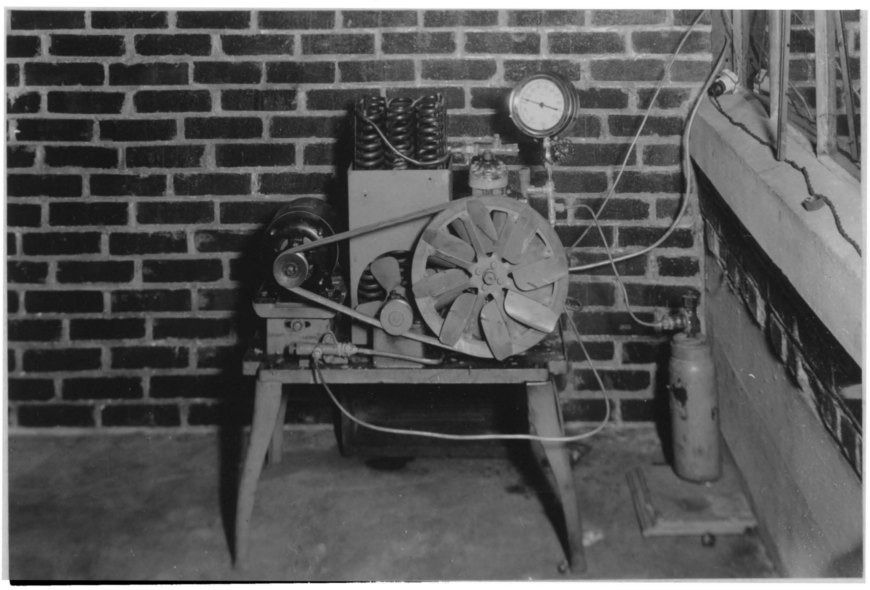
Front View



Back View



Interior View



"Methyl Chloride" Compressor