

A STUDY OF THE OPERATING CHARACTERISTICS
OF AN AIR COOLING SYSTEM

A THESIS SUBMITTED

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P R E F A C E

The rapid advance made by refrigerating engineering in the past few years has given rise for the need of specific information concerning the efficient operation of the recent developments.

This paper has been prepared for the purpose of presenting detailed as well as general information concerning the operation of the modern air cooling machine, and to present it in such a manner that it will be of interest and value to those who may be interested, even mildly, in any application of this refrigerating principle.

It differs from the other writing concerning this subject, primarily in the fact that it is free from the general influence of "sales points" that characterizes the majority of the material that has been prepared in writing, mostly, by the manufacturers.

GENERAL SURVEY.

Refrigerating engineering has become one of the most important phases of industrial and economic activity, and its increasing importance has brought out the need for research in this field, not for the purpose of seeking new applications, but for the more efficient utilization of the methods and equipment at hand.

Industrial enterprises, such as packing houses, cold storage shipping units, creameries and dairies, ice cream manufacturing plants, and food preservation industries have made the application of refrigeration universal.

Other uses of refrigeration are indicated by the installation of refrigerating machinery in all modern hotels, restaurants, apartment houses, hospitals, theatres, and public buildings. Of late, refrigerating processes are being used quite extensively in the rayon industry.

New uses for "cold" are being found every day, and the demand for information and data on refrigerating processes and equipment has become acute.

A comparatively new system of refrigerating has been placed on the market, and this system of direct air cooling gives promise of excellent results when used not only in connection with a cold storage plant, but also in the cooling and conditioning of air in factories and in public buildings.

The available data regarding recent installation of refrigerating systems of this nature is very limited, and there has been very little written on the subject.

The work that is presented herein supplements that which has been done by a few of the manufactures, only the results found from these tests, and others that may follow, will be presented as general information entirely free from bias, prejudice or sales propaganda.

PURPOSE.

The purpose in undertaking this work was to determine by experiment and observation the operating characteristics of a direct air cooling system and to present this information in such a manner that it will meet, in part, the demand for such material.

The special ends or determinations to which these tests have been directed are: (1) An investigation of the relation of humidity to the heat transfer rate; (2) the effect of the coefficient of performance on the refrigerating efficiency; and (3) the resultant effects on the operation of the compressor of the system due to the variation of conditions under which the air cooling unit was operated.

GENERAL THEORY.

The general principles or theory upon which this particular type of air cooling system operates is that the air circulated will produce the desired refrigerating effect by means of its sensible heat.

In this system, the air is cooled by passing over coils through which a refrigerating media is circulated, and is then blown into the room. The air, upon being circulated about the room, will rise a few degrees in temperature and thereby produce the refrigeration or cooling effect.

The whole system of operation, and the results produced, is dependent directly upon the amount of the refrigerating media circulated through the evaporating coils.

The first consideration to make then in the use of this type of machine should be with the method or system employed to control the amount of the refrigerant circulated.

The most plausible means or methods for control, in the light of present developments, are: (1) Hand control or regulation of a master valve; (2) Float control; and (3) Thermostatic control.

The results of any tests made to determine efficiency or general operating characteristics will be contingent upon the method of refrigerant circulation control adopted.

Method and Apparatus Used.

All of these tests were made with the refrigerant circulation controlled by means of hand regulation. This method of control was adopted because it is best suited to the installation. All other determinations were made according to the specifications of the American Society of Refrigerating Engineers.

As all determinations were made according to standard specifications this portion of the paper will be given over to a brief description and methods of operating the system, rather than to take up the theoretical considerations involved.

Compressor.

The tendency in designing ammonia compressors is toward higher operating speeds; and within recent years, it has been possible to obtain compressors with capacities from 25 to 50 tons of refrigeration per day, operating at speeds from 400 to

500 revolutions per minute. The advantages of higher speeds are obvious. Electric power is being used more and more for refrigerating plants, and it is desirable to avoid speed reduction from the motor to the compressor in order to obtain an efficient drive. Other advantages are (1) saving in space required; and (2) weight, which means lower first cost.

The compressor tested operates at 220 revolutions per minute; the driving motor operating at 880 revolutions per minute.

On the upward stroke, the cold vapor of the refrigerant, which comes from the cooling coils through the suction pipe, is drawn into the cylinder space below the piston. During the return (downward) stroke, the vapor passes through the suction valve into the space on the top side of the piston. At the end of the downward stroke, the suction valve closes, and the vapor of the refrigerant is then compressed during the next upward stroke. When the pressure of the vapor becomes great enough, the discharge valves open, and the vapor is forced out of the cylinder.

The cylinders are cooled by the circulation of water in a water jacket about the cylinders.

Volumetric Efficiency and Clearance.

The ratio of the volume of the vapor of the refrigerant, which is actually handled to the volume of the piston displacement, is called the volumetric efficiency.

There are a number of reasons why the volumetric efficiency of a compressor is less than 100%: (1) The amount of vapor of a refrigerant that will fill the volume of the piston displacement at the temperature and pressure in the cooling coils of the evaporator is always greater than the amount actually taken in.

(2) The vapor of a refrigerant is heated by throttling when it enters the cylinder of the compressor, and its heating by throttling produces a higher temperature than that corresponding to the pressure; in other words, if the vapor is dry and saturated in the suction pipe, it will be superheated in the cylinder of the compressor; (3) another condition tending to reduce the actual volume of the vapor of the refrigerant that enters the cylinder is the lowering of the density of the vapor of the refrigerant in the cylinder because its pressure is less than the pressure in the cooling coils of the evaporator; (4) still another reason for the reduced volume of the refrigerant entering the compressor cylinder is that there is a larger space inside the cylinder than is actually swept through by the piston. This excess volume is called the clearance. On this particular machine the clearance is very small, yet it produces an appreciable effect.

There are always some leakage and other losses in a compressor, so that the actual volume delivered will be somewhat less than the theoretical displacement of the cylinder. The losses are due to incomplete filling of the cylinder of the compressor at each stroke and leakage through the valves and around the piston.

In these tests no attempt has been made to determine the exact effects of the amount of clearance in the operation of the machine, nor to make a complete analysis of the factors affecting the volumetric efficiency. Such investigation is beyond the scope of this report. The reasons given above are only for the purpose of showing why the volumetric efficiency is always less than 100% and in reality approximates the values determined from these tests.

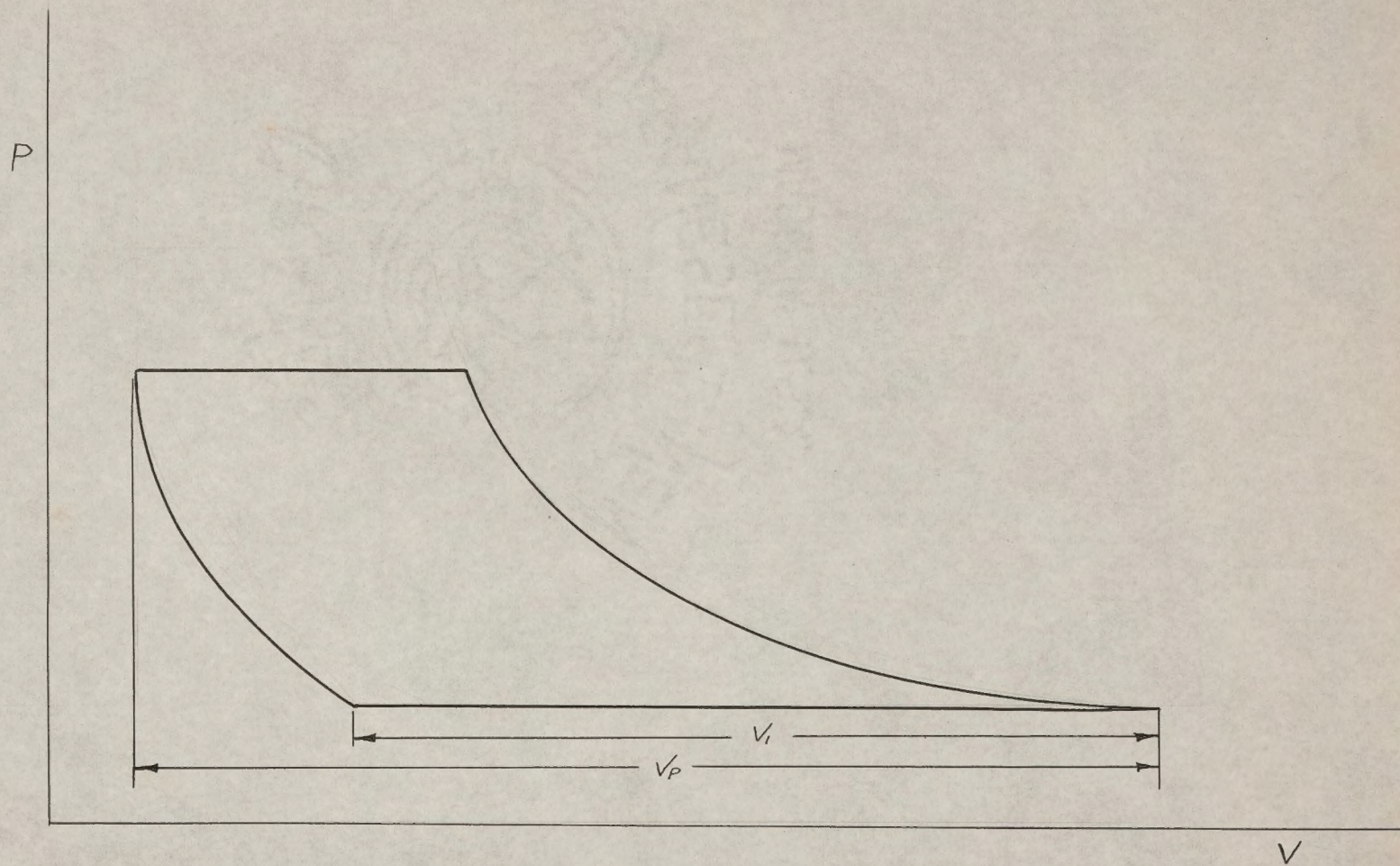


FIGURE A.

The diagram illustrates an ideal compressor card with clearance. The gas or vapor of the refrigerant which remains in the clearance space is --

$$V_c = C V_p$$

V_p = piston displacement of the cylinder:

C = percentage of clearance.

The apparent volumetric efficiency is the ratio of V_1 : V_p

$$V_1 = V_p + V_c - V_d$$

$$V_d = V_c \left\{ \frac{P_2}{P_1} \right\}^{\frac{1}{n}}$$

$$\frac{V_c}{V_p} = m$$

$$\begin{aligned} V_1 &= V_p + m V_p - m V_p \left\{ \frac{P_2}{P_1} \right\}^{\frac{1}{n}} \\ &= V_p \left[1 + m - m \left\{ \frac{P_2}{P_1} \right\}^{\frac{1}{n}} \right] \end{aligned}$$

$$\frac{V_1}{V_p} = 1 + m - m \left\{ \frac{P_2}{P_1} \right\}^{\frac{1}{n}}$$

The volumetric efficiency may be determined from the compressor indicator card by measuring the lengths of the lines V_1 and V_p and determining the ratio between. At best such a method is only an approximation, but it is sufficiently accurate for the results desired and was the method employed.

Condenser.

The condenser is a multi-pass unit. It is 8 inches in diameter by 18 feet long and contains seven two-inch No. 10 gage seamless steel tubes. The condenser is complete with gas, liquid and drain valves. The cooling water can be raised to any temperature desired by means of an auxiliary steam connection.

Receivers.

The liquid ammonia receiver is made up of two vertical units, each 12 inches in diameter by 6 feet long, complete with inlet, outlet, and drain valves, and gage glass connections. By the alternate use of the receivers the amount of ammonia being circulated at any time can be determined.

Heater.

Because of the size of the room it was necessary for test purposes to have some means of absorbing any excessive amount of "cold" produced. To accomplish this, electric resistor heaters were installed. The electric energy absorbed by these heaters was measured by means of suitable instruments, and the energy was taken as being equivalent to the amount of "cold" absorbed or neutralized.

Cooling Unit.

One of the chief difficulties encountered during the test was the accurate measurement of the ammonia circulated. This difficulty could be overcome to a larger extent by the installation of graduated gage glasses.

Bubbling of the liquid ammonia in the gage glass contributed to the difficulties encountered, but it was discovered that this troublesome feature could be overcome by increasing the discharge pressure so that the saturation temperature of the refrigerant would always be above the room temperature.

The greatest source of error entering into the results determined from the tests would probably be in the measurement of the ammonia circulated. Some error would be introduced due to the fact that on successive tests the same therm-

ometers were not used. In addition there is the natural error introduced by the opening of the door in the cold room and the lighting required.

Every precaution was taken to avoid the introduction of error due to inaccurate readings and observations.

Observations and Operation.

All observations are given here in the form of the tabulated data.

In the operation for each test the discharge pressure and the suction superheat were kept constant. The tests were then classified according to the varying conditions due to change of humidity; velocity of air circulated; heat input to the cold room; and load variation.

- A - Pressure by Gage in Discharge of Compressor. Pounds Per Square Inch
- B - Pressure by Gage in Suction of Compressor. Pounds Per Square Inch
- C - Voltage on Heater, Volts.
- D - Current Through Heater, Amperes.
- E - Power Input to Motor, Kilowatts.
- F - Speed of Fan, Revolutions Per Minute.
- G - Speed of Motor, Revolutions Per Minute.
- H - Speed of Compressor, Revolutions Per Minute.
- I - Compressor Room Temperature, Degrees Fahrenheit.
- J - Temperature Cold Room, Degrees Fahrenheit.
- K - Temperature of Gas at Discharge of Compressor, Degrees Fahrenheit.
- L - Temperature of Liquid Leaving Condenser, Degrees Fahrenheit.
- M - Temperature of Liquid Leaving Receivers, Degrees Fahrenheit.
- N - Temperature of Vapor Leaving Expansion Valve, Degrees Fahrenheit.
- O - Temperature of Gas Leaving Cooling Coils, Degrees Fahrenheit.
- P - Temperature of Gas of Suction of Compressor, Degrees Fahrenheit.
- Q - Temperature of Cooling Water, Degrees Fahrenheit.
- R - Temperature of Cooling Water Leaving Condenser, Degrees Fahr.
- S - Temperature of Cooling Water Leaving Compressor Jacket, Degrees Fahr.
- T - Humidity in Cold Room, Percent.

Data Run No. 1

A	B	C	D	E	F	G	H	I	J
120	5	222	20.2		1170	800	200	78	19
120	2.5	222	21.0		1100	880	220	76	38
120	5	222	20.8		1100	880	220	76	39
120	8	222	20.8		1100	880	220	76	29
120	11	222	20.8		960	880	220	77	25
120	11	222	20.8		1140	880	220	76	24
120	11	222	20.8		1120	880	220	76	23
120	10	222	20.8		1120	800	200	76	21
120	9	222	20.8		1120	800	200	75	19

K	L	M	N	O	P	Q	R	S	T
136	68	68	-20		17	50	70	58	54
128	66	66	-20		40	50	70	56	67
140	68	68	-20		45	50	73	54	67
168	68	68	-20		42	50	70	54	60
172	72	72	-12		-5	50	73	54	59
150	69	69	-12		-5	50	70	54	59
138	72	72	-12		-10	50	72	53	59
137	68	68	-12		-10	50	70	52	59
143	69	69	-14		-10	50	72	54	59

Run No. 2

A	B	C	D	E	F	G	H	I	J
120	10	223	23.0		1730	880	220	70	18
120	10	223	23.0		1730	880	220	72	18
120	10	223	23.0		1730	880	220	73	17
120	10	223	23.2		1740	880	220	73	16
120	14	223	23.0		1740	880	220	73	19
120	13	223	23.0		1720	880	220	73	18
120	12	223	23.0		1720	880	220	73	17
120	13	223	23.0		1720	880	220	74	17
120	14	223	23.0		1720	880	220	73	18
120	16	223	23.0		1720	880	220	72	20
120	16	223	23.0		1720	880	220	72	20
K	L	M	N	O	P	Q	R	S	T
139	69	69	-10		-5	50	69	53	69
154	72	72	-10		-5	50	72	54	69
165	72	72	-10		4	50	72	54	69
170	74	74	-10		16	50	71	56	69
178	74	74	-5		-5	50	74	56	70
162	72	72	-5		-5	50	72	54	70
162	66	66	-5		-8	50	72	54	70
160	68	68	-10		-6	50	72	54	70
150	70	70	-5		-4	50	72	53	71
138	71	71	-4		2	50	72	52	72
131	70	70	-2		2	50	72	52	73

Run No. 3

A	B	C	D	E	F	G	H	I	J
120	9	000	00.0		1145	860	215	76	10
120	23	218	83.0		1145	860	215	75	40
120	18	218	82.0		1150	856	214	73	48
120	12	218	81.6		1150	856	214	73	49
120	12	217	81.6		1150	856	214	72	53
120	13	218	81.6		1150	856	214	72	54
120	15	218	81.6		1150	860	215	71	55
120	16	218	81.6		1150	860	215	71	56
120	18	218	81.6		1150	860	215	76	57

K	L	M	N	O	P	Q	R	S	T
127	73	73	00		-5	50	72	52	59
184	72	72	00		6	49	72	48	62
186	74	74	00		0	48	72	50	55
180	71	71	00		10	50	71	52	50
162	72	72	00		26	49	72	56	47
172	74	74	00		28	49	72	54	43
170	68	68	-2	2	-2	48	71	55	42
156	70	70	00	2	00	49	71	52	41
140	72	72	2	4	2	49	72	52	41

Run No. 4

A	B	C	D	E	F	G	H	I	J
120	18	000	00.0		1713	848	212	80	31
120	22	218	88.4		1713	848	212	80	38
122	20	219	82.8		1713	848	212	81	59
120	17	219	82.2		1713	848	212	81	66
120	18	219	82.0		1712	848	212	80	74
120	22	219	81.6		1712	844	211	76	81
123	37	219	81.8		1712	844	211	76	78
116	28	219	81.6		1720	844	211	76	79
120	27	219	81.6		1720	844	211	75	85

K	L	M	N	O	P	Q	R	S	T
116	74	74	2		2	49	71	52	100
92	72	72	10		7	49	72	52	95
88	72	72	10		7	48	72	50	95
142	68	68	10		32	49	72	52	98
162	73	73	15		50	49	73	53	99
176	65	65	20		62	49	70	54	99
153	74	74	38		24	48	70	54	94
119	71	71	28		33	48	68	52	97
159	68	68	19		62	48	69	53	95

Run No. 5

A	B	C	D	E	F	G	H	I	J
120	11	220	84.2		1720	852	213	80	48
114	7	221	83.0		1720	852	213	80	63
120	11	220	82.4		1720	852	213	80	79
120	16	220	82.0		1720	852	213	80	81
120	29	220	82.0		1720	860	215	79	79
125	37	221	82.0		1720	860	215	79	74
120	23	220	82.0		1720	860	215	78	74
120	26	220	82.0		1720	848	212	78	75
120	28	220	82.0		1720	848	212	77	71

K	L	M	N	O	P	Q	R	S	T
172	70	70	-7	2	00	51	72	54	78
171	69	69	-13	32	27	50	69	54	74
181	72	72	-7	60	51	50	72	54	66
189	72	72	1	64	64	50	70	55	80
186	74	74	7	44	69	50	72	54	65
130	74	74	23	24	22	50	71	54	71
126	68	68	7	44	32	50	70	52	83
168	73	73	14	38	52	50	71	66	74
148	74	74	14	32	18	50	70	52	84

Run No. 6

A	B	C	D	E	F	G	H	I	J
120	15	000	00.0		1720	860	215	80	29
120	22	218	82.0		1720	860	215	80	50
120	14	219	82.2		1720	860	215	81	56
120	14	219	81.6		1720	860	215	80	60
120	20	221	82.4		1720	860	215	80	65
124	20	221	82.4		1740	860	215	80	64
120	16	221	82.4		1740	860	215	80	62
120	16	221	82.4		1740	860	215	78	63
120	25	224	83.2		1740	860	215	78	65
120	22	225	83.2		1740	860	215	78	66
120	15	226	83.8		1740	860	215	77	66

K	L	M	N	O	P	Q	R	S	T
168	72	72	-3	-2	-4	57	70	62	65
133	74	74	8	8	8	57	72	59	65
154	68	68	-5	26	22	57	70	59	61
169	69	69	-2	38	38	57	70	58	58
188	71	71	7	24	33	57	70	92	80
182	75	75	5	12	6	57	72	59	58
176	73	73	0	19	14	57	70	58	58
185	62	62	3	29	38	56	70	60	58
154	71	71	10	13	8	56	71	59	61
138	71	71	7	15	6	57	70	59	61
162	70	70	0	30	26	56	68	59	60

Run No. 7

A	B	C	D	E	F	G	H	I	J
120	25	225	20	4.5	1114	880	220	76	29
120	17	225	20	4.5	1114	880	220	77	26
120	8	226	20	4.5	1146	882	221	78	19
120	8	226	20	4.5	1146	882	221	78	18
K	L	M	N	O	P	Q	R	S	T
72	72	68	10	10	12	58	70	58	72
78	69	70	00	3	3	58	70	58	67
126	69	70	-12	-5	3	58	70	70	60
148	68	71	-13	-4	15	58	70	64	59

Run No. 8

A	B	C	D	E	F	G	H	I	J
120	10	226	20		1685	885	222	78	18
120	9	226	20		1685	885	222	78	17
120	9	225	20		1676	884	222	78	17
K	L	M	N	O	P	Q	R	S	T
162	72	70	-10	0	22	58	72	61	59
170	72	71	-11	5	27	58	70	62	60
172	72	71	-12	9	34	58	72	61	60

Run No. 9

A	B	C	D	E	F	G	H	I	J
150	10	224	42.0		1687	881	220	78	32
150	20	223	41.8		1687	878	219	77	33
150	13	224	42.0		1687	878	219	77	32
K	L	M	N	O	P	Q	R	S	T
122	78	78	-7	15	5	58	81	61	64
154	83	81	6	5	5	58	82	62	60
158	82	80	-4	12	12	58	82	62	58

Run No. 10

A	B	C	D	E	F	G	H	I	J
150	15	224	41.6		1152	880	220	77	33
150	23	224	41.4		1152	880	220	77	37
150	18	225	41.4		1160	880	220	78	38
150	15	224	41.4		1160	880	220	78	35
K	L	M	N	O	P	Q	R	S	T
173	83	82	-2	0	20	58	82	62	55
147	83	81	9	9	8	58	82	60	55
120	83	82	3	2	2	59	82	60	53
138	80	81	-2	-1	-2	58	82	62	51

Run No. 11

A	B	C	D	E	F	G	H	I	J
150	15	223	57.2	9.25	1152	872	218	76	38
145	11	223	57.0	8.25	1152	872	218	76	45
150	15	223	56.8	8.25	1152	872	218	76	46
150	18	223	56.8	8.25	1152	868	217	76	46
150	15	223	56.8	9.00	1152	868	217	76	45

K	L	M	N	O	P	Q	R	S	T
110	82	76	-4	6	00	62	82	62	64
144	82	78	-9	19	24	62	82	65	62
166	83	79	-2	18	30	62	83	66	59
174	84	80	1	4	1	62	82	66	53
180	83	80	-3	12	16	62	83	66	50

Run No. 12

A	B	C	D	E	F	G	H	I	J
150	18	223	57.6	9.5	1674	880	220	77	45
150	19	223	58.6	9.75	1674	880	220	77	44
150	17	223	58.6	9.25	1674	880	220	78	43

K	L	M	N	O	P	Q	R	S	T
189	84	80	1	12	26	62	84	67	50
196	84	82	3	7	5	62	84	67	52
196	82	80	0	16	19	62	82	67	53

Run No. 13

A	B	C	D	E	F	G	H	I	J
150	22	223	61	10.25	1686	880	220	78	46
150	21	221	82	10.0	1686	880	220	78	45
K	L	M	N	O	P	Q	R	S	T
202	83	82	8	10	8	62	84	68	55
191	82	82	4	9	5	62	82	67	56

Run No. 14

A	B	C	D	E	F	G	H	I	J
150	24	222	81.8	9.75	1137	876	219	78	44
150	14	222	80.8	8.5	1137	876	219	78	51
155	22	222	80.6	10.0	1145	870	218	78	58
145	18	222	80.6	9.0	1145	870	218	79	59
K	L	M	N	O	P	Q	R	S	T
108	84	79	7	8	8	63	84	64	61
153	82	80	-4	19	18	63	82	66	46
188	85	81	7	9	8	63	85	68	39
192	76	82	1	17	19	63	81	68	33

Run No. 15

A	B	C	D	E	F	G	H	I	J
150	22	222	81	10.0	1700	870	218	80	57
150	25	222	81	10.0	1682	870	218	80	56
142	20	222	81	9.5	1682	870	218	80	54

K	L	M	N	O	P	Q	R	S	T
198	83	82	7	18	18	64	83	68	37
176	81	82	8	14	9	64	80	68	40
190	80	81	4	23	38	64	80	68	38

Run No. 16

A	B	C	D	E	F	G	H	I	J
150	18	223	60.0	9.5	1150	892	226	76	51
155	25	224	56.0	10.75	1150	892	226	76	56
150	19	223	55.6	9.75	1177	885	221	76	56
150	20	223	55.6	9.75	1177	885	221	76	57

K	L	M	N	O	P	Q	R	S	T
170	84	78	2	15	27	68	83	87	58
185	84	80	10	10	11	68	84	72	64
188	83	81	3	17	24	68	82	73	63
196	83	80	4	8	25	68	83	74	63

Run No. 17

A	B	C	D	E	F	G	H	I	J
150	21	223	57.2	10.0	1682	873	218	76	54
150	24	224	57.2	10.0	1682	873	218	76	53
150	23	223	57.2	10.0	1682	873	218	77	53
150	21	224	57.2	10.0	1682	873	218	77	54

K	L	M	M	O	P	Q	R	S	T
204	82	82	6	24	28	68	83	74	72
210	84	82	9	11	17	68	83	75	70
206	84	83	8	10	12	68	82	75	75
206	82	82	4	31	32	68	82	75	78

Run No. 18

A	B	C	D	E	F	G	H	I	J
145	15	223	57.2	9.00	1700	887	222	79	35
150	18	223	57.2	9.75	1700	887	222	80	39
150	16	223	57.2	9.25	1702	882	220	80	40
150	18	223	57.2	9.75	1702	882	220	77	41

K	L	M	N	O	P	Q	R	S	T
164	80	80	-3	12	17	68	80	70	67
178	82	80	-2	6	4	68	81	71	75
182	81	81	-1	15	16	68	81	72	78
188	82	81	2	6	4	68	82	73	75

Run No. 19

A	B	C	D	E	F	G	H	I	J
150	15	223	56	9.25	1170	880	220	78	43
145	13	223	56	8.75	1170	880	220	78	46
150	15	223	56	8.75	1170	880	220	78	48

K	L	M	N	O	P	Q	R	S	T
180	81	81	-1	2	1	68	81	73	71
184	80	81	-5	13	16	68	80	73	74
188	81	80	-3	13	28	68	81	74	74

Run No. 20

A	B	C	D	E	F	G	H	I	J
150	33	223	55.6	11.0	1165	835	209	76	57
145	19	224	55.6	9.75	1165	887	222	76	57
150	18	224	55.6	9.75	1165	887	222	76	58

K	L	M	N	O	P	Q	R	S	T
124	82	81	19	19	19	67	81	69	99
140	80	82	2	16	12	68	81	70	91
166	81	80	3	8	6	68	81	70	96

RESULTS

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21
Temperature of Cold Room, Degrees Fahr.	24	18	55	81	74	65	23	17.3	32.3	35.8	44.0	44.0	45.5	53.0	56	55	53.5	39.0	45.6	57.6	88.0
Relative Humidity in Cold Room, Percent	60	70	43	98	75	60	65	60	61	53	58	52	55	45	38	62	74	74	73	95	25
Temperature Difference, Outside and Inside Room	52	55	17	-5	5	14	57	60	45	42	30	33	33	25	24	20	23	40	32	19	-7
Weight of Ammonia Circulated, Pounds per Hour	89.5	106	139.2	227	220	182	98.5	80.4	108.8	134.4	104.6	119.5	142.3	117.0		121.6	137.2	117.8	115.1	164.8	
Cooling Water Supplied to Condenser, G. P. M.	3.88	5.00	6.17	9.45	9.85	12.70	9.24	7.32	5.04	5.45	5.54	6.95	8.74	6.62	10.50	9.20	10.30	9.52	8.55	11.80	4.40
Cooling Water Supplied to Jacket, G. P. M.	5.45	3.08	3.03	3.81	4.74	4.25	1.91	4.87	4.42	4.52	3.45	3.20	3.12	3.05	3.50	2.59	2.63	4.07	2.93	2.30	1.94
Refrigerating Effect, B. T. U. per Pound	500	490.7	497.2	513.7	512.2	501.8	492.0	495.5	486.8	478.1	488.5	485.3	481.1	485.6	487.3	485.6	486.5	484.5	484.9	484.5	478.3
Compressor Effect, B. T. U. per Pound	58.0	71.1	67.7	47.5	52.6	69.7	39.5	74.7	54.9	60.5	60.6	85.7	90.2	64.7	77.7	80.6	90.2	76.9	80.0	54.4	50.7
Available Cooling Effect, B. T. U. per Hour	44700	52000	69300	116400	112800	91400	48500	39800	52900	64300	51000	58000	68600	56900		59000	66800	57000	55900	79800	
Tons of Refrigeration Produced	3.73	4.34	5.78	9.7	9.37	7.6	4.04	3.32	4.41	5.35	4.25	4.83	5.71	4.74		4.92	5.57	4.75	4.65	6.65	
Theoretical Horsepower of Compressor	2.02	2.96	3.71	4.24	4.55	4.99	1.53	2.36	2.34	3.20	2.5	4.03	5.05	2.98		3.85	4.87	3.56	3.62	3.52	
Total Radiation, B. T. U. per Hr. per Degree Temp. Diff.	1492	2166	1540	2530	2250	1750	1980	1917	1720	1956	1285	1550	1813	1051		1272	1643	1679	1316	1710	
Heat Transfer Rate, Radiation per Sq. Ft. Cooling Surface	10.7	15.5	11.0	18.1	16.1	12.5	14.2	13.7	12.3	14.0	9.18	11.1	12.95	7.5		9.09	11.74	12.0	9.4	12.2	
Superheat of Gas at Suction of Compressor, Degrees Fahr.	22	3.8	12.5	30.2	33.5	15.4	1.5	14.2	12.6	0.0	12.5	8.6	2.5	8.4	10.0	6.6	10.7	8.0	11.2	5.0	2.0
Superheat of Gas at Discharge, Degrees Fahr.	73.7	82.3	87.3	79.3	88.3	96.3	33.3	95.3	60.5	60.5	70.5	109.5	112.5	75.5	103.5	100.5	121.5	93.5	99.5	58.5	41.5
Coefficient of Performance	8.62	6.90	7.35	10.8	9.74	7.20	12.45	6.64	8.87	7.90	8.06	5.67	5.34	7.50	6.27	6.03	5.40	6.30	6.05	8.90	9.44
Heat Input to Cold Room, B. T. U. per Hr.	15300	18300	60700	61000	61500	63100	15370	15370	32100	31700	43250	44300	54200	61400	61400	43250	43700	43700	42700	42500	16200
Radiation Into Room, B. T. U. per Hr. per Deg. Temp. Diff.	565	613	506		1028	2020	581	407	473	776	258	415	436			788	1005	332	413	1960	
Electrical Horsepower of Compressor											11.51	12.72	13.60	12.50	13.20	13.31	13.40	12.50	11.72	13.05	9.73
Electrical Horsepower per Ton of Refrigeration											2.71	2.64	2.38	2.64		2.71	2.40	2.63	2.52	1.96	
Indicated Horsepower of Compressor							5.59		7.59	7.97	7.28	7.79	7.66	6.87		7.71	7.67	6.82	6.39	7.80	
Indicated Horsepower per Ton of Refrigeration							1.68		1.72	1.49	1.71	1.61	1.34	1.45		1.57	1.38	1.44	1.37	1.17	
Refrigerating Cycle Efficiency, Percent							42.3		31.0	40.2	34.6	51.7	66.0	43.4		50.0	63.6	52.3	56.8	45.2	
Mechanical Efficiency of Compressor, Percent											63.2	61.0	56.3	55.0		57.8	57.4	54.6	54.4	59.7	
Volumetric Efficiency of Compressor, Percent							65		71	67	67	69	64	66		68	68	71	70	68	55

Run No. 21

A	B	C	D	E	F	G	H	I	J
150	11	218	23.0	7.5	00	884	221	80	58
145	21	217	23.0	9.0	00	884	221	80	69
145	20	220	23.0	9.5	00	884	221	80	77
153	10	211	22.2	8.0	00	884	221	80	86
143	5	212	22.4	7.25	00	884	221	80	89
154	5	212	22.4	6.75	00	884	221	80	88
141	5	210	22.4	7.00	00	884	221	81	88
145	7	210	22.4	7.25	00	884	221	81	90

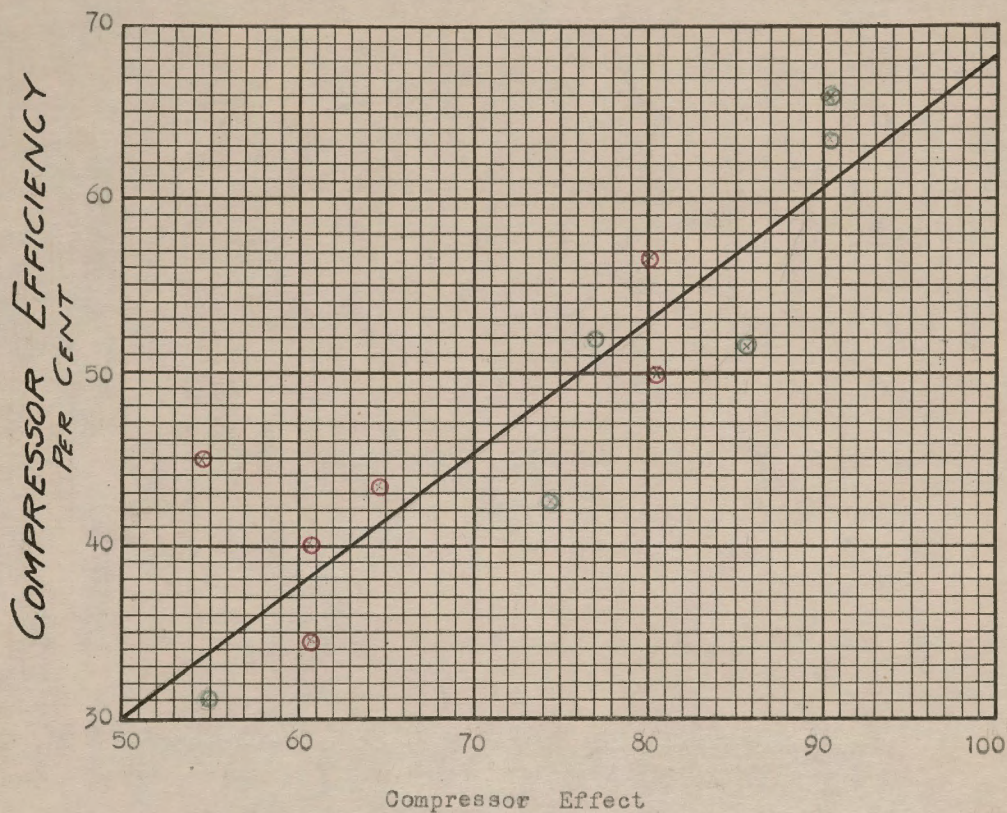
K	L	M	N	O	P	Q	R	S	T
158	84	76	-4	-4	-6	66	82	66	58
102	82	78	9	7	5	66	82	66	45
90	82	80	9	7	8	68	82	66	36
112	83	80	-9	-11	-8	68	82	82	28
122	82	80	-15	-13	5	68	82	82	26
130	84	81	-16	-9	25	68	82	83	24
143	83	80	-15	-15	30	68	80	72	23
148	82	80	-9	-11	-8	67	83	72	22

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FIG. NO. 1.

TITLE:-

Relation Between Compressor Effect and Compressor Efficiency

INTERPRETATION OF CURVES

○ Indicates Heavy Load

○ Indicates Light Load

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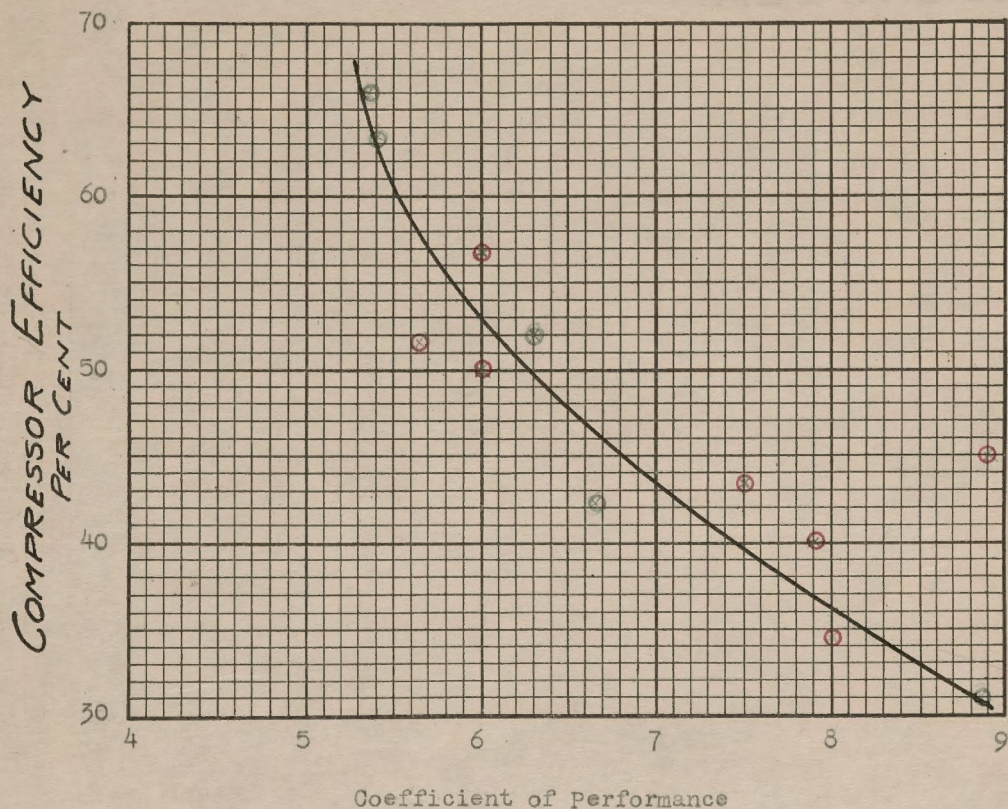


FIG. NO. 2.

TITLE:-

Relation Between Coefficient of Performance and Compressor Efficiency

INTERPRETATION OF CURVES

○ Indicates Heavy Load

○ Indicates Light Load

CONCLUSIONS

The conclusions reached from this study are presented here in the form of interpretations of the derived curves and graphs.

Figure 1 shows that there is a straight line relationship between compressor efficiency and compressor effect. The equation for the relationship being --

$$E_c = 0.766 C_e - 8.3$$

E_c = Compressor efficiency.

C_e - Compressor effect.

The indication is that the compressor operates more efficiently at the heavier loads. Therefore, in making a selection of a compressor for any installation, one should be selected so that it will operate at rated capacity.

From Figure 2 it appears that the refrigerating cycle efficiency varies inversely with the coefficient of performance. Thus it would seem that the best operation to be obtained would be with the ratio of the work done by the cooler to that done by the compressor low. This is contrary to the general opinion of refrigerating engineers.* This is probably due to the fact that the compressor operates most efficiently under heavy load. It is evident that a point representing some coefficient of performance will be reached that will give the highest possible

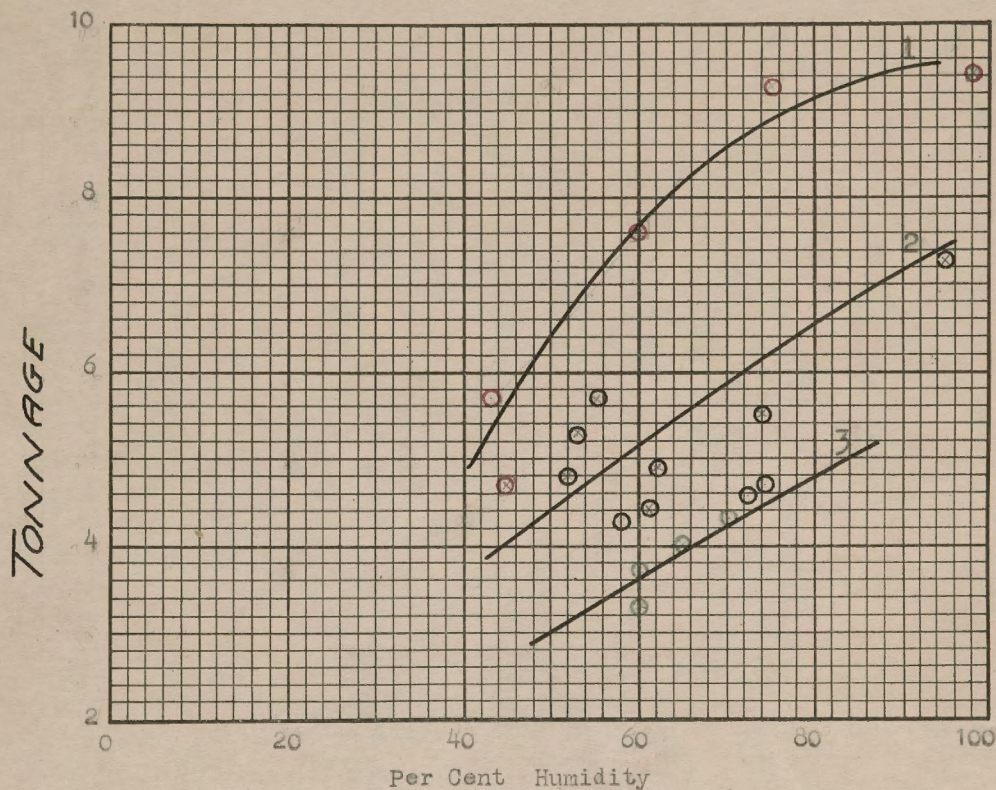
*"Handbook of Refrigerating Engineering" by W. R. Woolrich, page 261.

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FIG. NO. 3.

TITLE:-

Relation Between Humidity and Tonnage (Varying Loads)

INTERPRETATION OF CURVES

- #1 Heavy Load
- #2 Medium Load
- #3 Light Load

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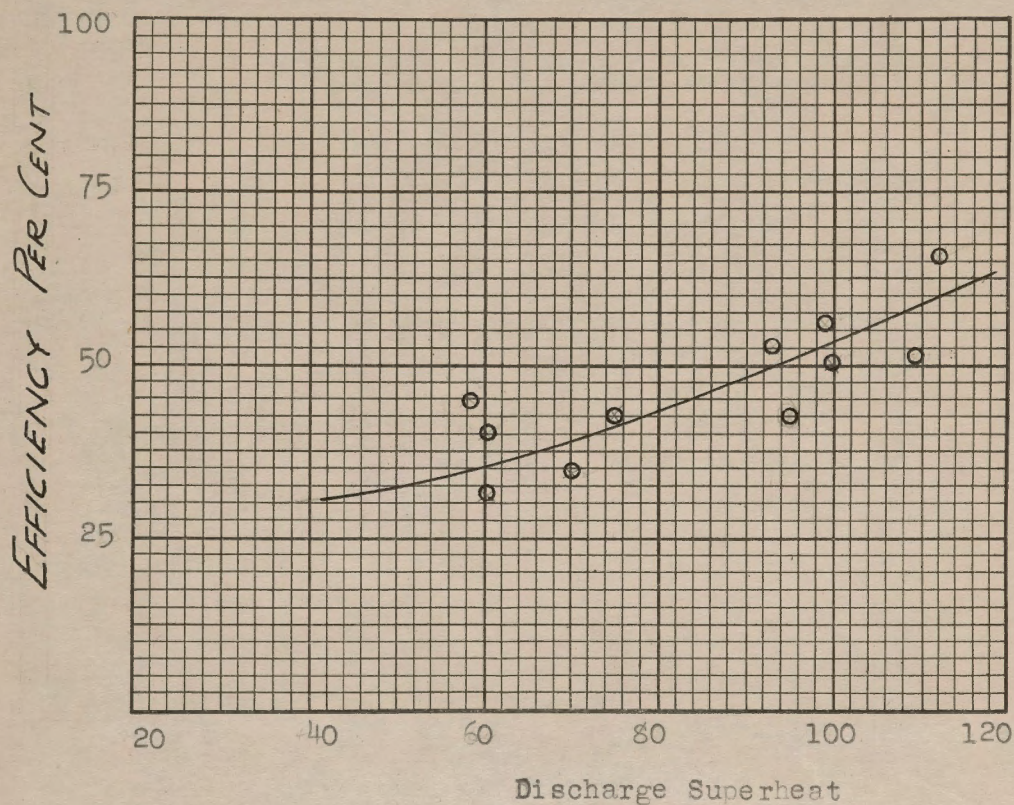


FIG. NO. 4

TITLE:-

Relation Between Compressor Efficiency and
Discharge Superheat

INTERPRETATION OF CURVES

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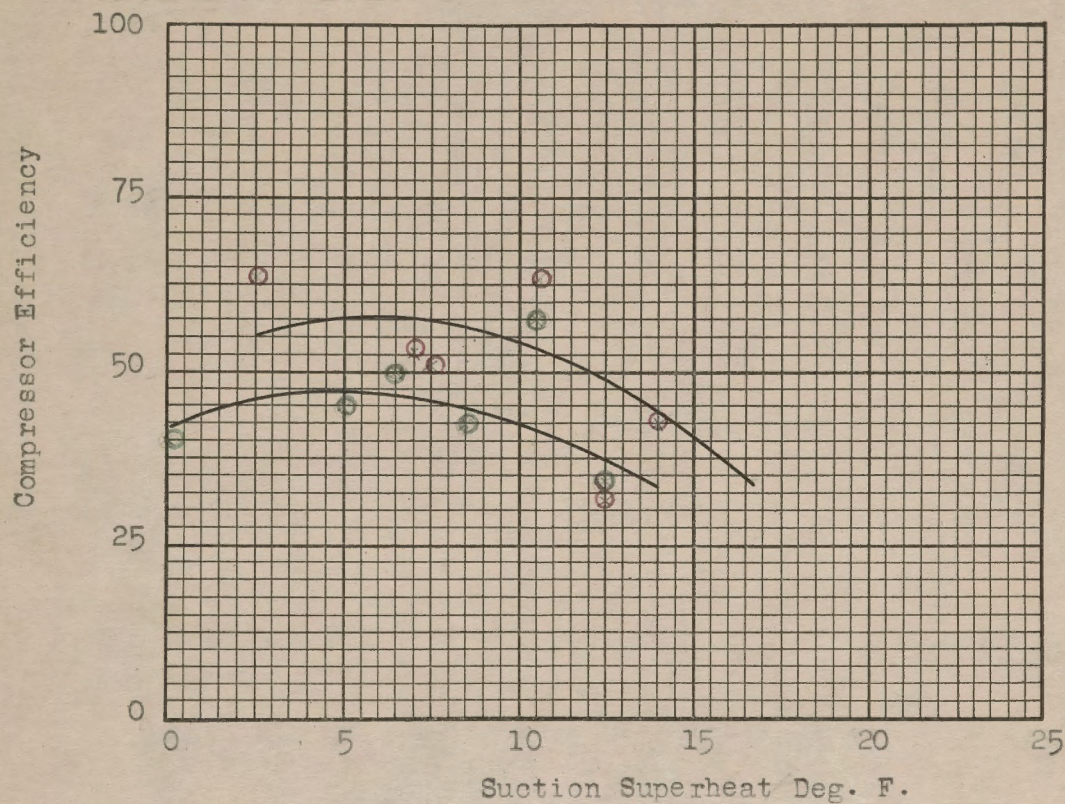


FIG. NO. 5

TITLE:-

Relation Between Compressor Efficiency and
Suction Superheat.

INTERPRETATION OF CURVES

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Name *g. m. adams*

Radiation - B.T.U. per Hr. per Deg. Temp. Diff.

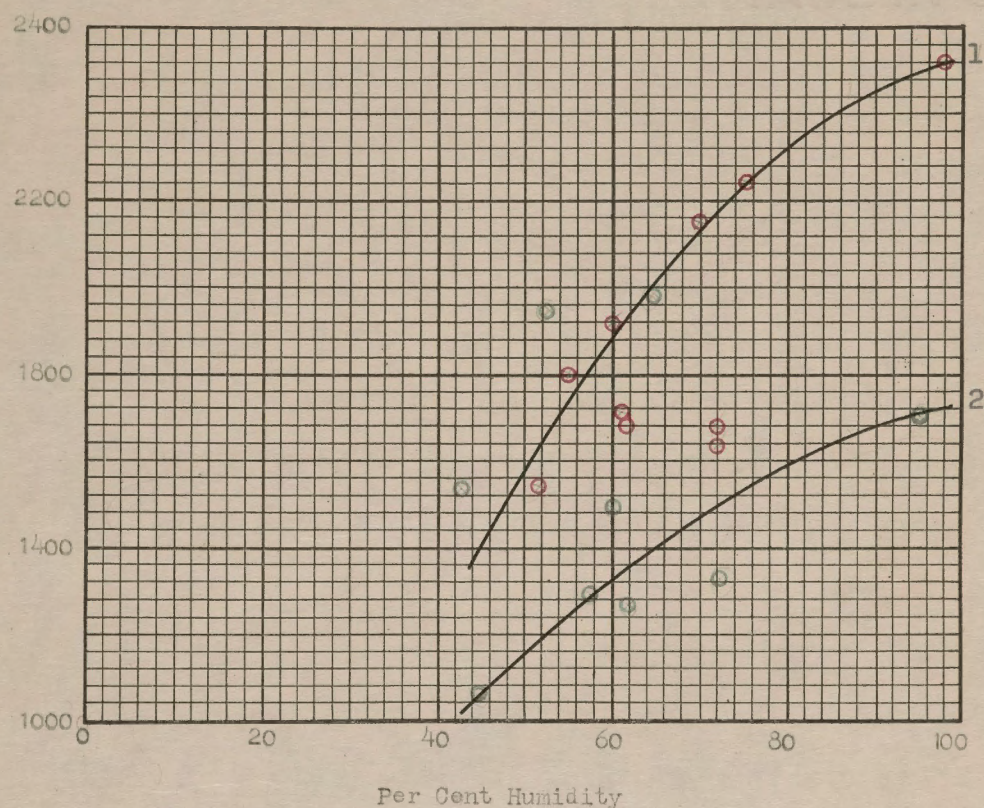


FIG. NO. 6.

TITLE:-

Variation of Heat Transfer With Humidity and Air Velocity

INTERPRETATION OF CURVES

- High Speed
- Low Speed

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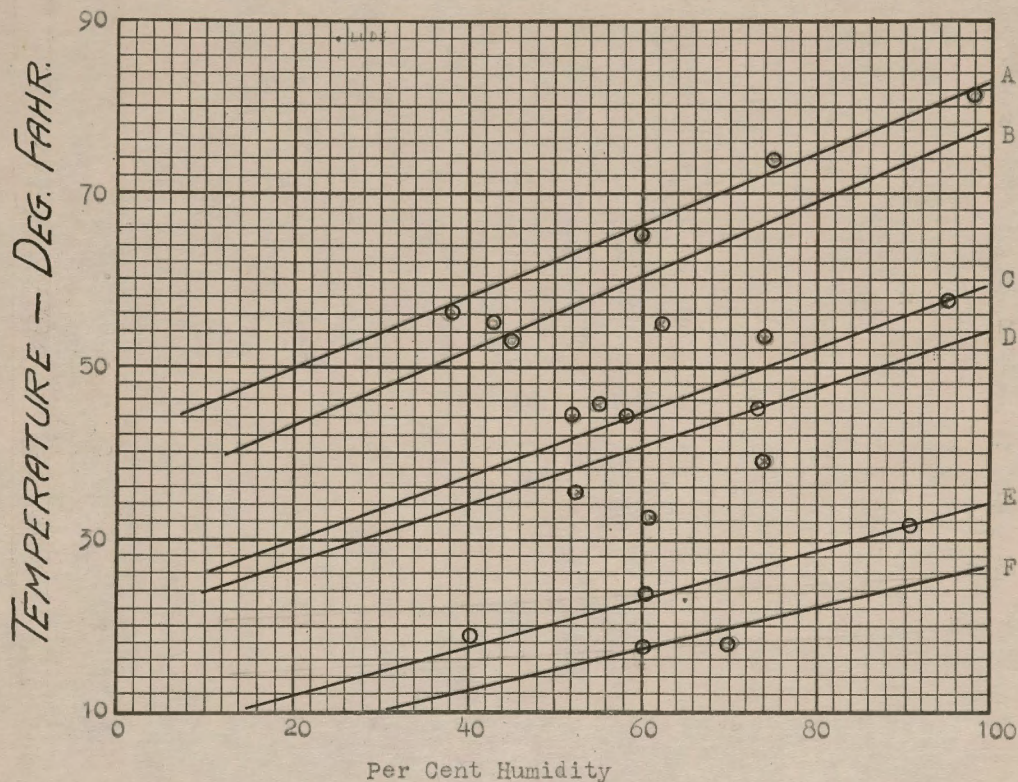


FIG. NO. 7

TITLE:-

Variation of Temperature With Humidity, Load, and Air Velocity.

INTERPRETATION OF CURVES

A	Heavy Load	High Speed
B	Heavy Load	Low Speed
C	Medium Load	Low Speed
D	Medium Load	High Speed
E	Light Load	Low Speed
F	Light Load	High Speed

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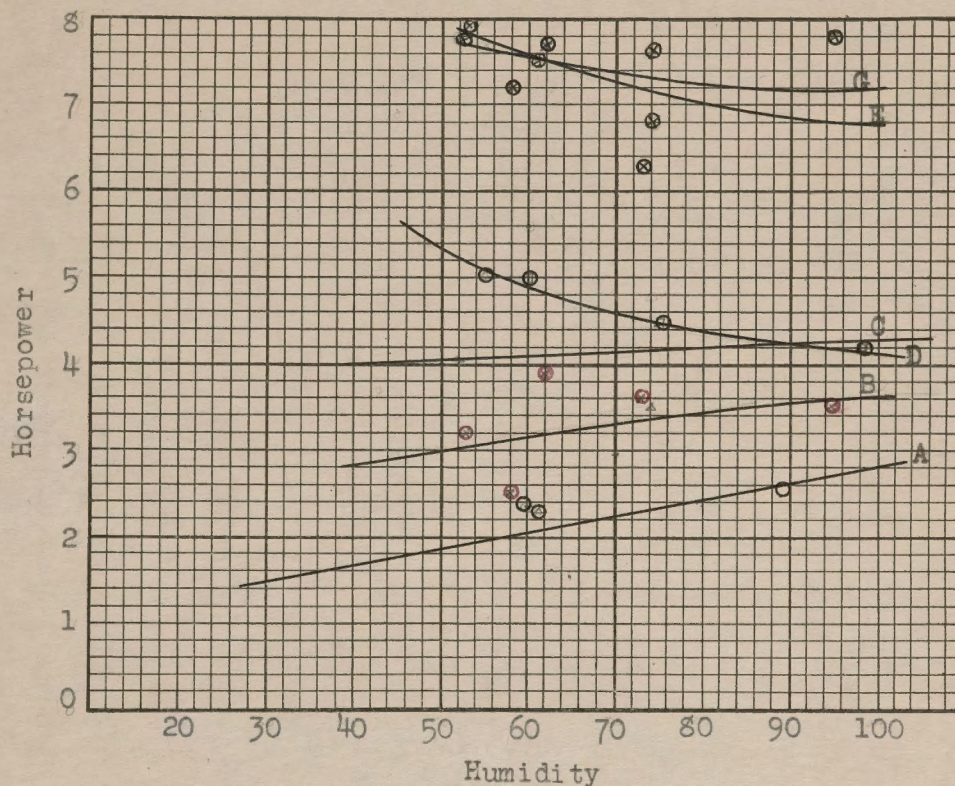


FIG. NO. 8.

TITLE:-

Variation of Horsepower with Humidity, Load
and Air Velocity.

INTERPRETATION OF CURVES

- A - Light Load, Low Speed, Theor. Horsepower
- B - Medium Load, Low Speed, Theor. Horsepower
- C - Medium Load, High Speed, Theor. Horsepower
- D - High Load, High Speed, Theor. Horsepower
- E - Medium Load, Low Speed, Ind. Horsepower
- G - Medium Load, High Speed, Ind. Horsepower

efficiency. Further study that would help to determine this point would be of value to operating refrigeration engineers.

Figure 3 shows that the tons of refrigeration also increases with the humidity in the room. Of course, the tonnage would be expected to increase with the load, but it is of interest that it increases with the humidity. This is due to the fact that the rate of heat absorption increases with the humidity.

Figure 5 shows that best efficiency is to be obtained with the suction side superheated. This is also contrary to the opinion held by the manufacturers. (See Appendix)

The point of best efficiency moves up the superheat scale as the load increases. Thus, it is necessary to vary the suction superheat with the load in order to obtain best performance.

Figure 6 indicates that the rate of heat absorption varies with the humidity of the surrounding air and also with the air velocity. This indicates that for rapid cooling a high fan speed should be used.

The temperature at which the cold room can be held varies with the humidity, air velocity and rate at which heat is being injected into the room, as shown by Figure 7.

From Figure 8 it appears that for low and medium loads the theoretical horse-power required varies directly with the humidity, while the actual horse-power varies inversely with

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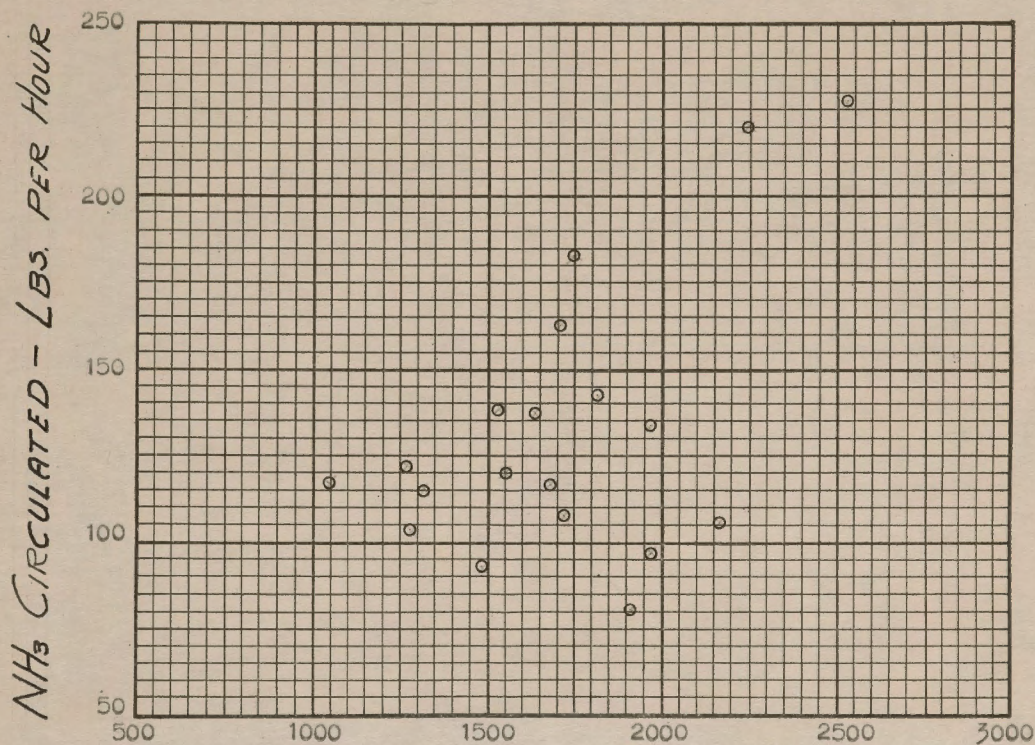


FIG. NO. 9.

TITLE:-

Ammonia Required for Rates of Radiations

INTERPRETATION OF CURVES

Relation indefinite. Further research required.

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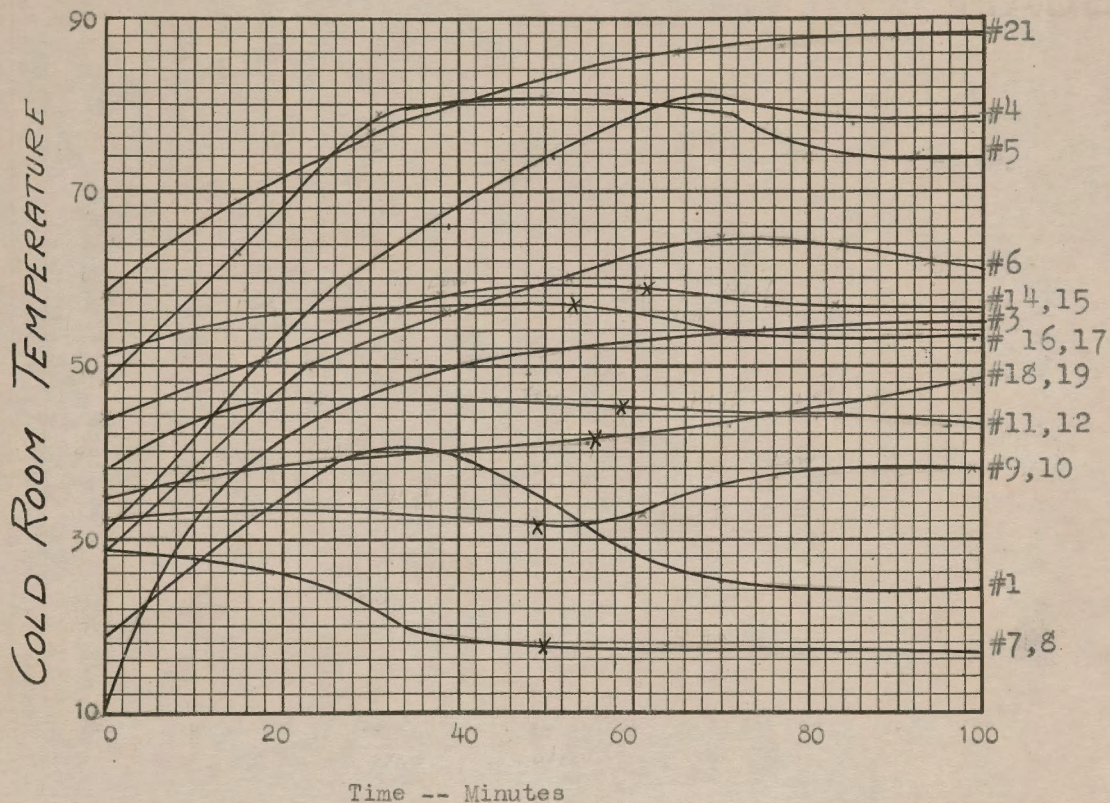


FIG. NO. 10.

TITLE:-

Change of Temperature With Time Under Different Conditions.

INTERPRETATION OF CURVES

Explanation of curves given on the following page.

TIME TEMPERATURE CURVE.

Interpretation:

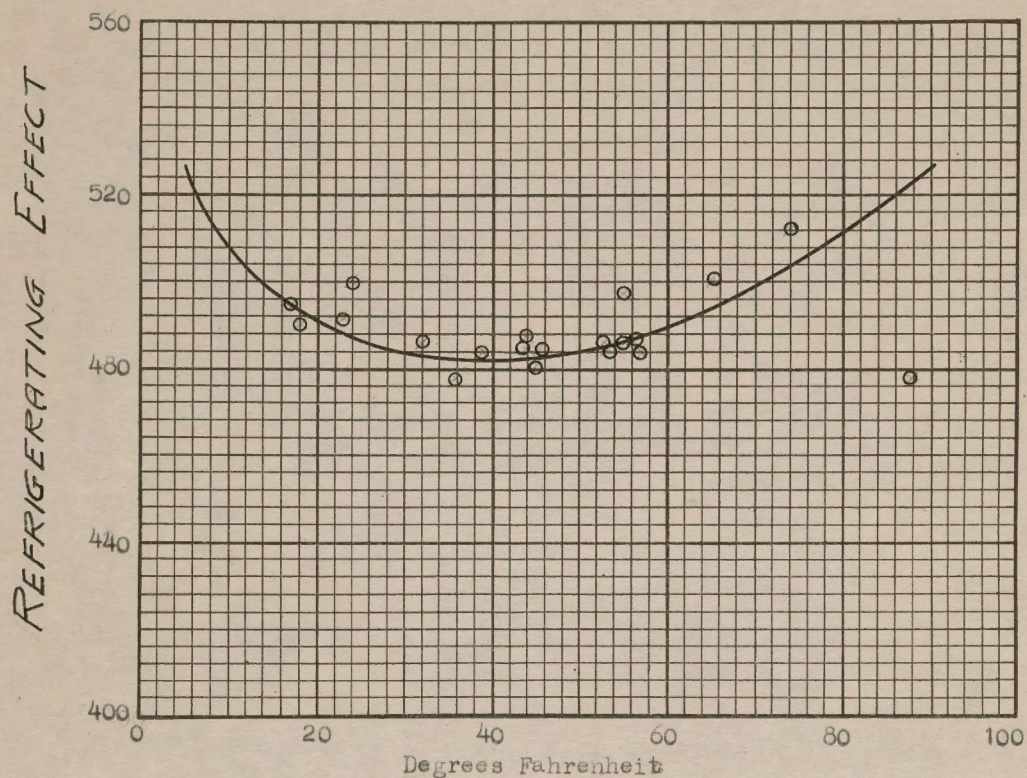
#1	Light Load	Slow Speed	Humidity 60
#3	Heavy Load	Slow Speed	Humidity 43
#4	Very Heavy Load	High Speed	Humidity 98
#5	Very Heavy Load	High Speed	Humidity 75
#6	Heavy Load	High Speed	Humidity 60
#7,8	Light Load	Low & High Speed	Humidity 63
#9,10	Medium Load	High & Low Speed	Humidity 56
#11,12	Medium Load	Low & High Speed	Humidity 55
#14,15	Heavy Load	Low & High Speed	Humidity 42
#16,17	Medium Load	Low & High Speed	Humidity 68
#18,19	Medium Load	High & Low Speed	Humidity 74
#21	Light Load	No Speed	Humidity 25

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Temperature of Cold Room

FIG. NO. 11.

TITLE:-

Relation of Refrigerating Effect to the Cold Room Temperature

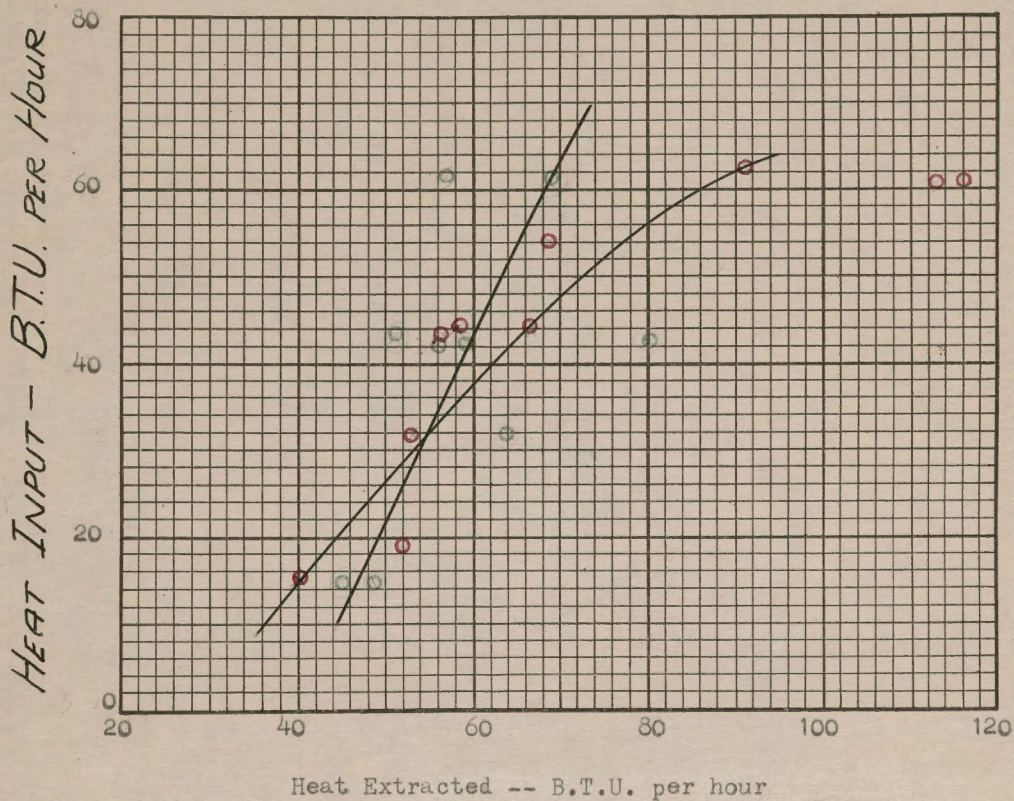
INTERPRETATION OF CURVES

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FIG. NO. 12.

TITLE:-

Relation of Heat Input to Heat Extracted

INTERPRETATION OF CURVES

- Operation of fan at low speed.
- Operation of fan at high speed.

the humidity. For very heavy loads the theoretical horsepower also varies inversely with the humidity. The probable reason for this is that for very heavy loads the compressor operates more efficiently.

Figure 10 shows the time required to bring the cold room to a constant temperature under specific conditions. These curves were plotted directly from observations made and show the actual performance of the machine.

The curve indicated by Figure 11 has very little practical value due to the fact that neither variable is dependent upon the other. Further study, using thermostatic control of the ammonia circulated, would probably reveal a relation between refrigerating effect and cold room temperature.

Figure 12 shows the relation of measured heat injected into the room to the heat extracted.

For the room to remain at a constant temperature it is evident that the heat extracted must be equal to the heat input. Therefore, the difference of these two, as indicated by the curves, represents the amount of heat leaking into the room from outside sources.

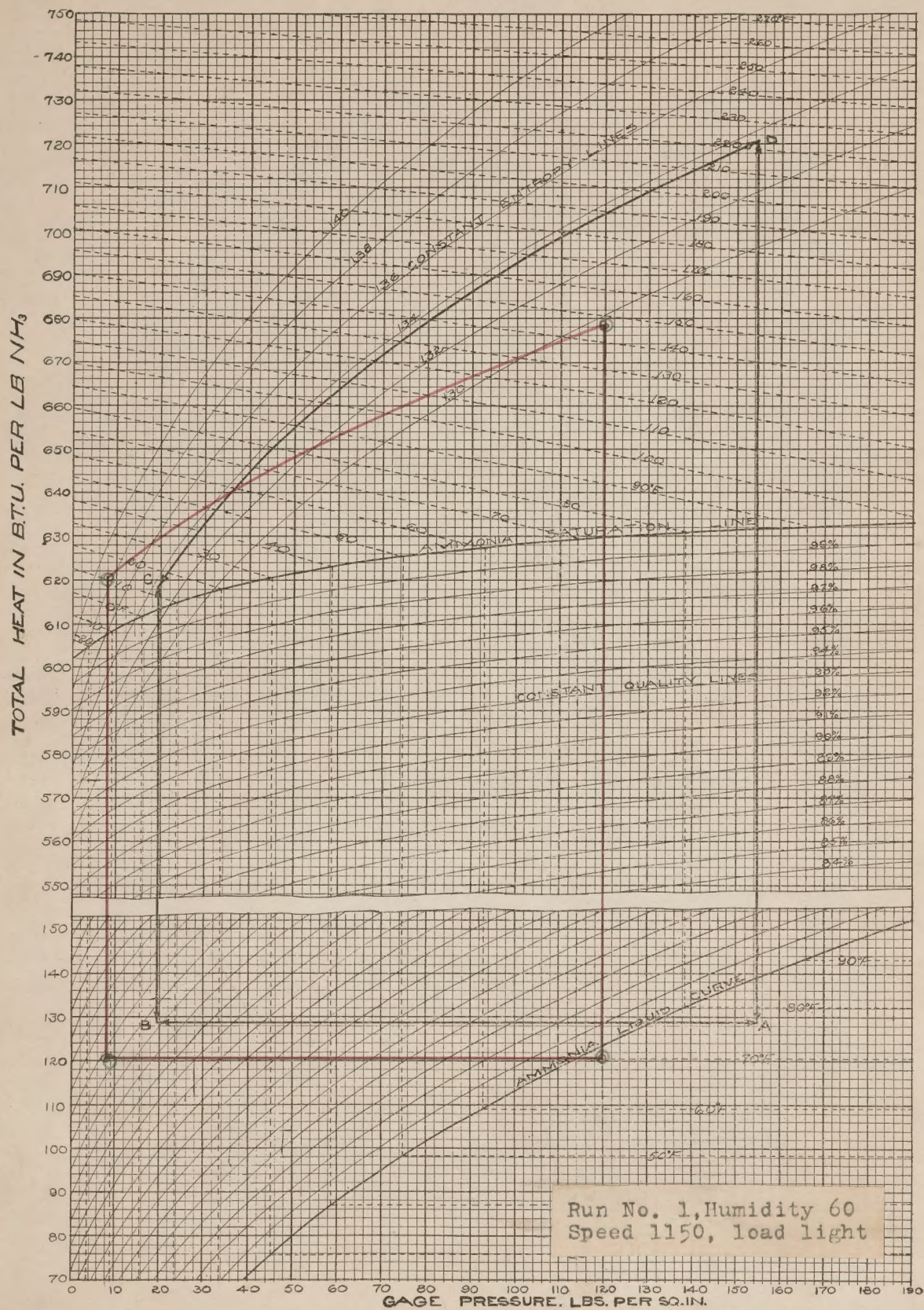
CONCLUDING STATEMENTS

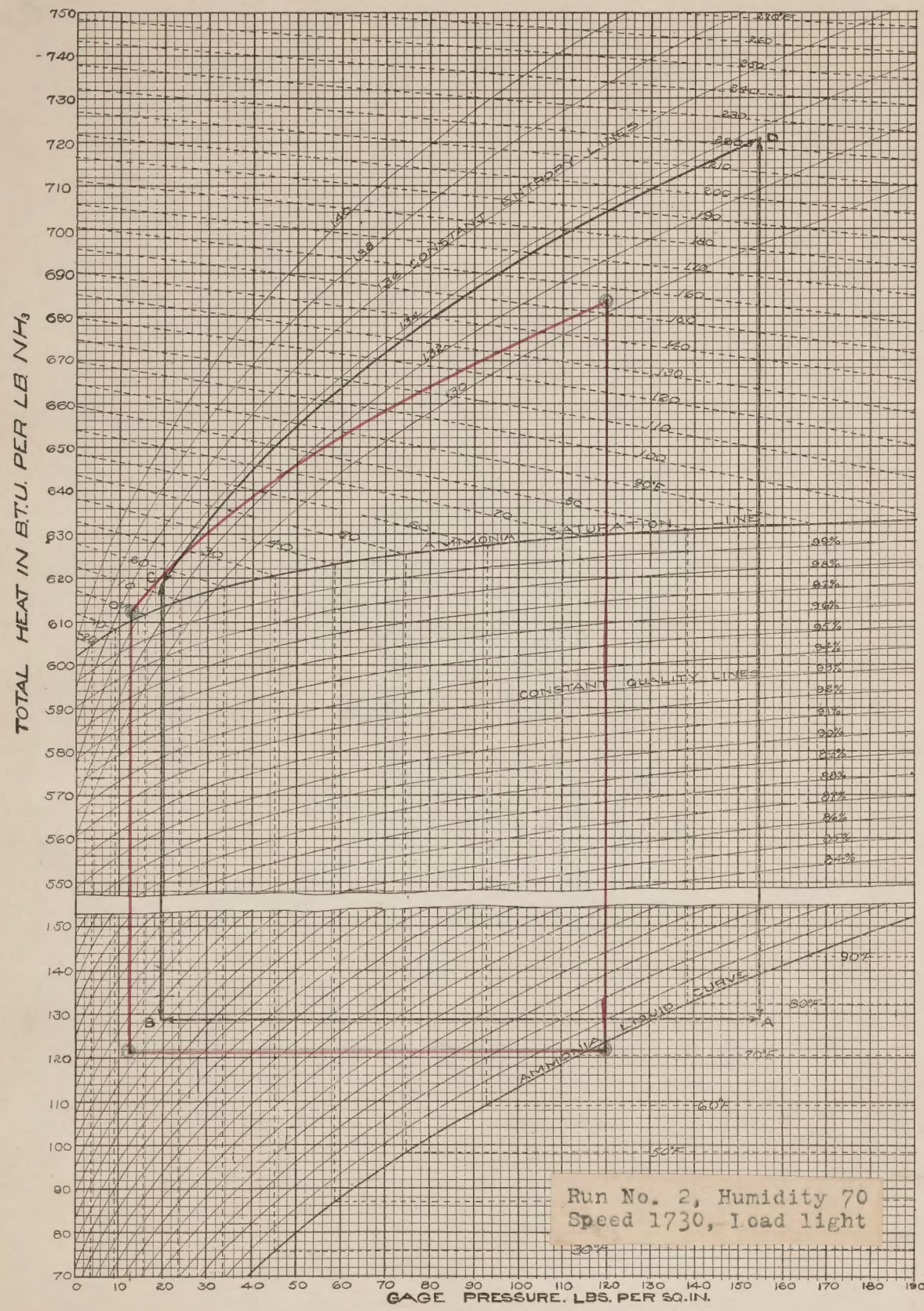
It can be seen from the results given that this has been a general study of all of the chief characteristics of the air cooling system. Many difficulties have been overcome, but there are many more yet to be eliminated. Each of the characteristics indicated above can well be made the subject for an entire thesis study. Since, as stated in the introduction, very little has been written on this subject, this paper may be used as a basis or foundation for further study. The results of this study indicate the general trends of the characteristics, but are not sufficient to establish them. Therefore, with this paper as a basis, and further research along the lines indicated herein, this study may become of great value.

It is suggested that a study be made of the operation with all conditions held constant except the coefficient of performance. This would indicate some things about which nothing has been written. A study of the effects of superheating in the discharge would bring out some important relationships. In this study little attention was paid the jacket water, but it is found to be important to make a study of it. Studies should be made using different kinds, or temperatures of cooling water in the condenser and jacket. Studies should also be made using different pressure ranges. In fact, after a study of this kind has been made, many things appear which should be studied.

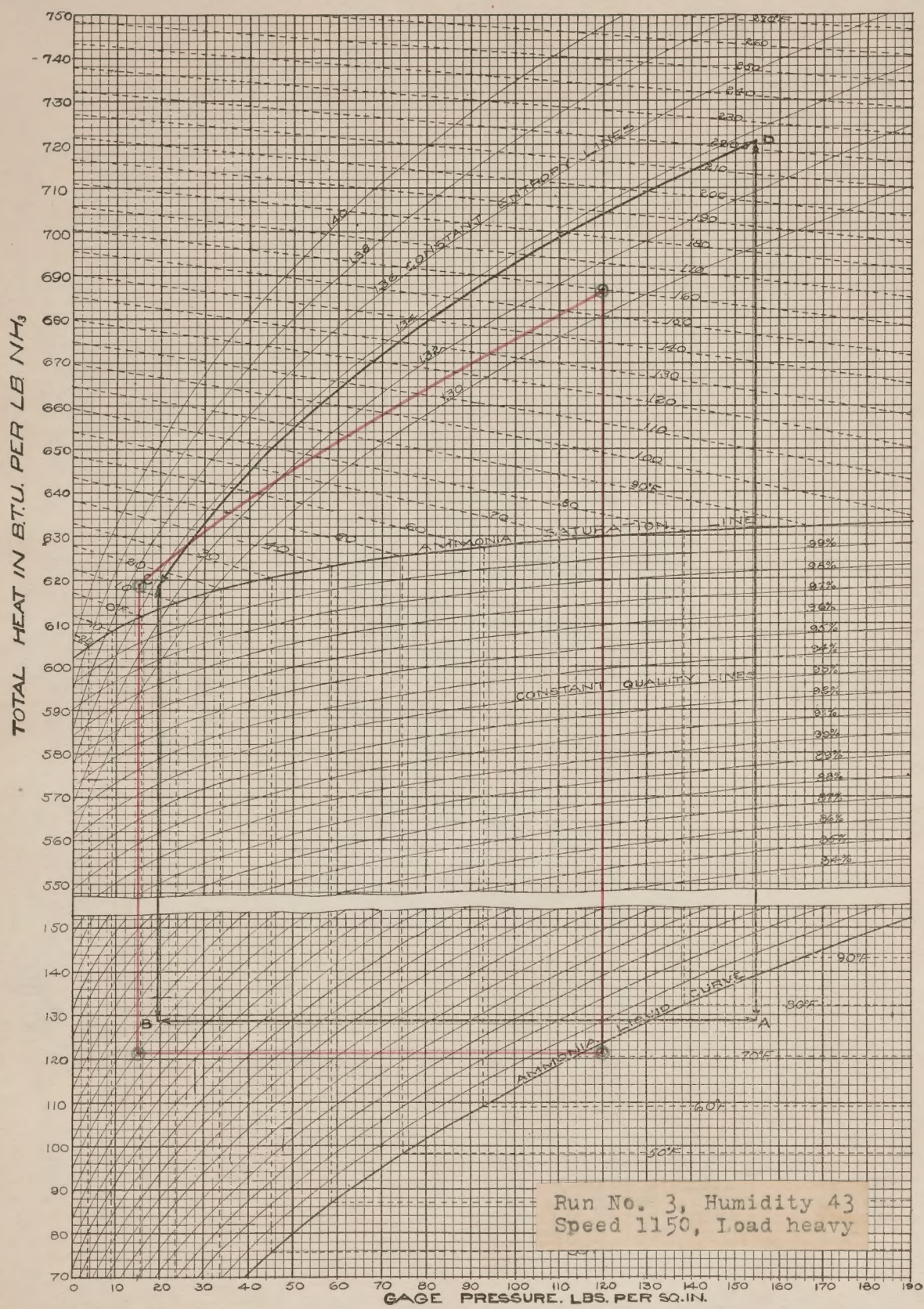
Further study, using float control and thermostatic control, would be interesting, and is desired by the manufacturers of this equipment.

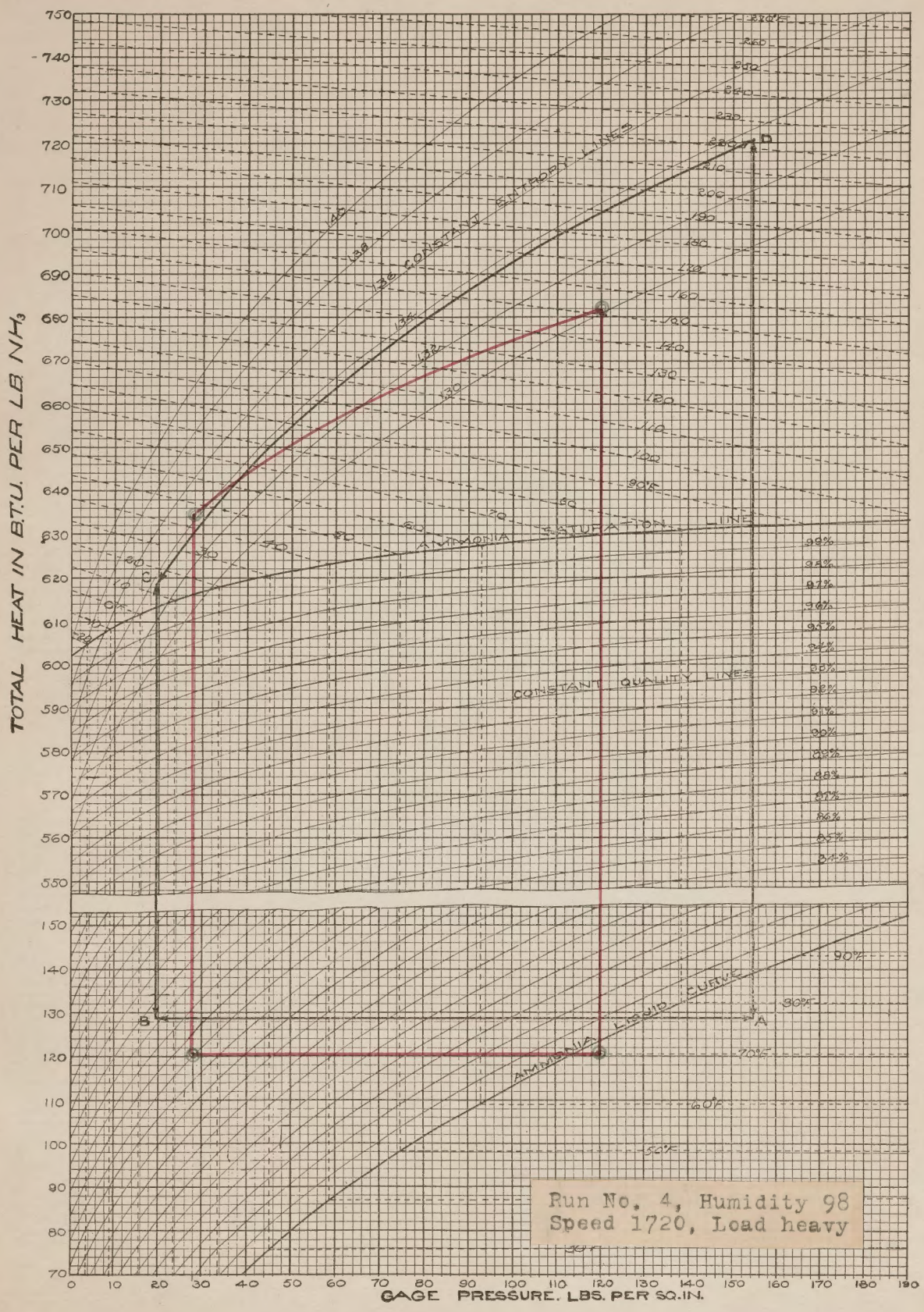
Thus, it can be seen that only a start has been made in the study of air cooling systems.

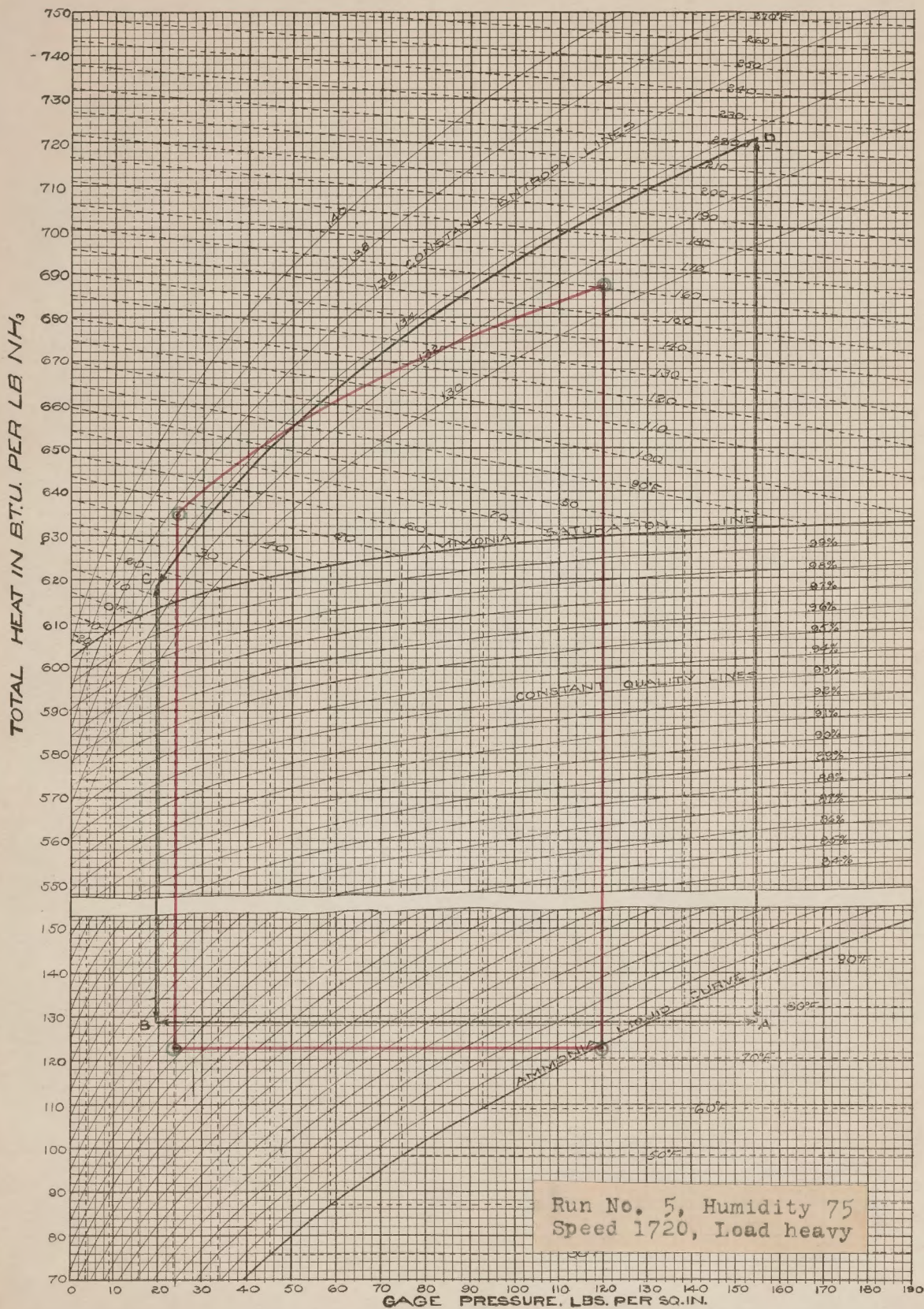


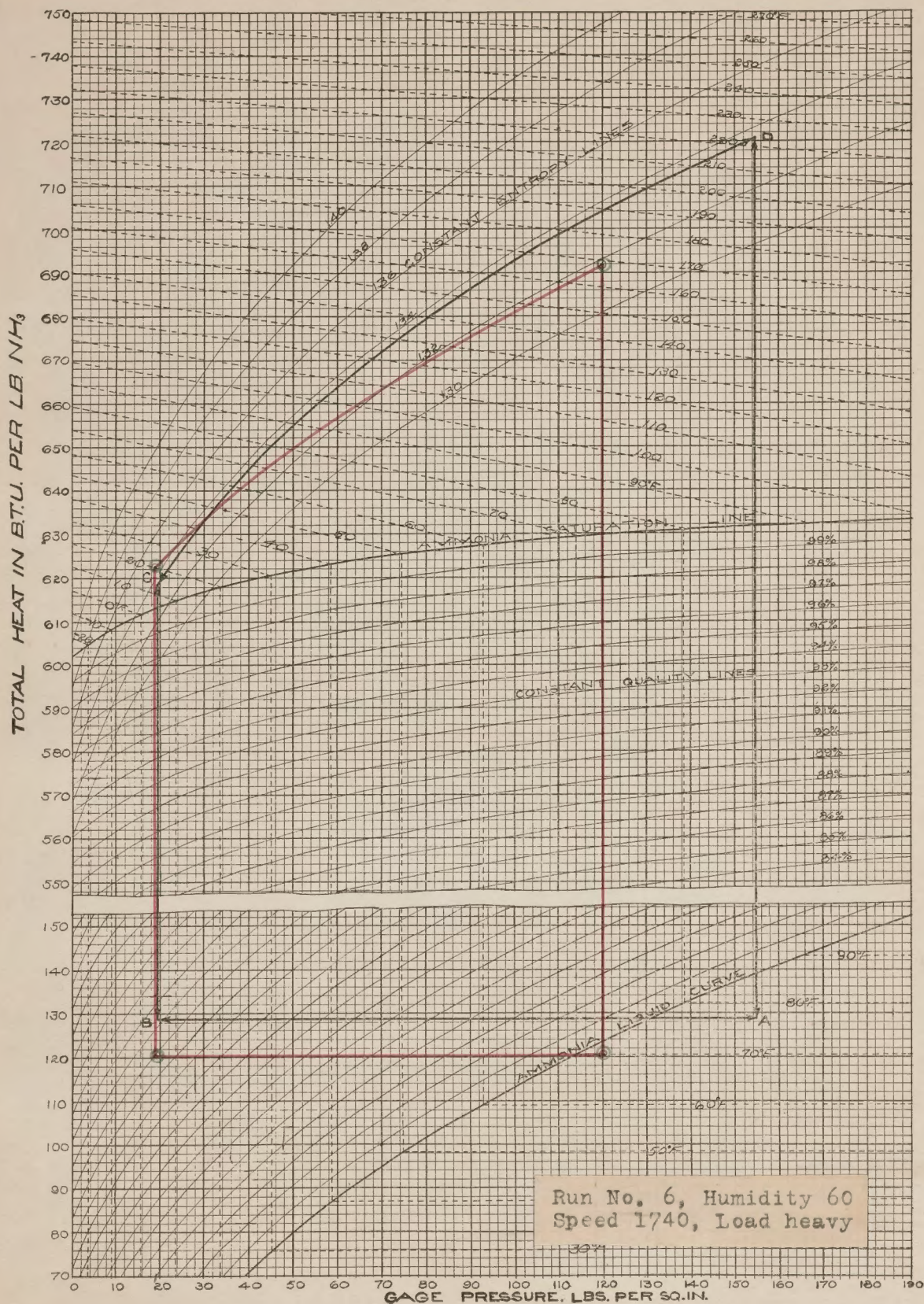


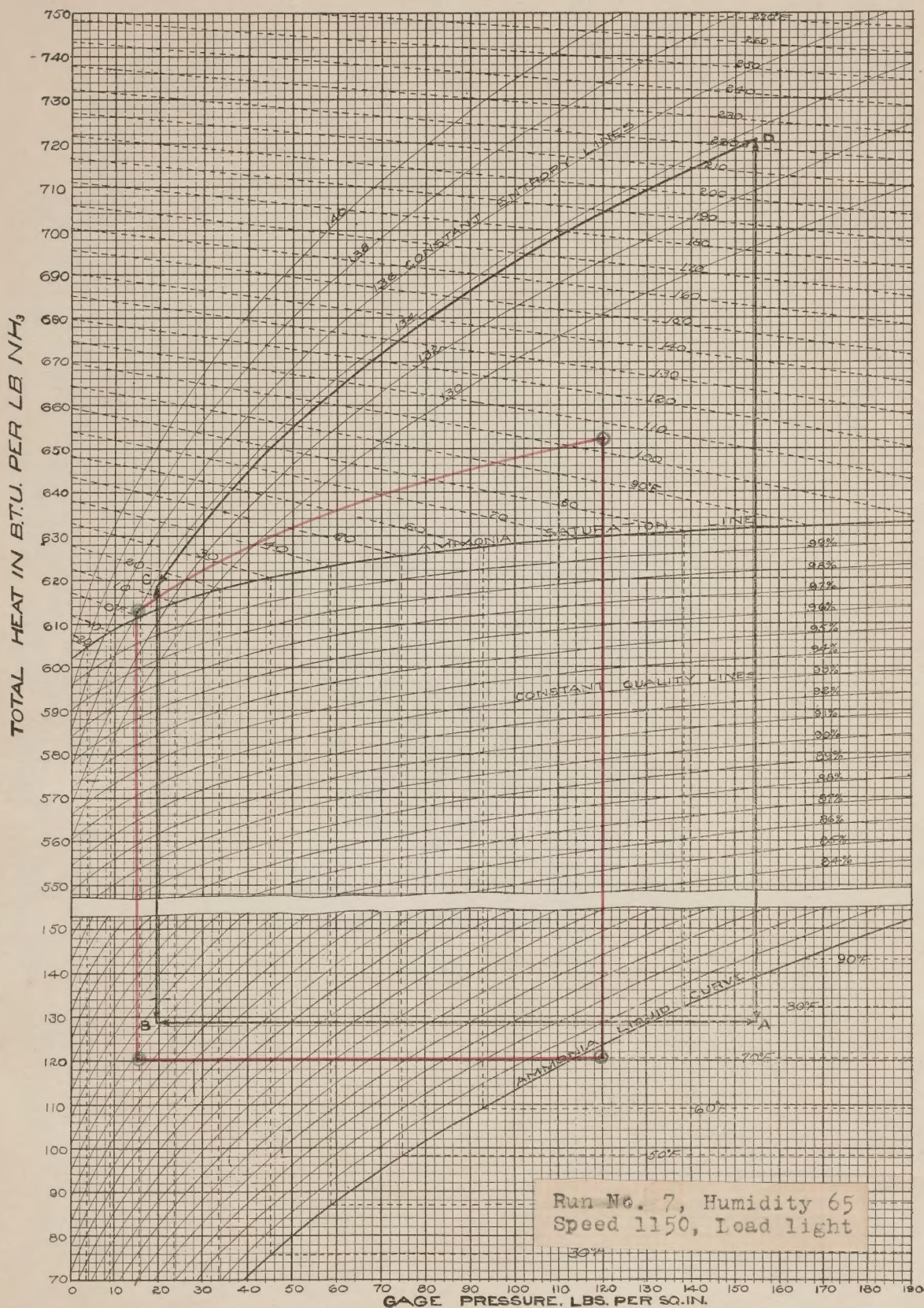
Run No. 2, Humidity 70
Speed 1730, load light

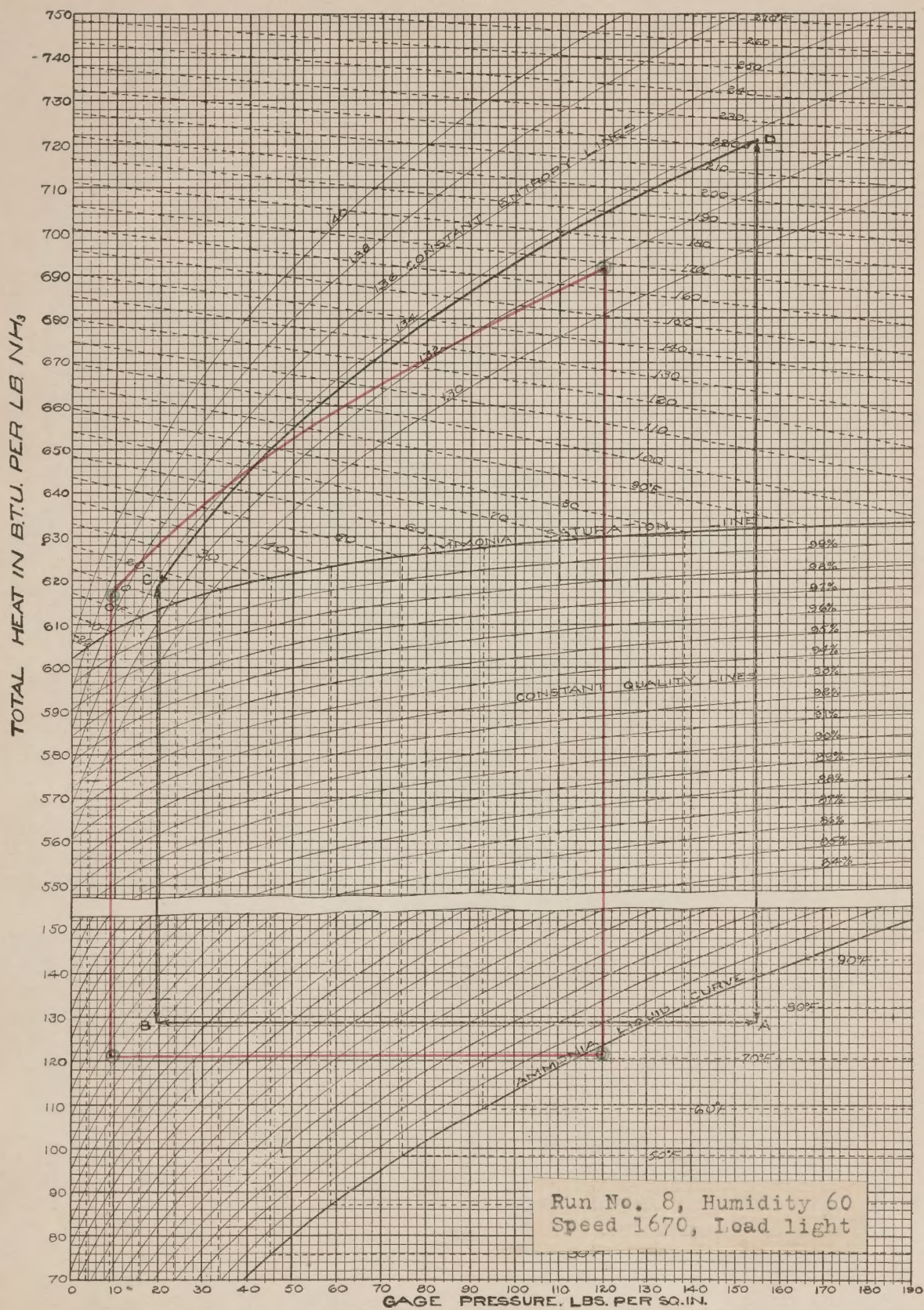


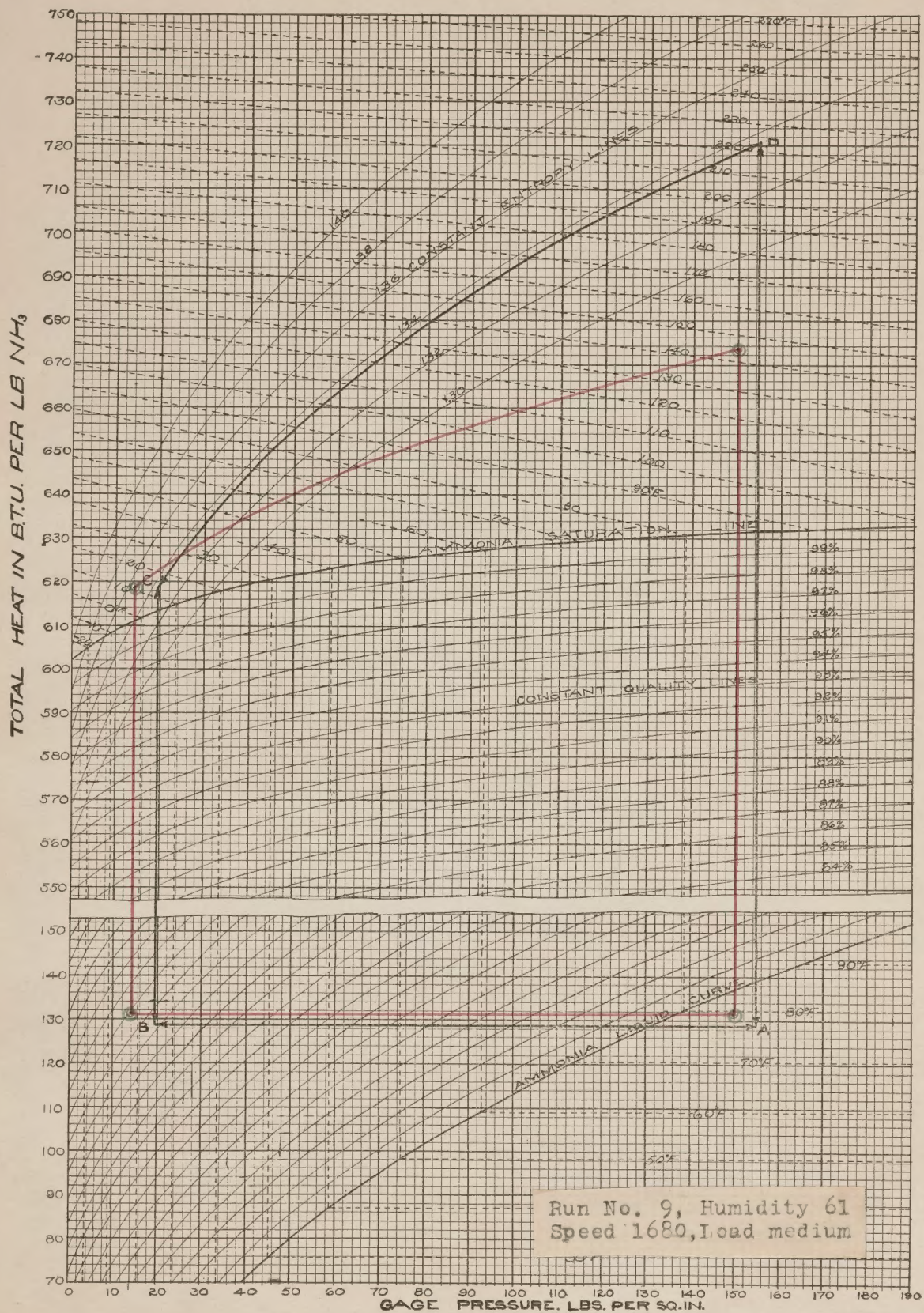


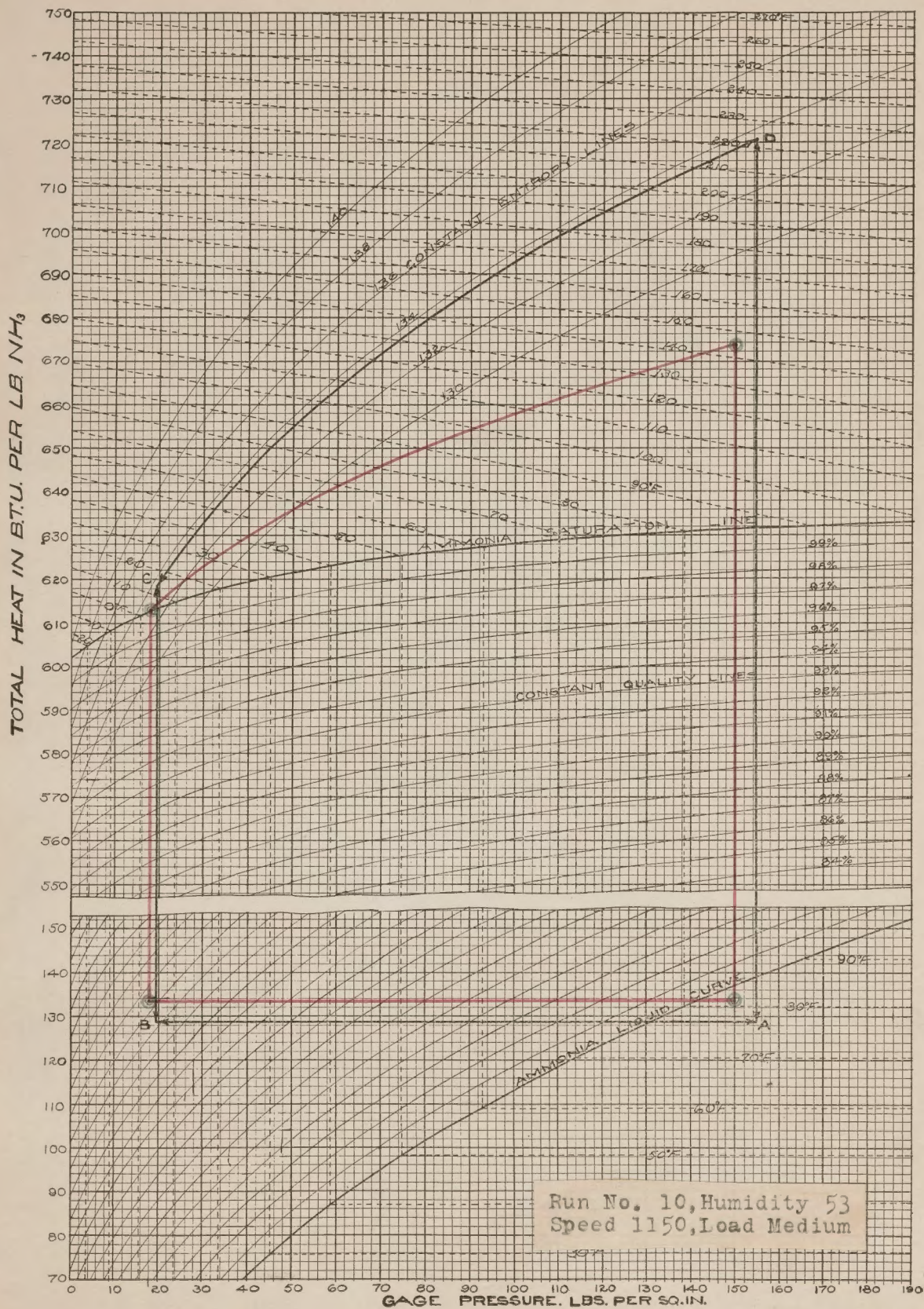


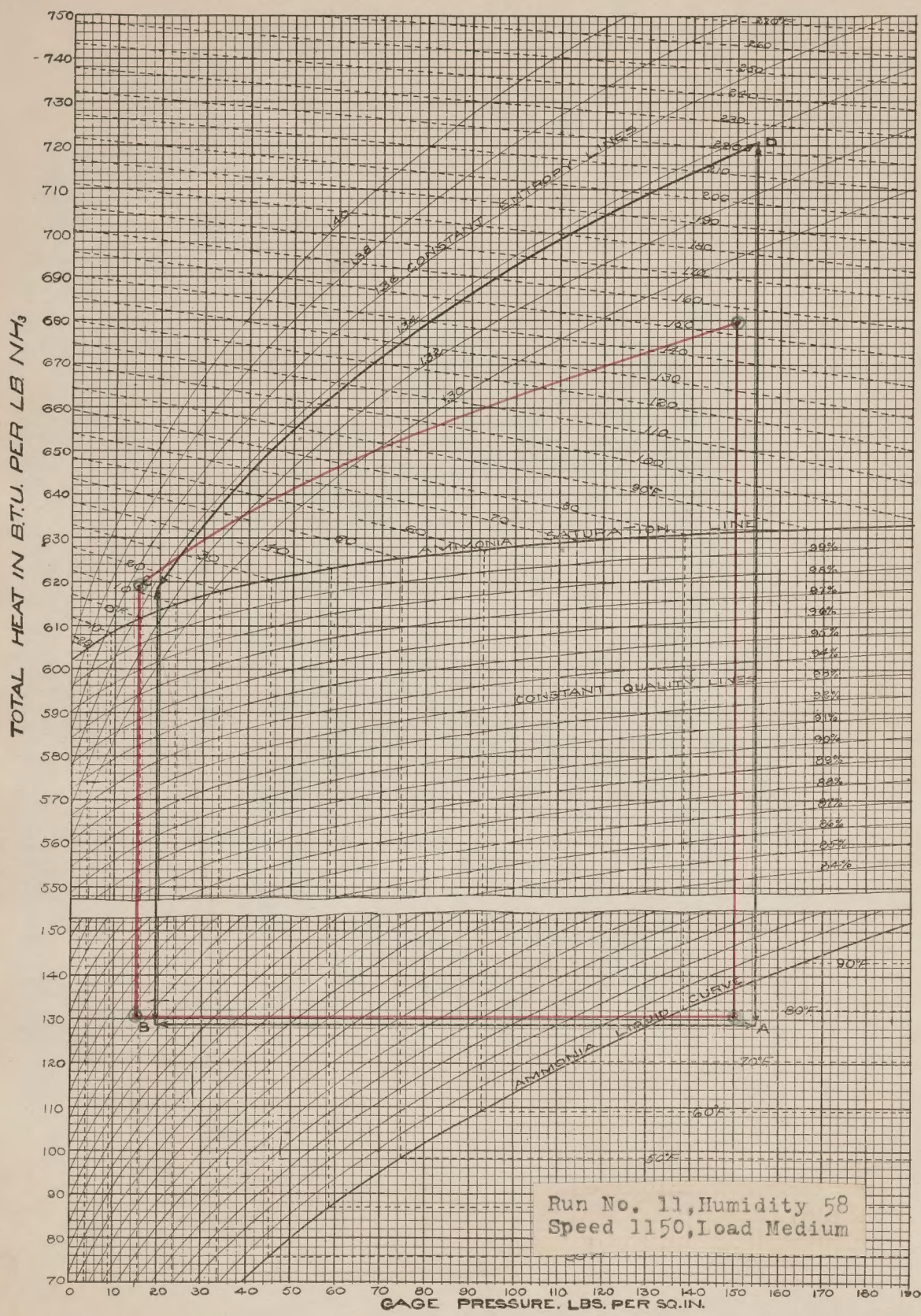


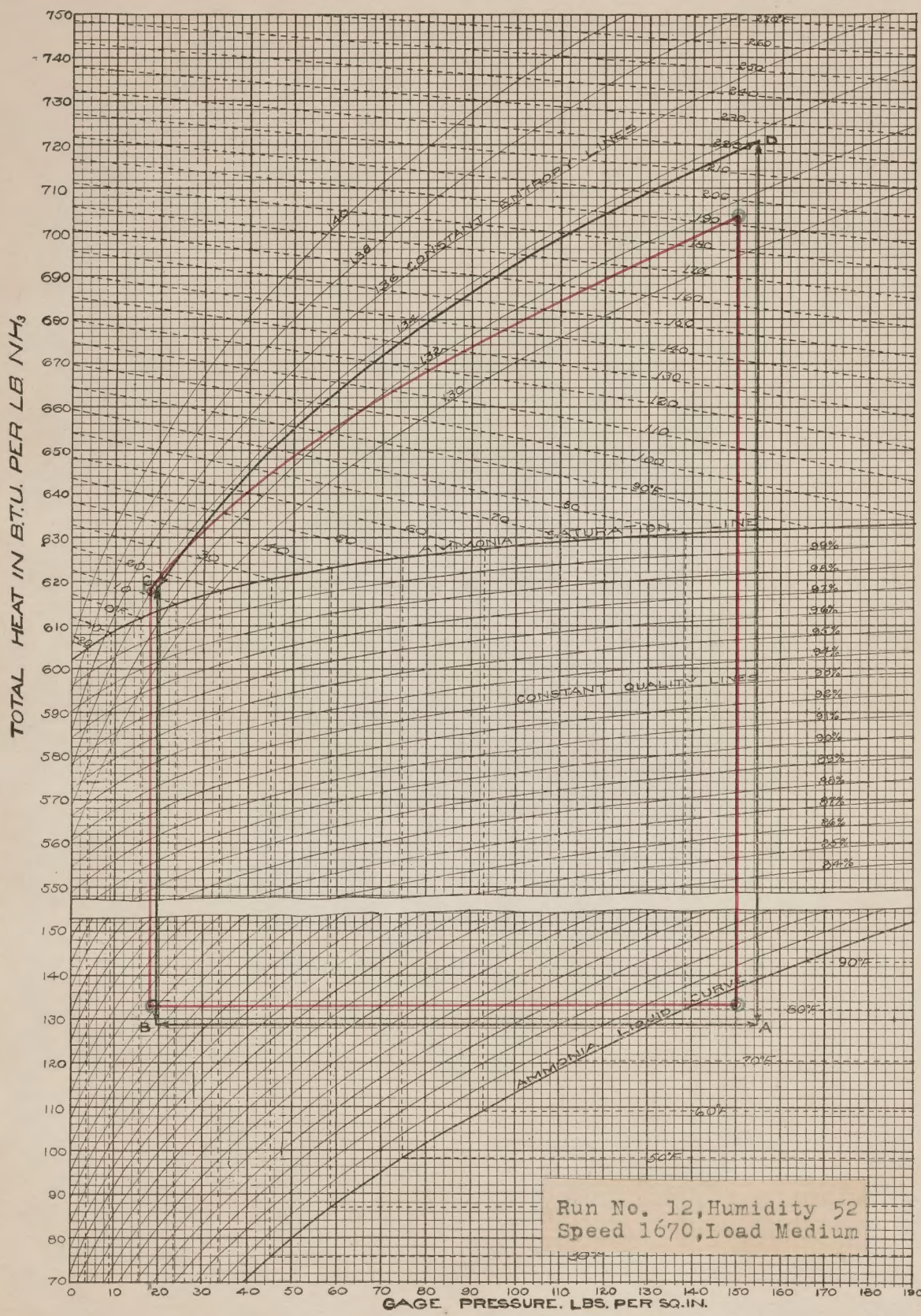


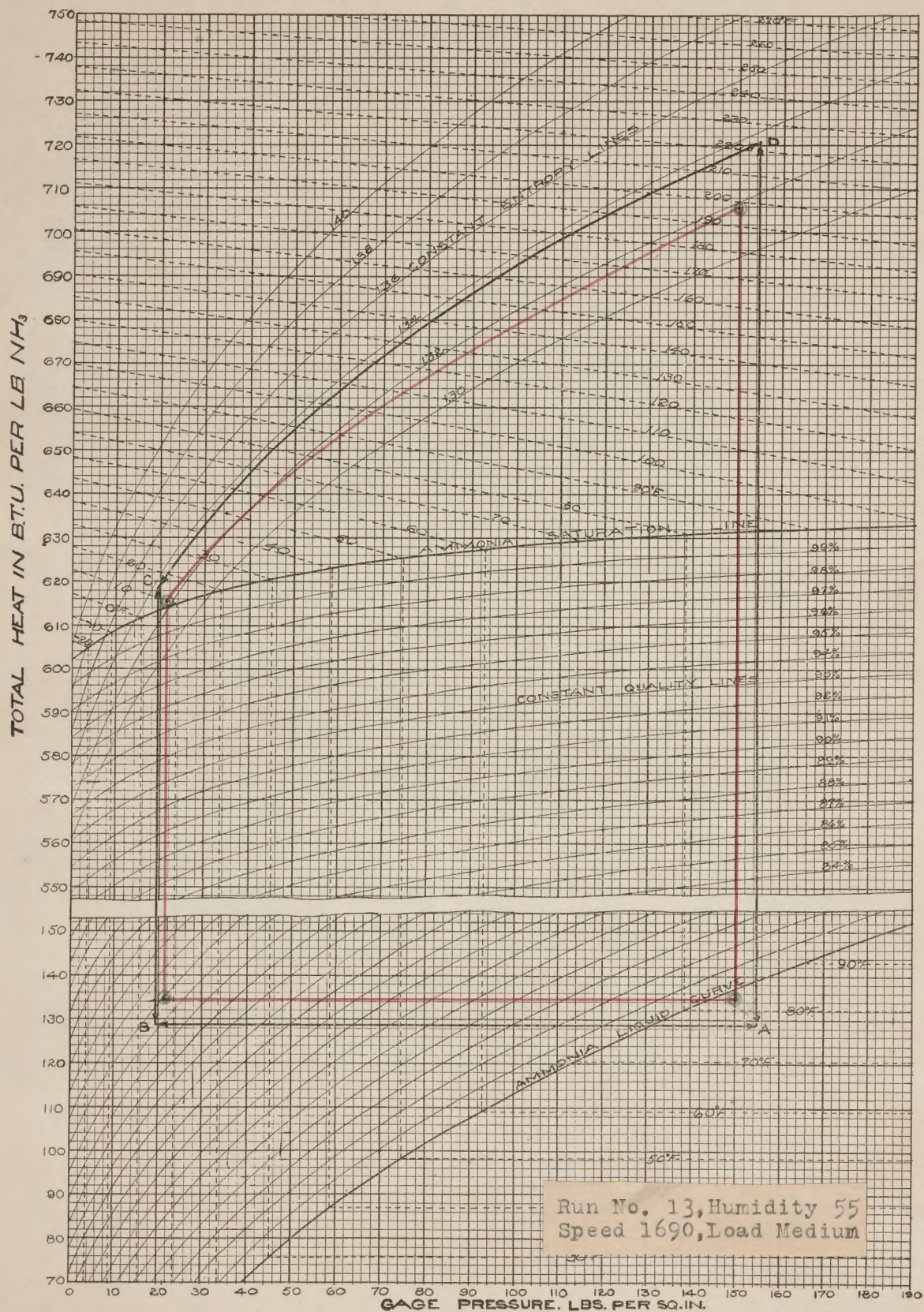


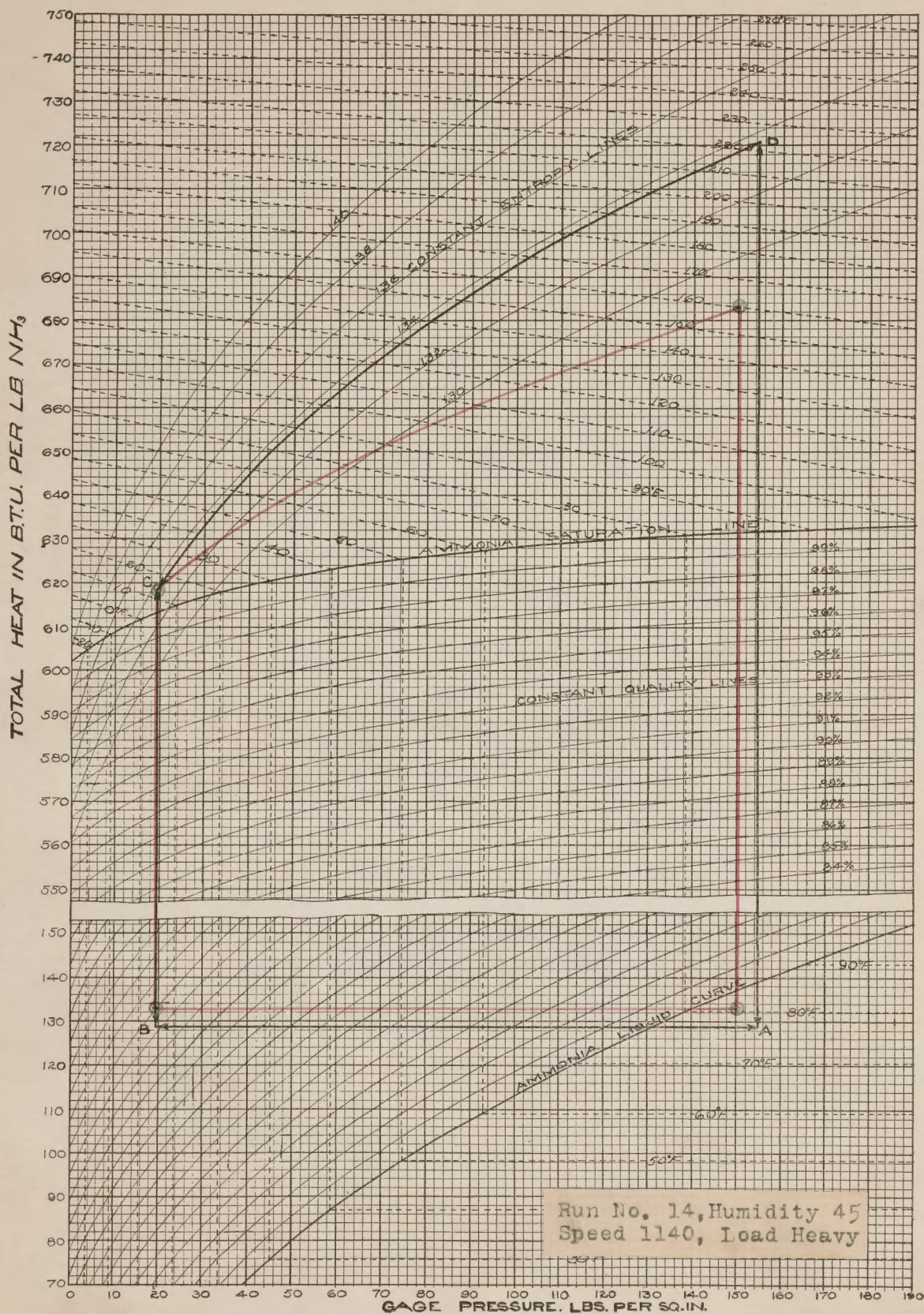


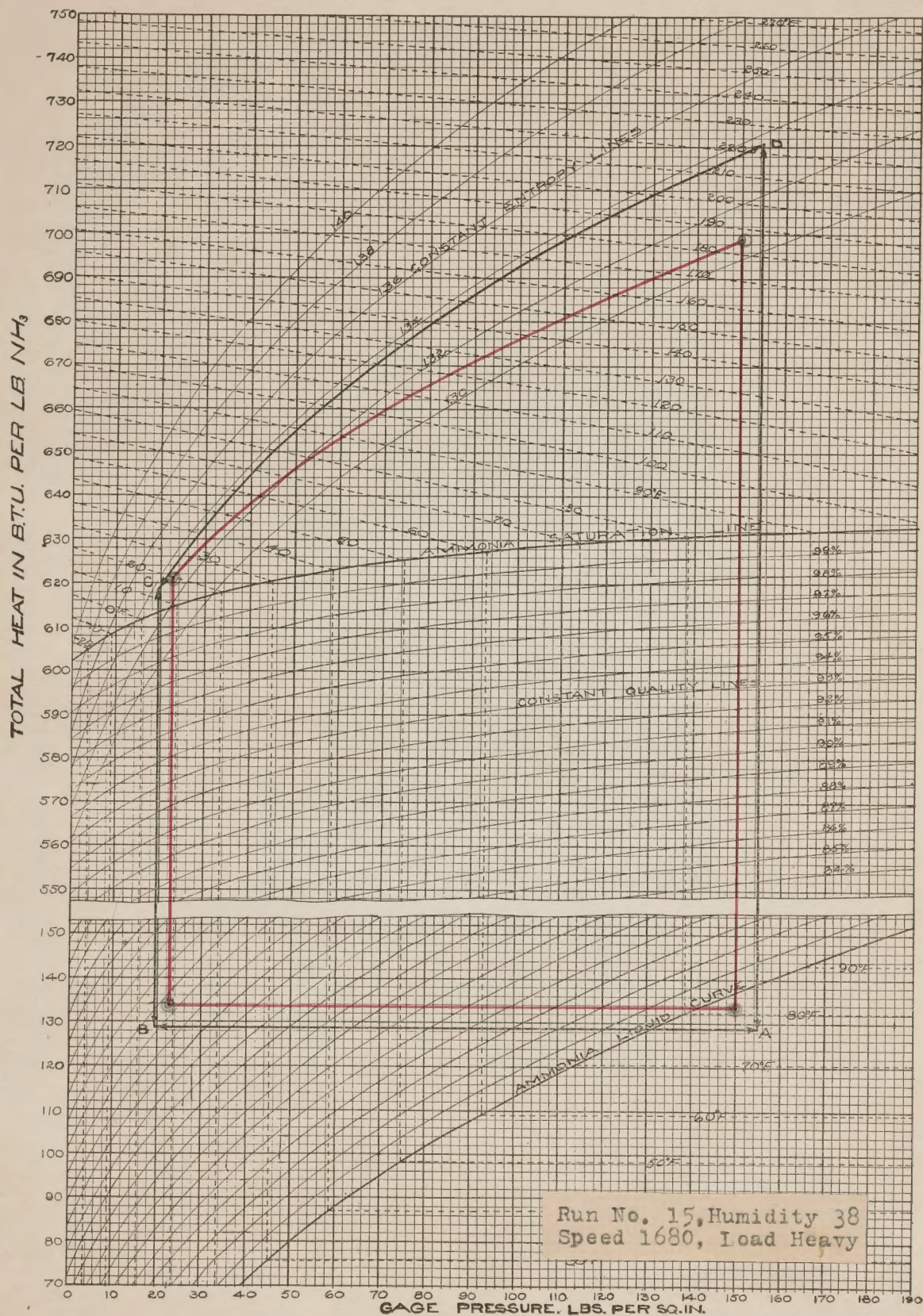


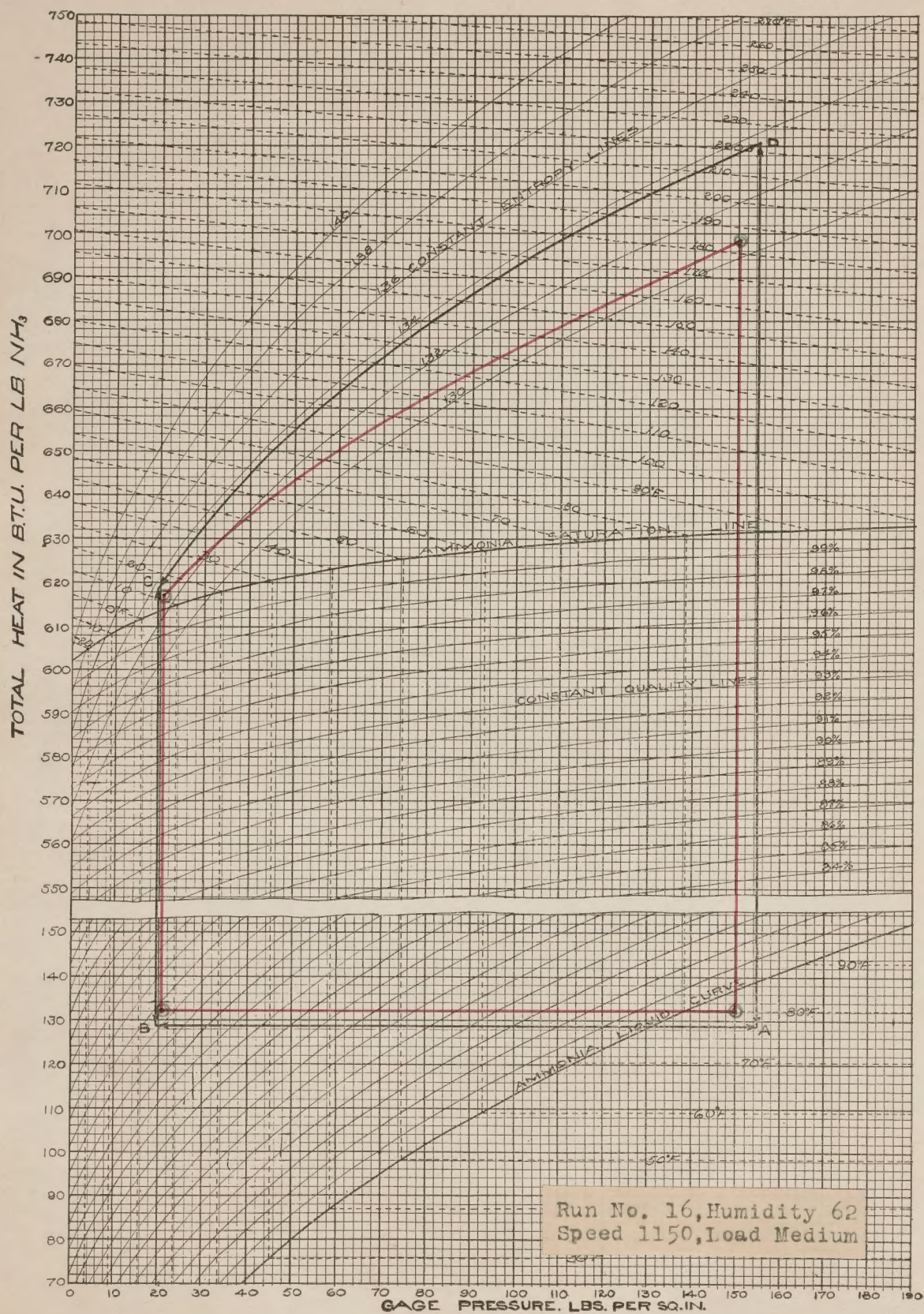


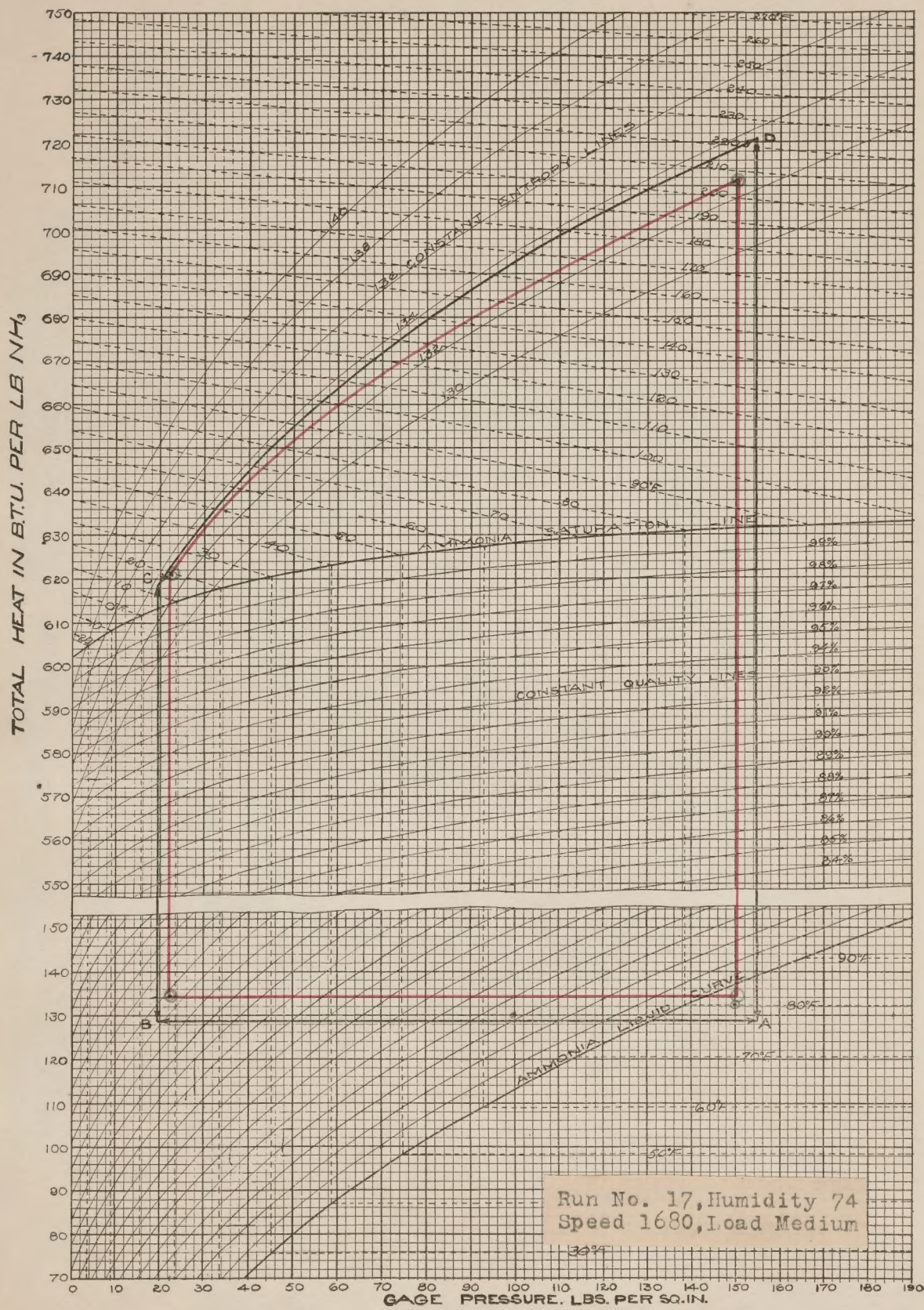


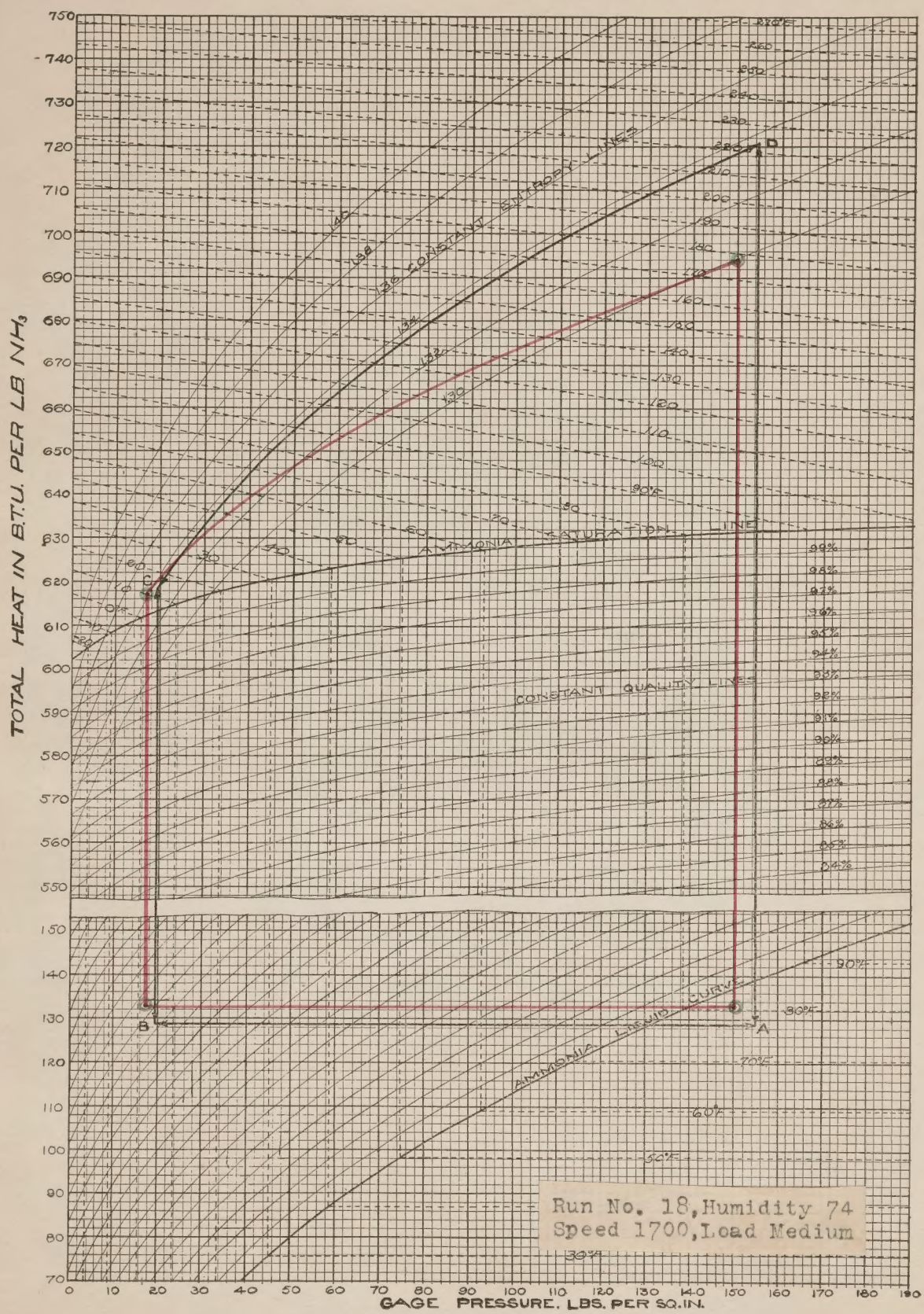




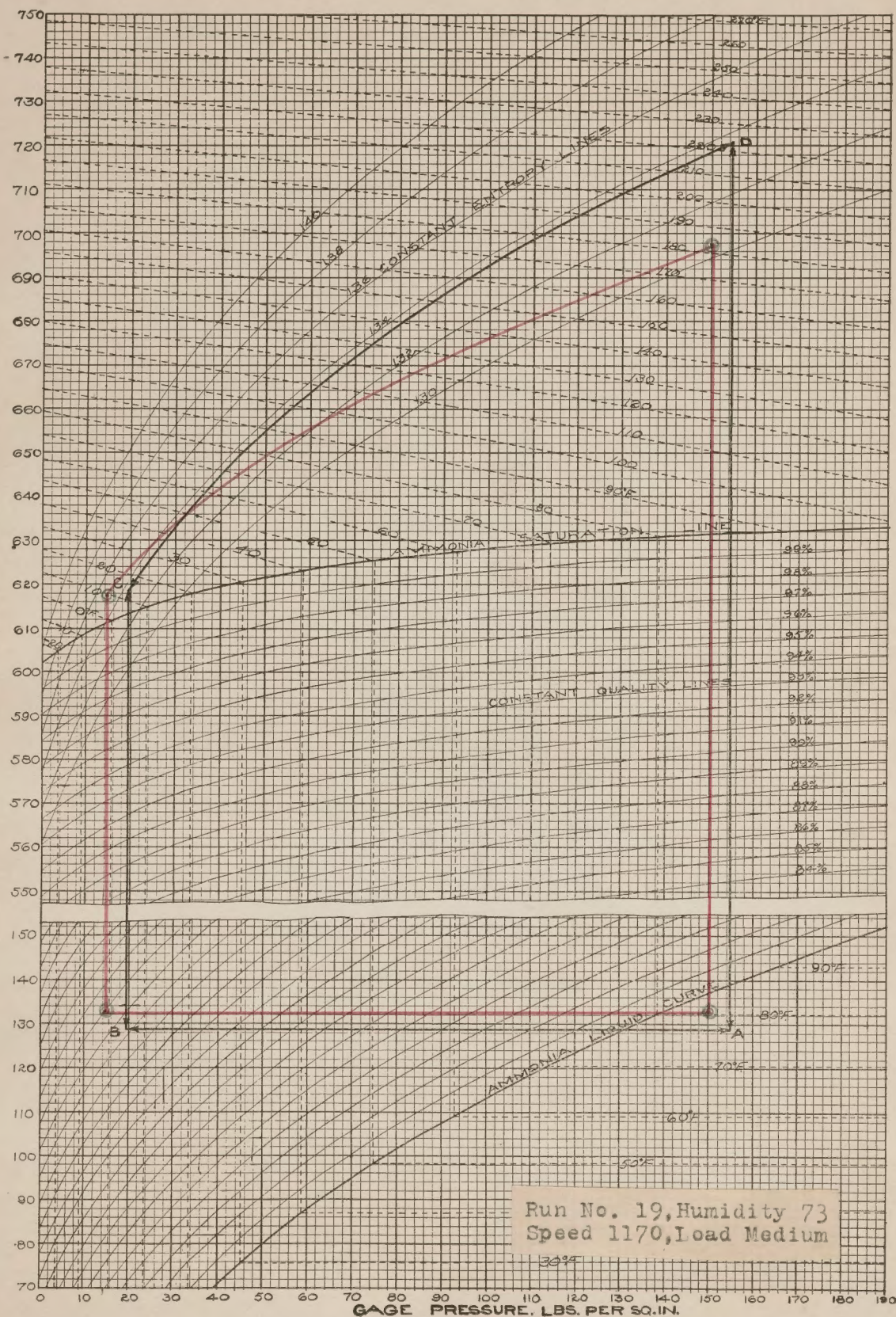


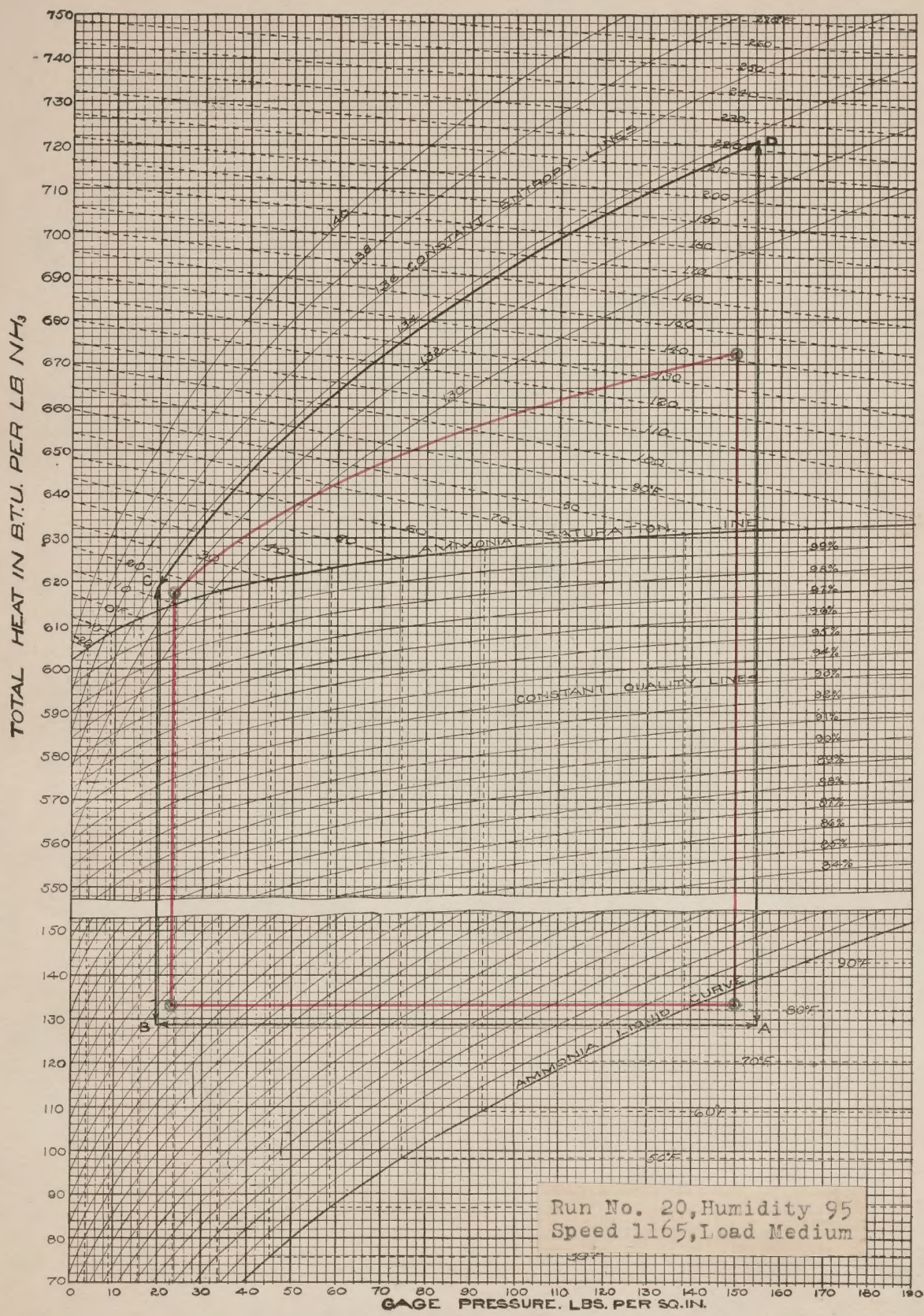


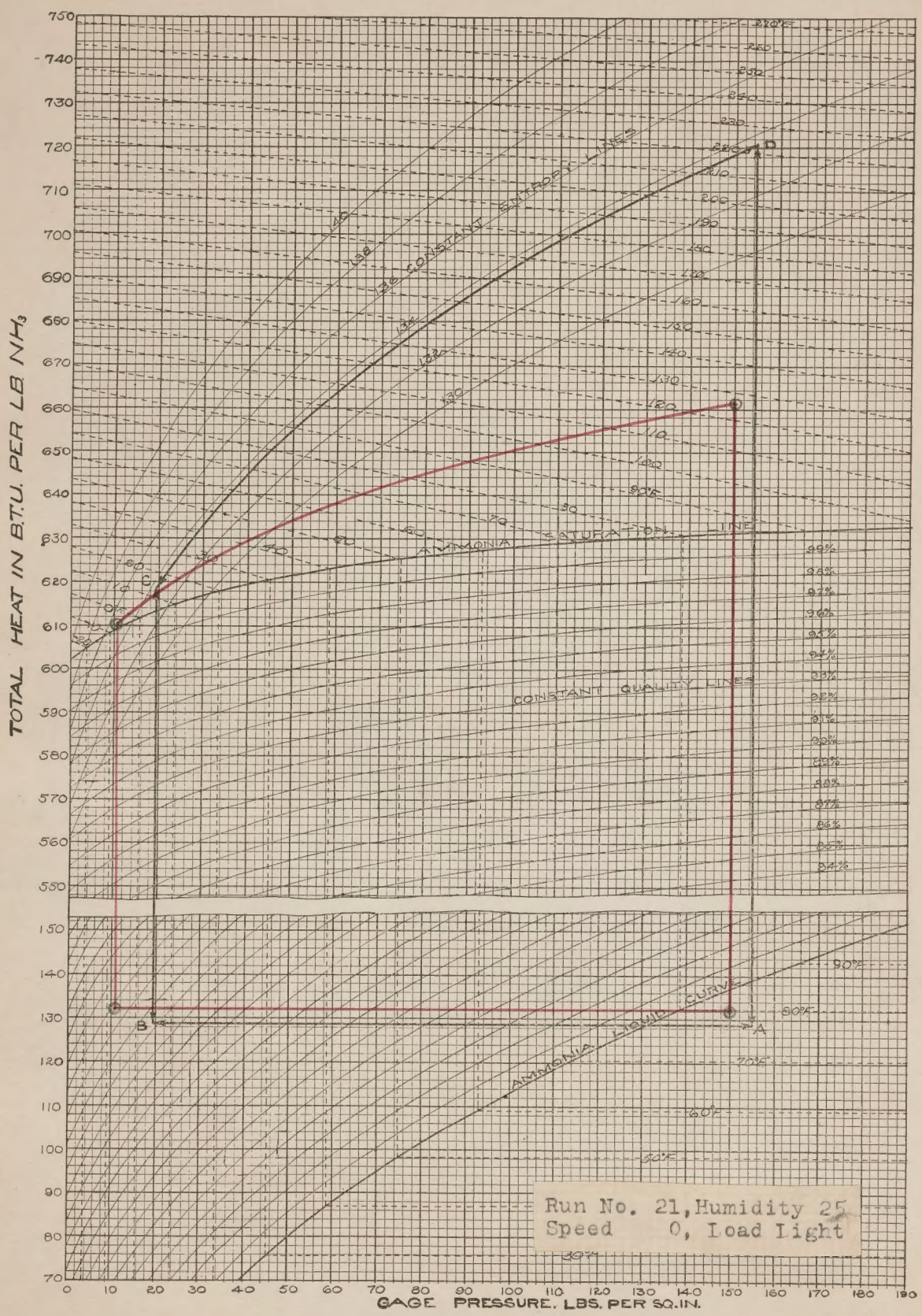




Run No. 18, Humidity 74
Speed 1700, Load Medium







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by

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A P P E N D I X

Section 2
Sheet 9
2/1/30

Basic	B.T.U.	Rating	#1522	at	1750	R.P.M.	-	6,860	B.T.U.	Per	Hour
"	"	"	"	"	1160	"	"	5,450	"	"	"
"	"	"	"	"	1750	"	"	13,350	"	"	"
"	"	"	"	"	1160	"	"	10,650	"	"	"
"	"	"	"	"	1160	"	"	17,020	"	"	"
"	"	"	"	"	870	"	"	14,350	"	"	"
"	"	"	"	"	1160	"	"	28,050	"	"	"
"	"	"	"	"	870	"	"	23,150	"	"	"

The above basic ratings are sensible heat transfer based on 5° mean effective temperature difference and a FLOODED AMMONIA SYSTEM.

Where Brine is the refrigerant, a correction must be made for Rate of Brine Flow. See Plot I (Curve No. C-104).

FROST: Above capacities are based on a clean, or but lightly frosted coil. A heavy frost will reduce the capacity of the unit, depending upon the thickness and conductivity of the frost.

To obtain the Sensible Heat capacity of a unit under room conditions, find the Condition Factor from the following table which corresponds to the desired room temperature and mean refrigerant temperature in the coil; and then multiply the Basic Rating of the unit by this Factor.

In selecting refrigerant temperatures for table below-

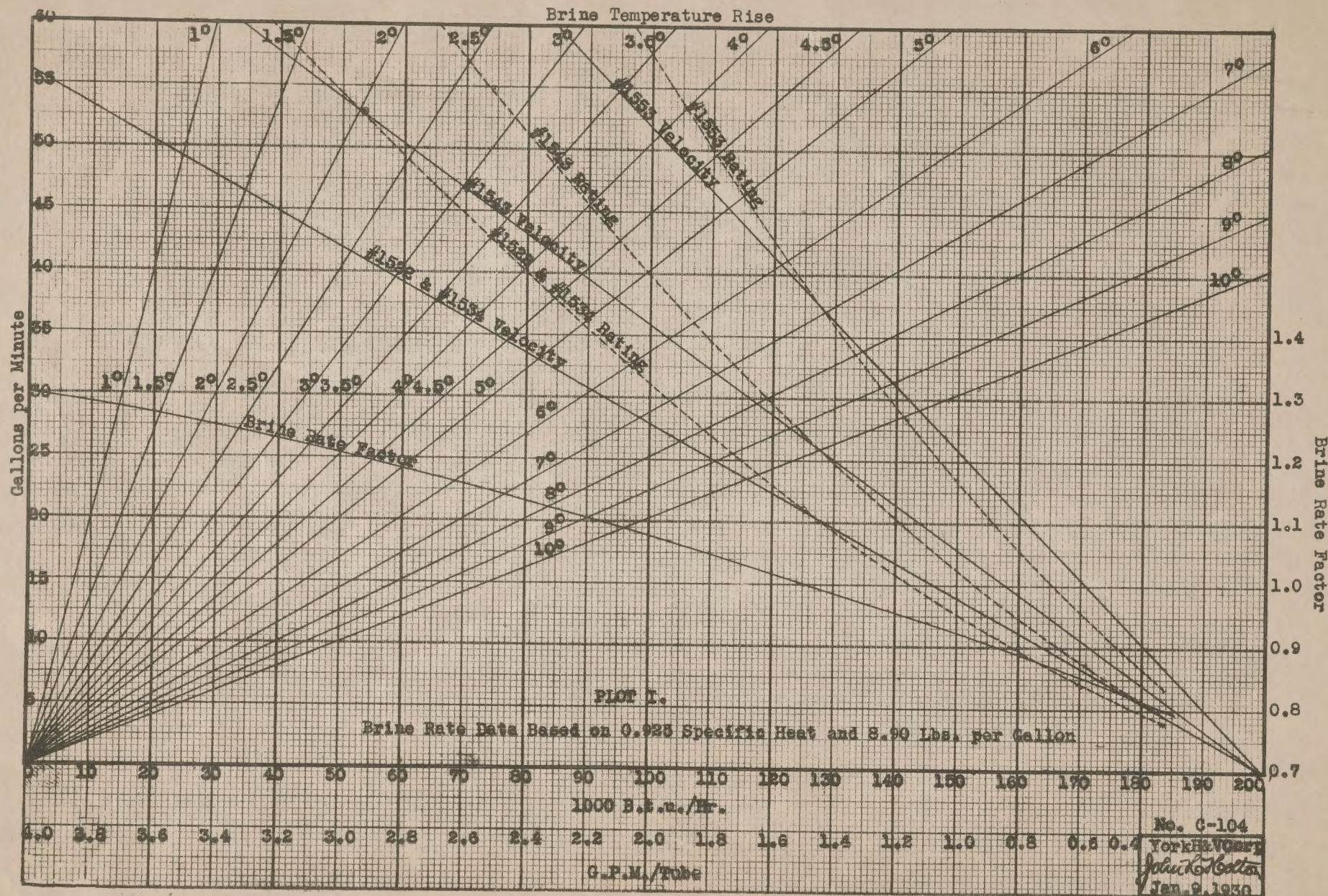
- (a) When ammonia system is employed, use entering ammonia temperature.
- (b) When brine is used, estimate brine temperature rise in coil, and thus determine and use average brine temperature.

CONDITION FACTORS

Room Temp.	AVERAGE TEMPERATURE OF REFRIGERANT (Brine or Ammonia) IN COIL																					
	+53°	+51°	+49°	+47°	+45°	+43°	+41°	+39°	+37°	+35°	+33°	+31°	+29°	+27°	+25°	+23°	+21°	+19°	+17°	+15°	+13°	
70°	2.96	3.30	3.64	3.99	4.34	4.68	5.02	5.37	5.72	6.05	6.40	6.74	7.08	7.42	7.76	8.10	8.44	8.78	9.12	9.46	9.80	
65°	2.06	2.40	2.74	3.08	3.43	3.77	4.12	4.47	4.81	5.16	5.51	5.85	6.19	6.53	6.87	7.21	7.55	7.89	8.23	8.57	8.91	
60°	1.17	1.52	1.86	2.22	2.55	2.92	3.26	3.60	3.95	4.28	4.63	4.98	5.32	5.67	6.02	6.47	6.70	7.04	7.40	7.75	8.10	
55°		.91	1.26	1.60	1.95	2.30	2.65	3.00	3.35	3.70	4.05	4.40	4.74	5.09	5.44	5.80	6.14	6.48	6.82	7.17		
50°						1.04	1.40	1.75	2.10	2.45	2.81	3.15	3.48	3.83	4.16	4.52	4.86	5.22	5.56	5.91	6.26	
45°							1.10	1.45	1.80	2.15	2.50	2.84	3.18	3.53	3.88	4.21	4.55	4.91	5.25	5.60	5.95	
40°								1.16	1.51	1.83	2.20	2.52	2.90	3.21	3.55	3.90	4.24	4.60	4.93	5.28	5.61	
44°									1.11	1.45	1.81	2.15	2.50	2.83	3.20	3.52	3.89	4.23	4.58	4.91	5.25	
42°										1.10	1.45	1.80	2.11	2.46	2.80	3.15	3.50	3.84	4.19	4.55	4.90	
40°											1.065	1.42	1.78	2.13	2.48	2.80	3.16	3.50	3.84	4.20	4.52	
38°												1.08	1.41	1.75	2.10	2.45	2.79	3.12	3.46	3.81	4.25	
36°													1.08	1.40	1.75	2.10	2.45	2.79	3.11	3.46	3.80	
34°														1.05	1.40	1.73	2.10	2.41	2.78	3.11	3.45	

AVERAGE TEMPERATURE OF REFRIGERANT (Brine or Ammonia) IN COIL

	+25°	+23°	+21°	+19°	+17°	+15°	+13°	+11°	+9°	+7°	+5°	+3°	+1°	-1°	-3°	-5°	-7°	-9°	-11°	-13°
32°	1.10	1.45	1.80	2.14	2.49	2.81	3.15	3.49												
30°		1.11	1.48	1.80	2.16	2.50	2.82	3.17	3.51											
28°			1.11	1.45	1.80	2.11	2.46	2.80	3.14											
26°				1.11	1.45	1.80	2.14	2.48	2.81	3.15										
24°					1.09	1.42	1.79	2.12	2.46	2.80										
22°						1.10	1.45	1.80	2.12	2.46	2.80									
20°							1.10	1.43	1.78	2.11	2.45									
18°								1.10	1.46	1.80	2.15	2.49								
16°									1.10	1.44	1.80	2.13	2.47							
14°										1.09	1.40	1.73	2.13	2.47						
12°											1.08	1.40	1.70	2.13	2.46					
10°												1.074	1.44	1.774	2.12					
8°													1.08	1.40	1.75	2.09				
6°														1.08	1.42	1.76	2.10			
4°															1.07	1.40	1.74	2.08		
2°																1.07	1.40	1.74	2.08	
0°																	1.08	1.41	1.75	2.09



SUBJECT:

DATE 1005

INDUSTRIAL COOLING UNITS
1500 SERIES

2/1/30

G E N E R A L

1. 1500 Series Cooling Units may be used with direct expansion ammonia, brine, or other recirculated refrigerants.
2. The total heat load to be absorbed by a Cooling Unit is made up of two parts, - the sensible heat and the latent heat loads. The sensible heat load is the heat to be absorbed to produce a change in Dry Bulb temperature. The latent heat load is the heat to be absorbed in condensing moisture from such air as is cooled below its Dew Point in passing thru the unit.
3. The absorption of latent heat by the coil takes place simultaneously with, and almost independently of, the absorption of the sensible heat. The coil can thus condense moisture at a considerable rate before its sensible heat capacity (i.e., its capacity to lower the Dry Bulb temperature of the air) is impaired to any considerable degree. Not until the latent heat load becomes more than 35% of the sensible heat load do we need recognize the reduction it causes in the sensible heat capacity of the unit.
4. The ratings of the units give their sensible heat capacities in B.t.u. per hour, which vary with the entering air temperature (Dry Bulb) and refrigerant temperature. Units are selected on the basis of the sensible heat (not total heat) requirement, which is the sum of the estimates of the following heat gains.
 - (a) Direct gain thru the building construction.
 - (b) The sensible heat of infiltration (heat to be absorbed in order to lower the incoming air from outside Dry Bulb to inside Dry Bulb).
 - (c) Any heat liberated in the room from lights, power, people, or product (live load).
5. The latent heat load is ignored for the purpose of selecting the unit (except as explained in paragraph 3 and in the footnote of Table I), but must be accounted for in the total heat load, which determines the capacity of the refrigerating equipment. The amount of moisture absorbed, and therefore the latent heat load, is dependent upon the Dew Point of the air and the refrigerant temperature. This moisture enters the room in three ways:
 - (a) Infiltration of air which is at a higher Dew Point than that of the room.
 - (b) Inward leakage of moisture, independent of infiltration, due to a higher vapor pressure outside.
 - (c) Moisture liberated by the product being cooled or stored, or brought in by any other means.
6. Rather than attempt to arrive at the latent heat load by laborious and uncertain estimates of these moisture gains on each job, we have prepared a table of factors for various applications

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by which the sensible heat requirement may be multiplied to give the total load directly. These factors (see Table I) are of necessity imperical, but each takes sufficiently correct account of the humidities, refrigerant temperatures, and room temperatures encountered in practice on the applications to which it applies.

NOTE:- Even though the nearest standard unit may be over capacity for the estimated requirements, the refrigeration equipment need not have greater capacity than that determined from the estimated sensible heat requirement multiplied by the total heat load factor from Table I.

DIRECT EXPANSION AMMONIA

7. In order to get the full rated capacity from the unit, the coil must be fully flooded. If the unit is not operating with the coil flooded, the capacity may vary anywhere from 100% of rating to 5% of rating, depending upon the percentage of flooded surface and the load on the unit. Piping Diagram S.D. 257.0 shows the recommended hook-up for automatically maintaining a flooded coil with any number of units supplied from one refrigeration plant. Piping Diagram S.D. 257.1 shows a hook-up which will automatically maintain a flooded coil where one unit is the only load upon the refrigeration equipment.

BRINE

8. Where brine is the refrigerant, the rate of brine circulation influences the capacity of the unit in two ways:

First - An increase in the quantity of brine gives a lower temperature rise, and consequently a lower average temperature of refrigerant, which results in a greater mean effective temperature difference and hence an increase in capacity.

Second - An increase in the brine quantity (and therefore its velocity) increases the rate of transfer from the brine to the tube wall, and therefore increases the output of the unit. The effect of brine rate is recognized and expressed in Plot I. (This plot is based on brine having a specific heat of .92 and a density of 1.07. With brines of other densities and specific heats, the error involved will be less than 5%.)

Since the rate of brine flow has such a vital bearing on the capacity of a unit, the brine rise should be relatively small, preferably 3 or 4 degrees. A brine rise of 6 degrees should not be exceeded unless the room temperature is as much as 25 degrees above average brine temperature. Diagram S.D. 257.2 shows the recommended method for piping a Brine installation.

SUBJECT:

Industrial Cooling Units
1500 Series

DATE

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TOTAL HEAT FACTOR

9. No matter what the refrigerant used, the total capacity of a unit when operating under a given set of refrigerant and air temperature conditions is vitally influenced by the humidity of the entering air. The following table gives, for various classes of rooms, probable room temperatures, refrigerant temperatures, and relative humidities, the factors by which the sensible heat load must be multiplied in order to get the total load. It is obviously impractical to make up a table covering all possible room temperatures, refrigerant temperatures, and humidities. It is therefore necessary that the men using this table exercise judgement, bearing the following conditions in mind: (a) With room temperature and refrigerant temperature fixed, an increase in humidity will increase the factor, and conversely a decrease in humidity will decrease the factor; (b) With room temperature and humidity fixed, a decrease in refrigerant temperature will tend to increase the factor, but this increase is of less magnitude than for conditions under "a".

T A B L E IFACTORS BY WHICH TO INCREASE SENSIBLE HEAT LOAD TO GET TOTAL LOAD

Class	Room Temp.	Refrigerant Temperature	Relative Humidity	Factor
Fruit Storage	32° - 38°	20°	90%	1.40 *
			80	1.20
Beef Cooling Curing	34 - 38	22	90	1.35
			85	1.30
Butter Storage	29 - 32	12	70	1.25
			60	1.15
Candy Mfg.	60 - 75	25	60	1.25
			50	1.15
Cheese Storage	40	25	90	1.50*
			80	1.30
Cheese Mfg.	55	32	90	1.60*
			80	1.40*
		40	90	1.50*
			80	1.30
Egg Storage	32	20	90	1.40*
Fur Storage	25 - 30	0	65	1.10
			60	1.05
Pork Cooling & Lard Storage	34 - 38	20	85	1.30
		10	80	1.30

(This Table continued on Sheet #4)

2/1/30

Class	Room Temp.	Refrigerant Temperature	Relative Humidity	Factor
Meat Cutting	50° - 55°	20° 10	85% 80	1.50* 1.55*
Milk Storage	32 - 38	15	85	1.35
Mushroom Growing	60	40	85 80	1.40* 1.30

*Decrease sensible heat capacity of the unit 10% where factor is above 1.35 (explained in paragraph 3). This reduction to be made before multiplying by the factor.

EFFECT OF FROST

10. The rating tables are based on a clean or but lightly frosted coil. A heavy frost will reduce the capacity of a unit, depending upon the thickness and conductivity of the frost. This drop in capacity is the result of an impaired heat transfer and a reduced air quantity. Reductions from this source may exceed 50% if defrosting is neglected. Frost is however under the control of the user and 20% (maximum) is enough to reduce the capacity of a unit as a factor of safety for frosting. -- Even with this allowance however the customer must still defrost at the necessary intervals if he wishes to maintain the specified room condition. Any reduction in sensible heat capacity to allow for frosting should be made before multiplying by the total heat factor.

There are a few applications where no reduction in capacity need be allowed for frost. These include:-

- Fur Storage.
- Chocolate Candy Storage.
- Candy Manufacturing.
- Any rooms which are kept dry and the product stored is in containers, such as Lard Storage or Brine Cartridges.
- Rooms where the refrigerant is 32° or above.

Equipment in such rooms, except those in class "e", will of course require defrosting at intervals, but frost should not build up rapidly after the room is once brought to temperature, so the capacity of the unit need not be reduced to compensate for frost.

USE OF TABLES AND CHARTS

11. DIRECT EXPANSION AMMONIA - EXAMPLE:

Assume the following conditions: Ammonia available at 28.5 lbs. back pressure, which corresponds to 15° F. Room temperature to be maintained at 35° F, with 75% relative humidity.

SUBJECT:

DATE

Industrial Cooling Units - 1500 Series

2/1/30

Determine the capacity of a #1534 Unit operating at 1750 r.p.m. with the coil completely flooded and with little or no frosting.

SOLUTION:

From the table of basic ratings we find that this unit is rated at 13,350 B.t.u. per hour for a 5 degree mean effective temperature difference. From the table of Condition Factors, for a 15° refrigerant and 35 degree room temperature, we interpolate a factor of 3.29. The sensible heat output of the unit therefore is $13,350 \times 3.29 = 44,000$ B.t.u. per hour. From Table I we find that the total heat factor is about 1.25 (see Butter Storage). The total load (refrigeration required) is therefore 44,000 $\times 1.25 = 55,000$ B.t.u. per hour or 4.58 tons.

12. BRINE AS REFRIGERANT - EXAMPLE:

Assume the same room conditions as in the ammonia problem, that is, the room to be maintained at 35° F. with 75% relative humidity. Assume that brine is available at 20° in sufficient quantity to limit the rise in brine temperature to not over 6° F.

Find the capacity of a #1534 Unit operating at 1750 r.p.m. with little or no frost.

SOLUTION:

The basic rating of 13,350 B.t.u. per hour is taken as in the previous problem. The rise in refrigerant temperature is assumed as 6°, therefore the average refrigerant temperature is 23°. Referring to the table of Condition Factors, interpolation gives a factor of 1.93. Neglecting for the moment the effect of brine rate, and assuming the transfer equal to that of a flooded ammonia system, multiply the basic rating by the Condition Factor.

$$13,350 \times 1.93 = 25,750 \text{ B.T.U. Per Hour - Sensible Heat.}$$

The total heat equals $25,750 \times 1.2 = 31,000$ B.t.u. per hour. (A total heat factor of 1.2 is used instead of 1.25 as in the ammonia problem since refrigerant temperature is higher.)

The above values must now be corrected for rate of brine flow. If the temperature rise of the brine is to be held at 6° F., the amount of brine required and the corrected B.t.u. capacity may be found from Plot I (Curve C-104), as follows:

Take the total heat of 31,000 B.t.u. per hour and locate this point on the bottom line, then move vertically upward to the oblique line marked 6°. From this intersection move horizontally to the 1534 velocity line. From this intersection read vertically to the 1534 rating curve. From this point on the rating curve, follow horizontally to the intersection with the 6° line. From here read vertically downward to the corrected B.t.u. capacity of 27,500 B.t.u. From this new rating point, read up to the 6° rise curve and from this intersection follow

2/1/30

- 6 -

horizontally to the left, where the brine required is read as 9.3 gallons per minute. We now have the total B.t.u. capacity of the unit as 27,500 B.t.u. per hour when operating with 9.3 gallons per minute of brine entering at 20°, and this brine rising to 26°.

In order to obtain the correct sensible heat capacity, divide the corrected total heat capacity by the total heat factor.

$$\frac{27,500}{1.2} = 22,900 \text{ B.T.U. Per Hour (The correct sensible heat rating of Unit for specified conditions).}$$

12. BRINE AS REFRIGERANT - EXAMPLE:

Assume the same room conditions as in the example problem, that is, the room to be maintained at 55° F. with 75% relative humidity. Assume that brine is available at 20° in sufficient quantity to limit the rise in brine temperature to not over 6° F.

Find the capacity of a 1254 Unit operating at 1750 r.p.m. with little or no frost.

SOLUTION:

The brine rating of 13,550 B.t.u. per hour is taken as in the previous problem. The rise in refrigerant temperature is assumed as 6°, therefore the average refrigerant temperature is 25°. Referring to the table of Condition Factors, interpolation gives a factor of 1.93. Neglecting for the moment the effect of brine rate, and assuming the transfer equal to that of a flooded ammonia system, multiply the brine rating by the Condition Factor.

$$13,550 \times 1.93 = 26,150 \text{ B.T.U. Per Hour - Sensible Heat}$$

The total heat equals 26,150 x 1.2 = 31,380 B.t.u. per hour. (A total heat factor of 1.2 is used instead of 1.35 as in the ammonia problem since refrigerant temperature is higher.)

The above values must now be corrected for rate of brine flow. If the temperature rise of the brine is to be held at 6° F., the amount of brine required and the corrected B.t.u. capacity may be found from Plot I (Curve C-104), as follows:

Take the total heat of 31,380 B.t.u. per hour and locate this point on the bottom line, then move vertically upward to the oblique line marked 6°. From this intersection move horizontally to the 1254 velocity line. From this intersection move vertically to the 1034 rating curve. From this point on the rating curve, follow horizontally to the intersection with the 6° line. From here read vertically downward to the corrected B.t.u. capacity of 27,500 B.t.u. From this new rating point, read up to the 6° rise curve and from this intersection follow

SUBJECT:

DEFROSTING COOLING UNITS

DATE

1/8/30

The use of refrigerants below freezing temperature causes frost to form on the coils of a cooling unit. The depth and amount of frost depends upon the temperature of the refrigerant and the amount of moisture in the room air. A light coating does not materially affect the capacity of the unit. A heavy coating of frost, however, if allowed to accumulate, will seriously reduce the air and B.T.U. capacity. The ideal method of operation is without frost, and in order to approximate this condition, the coils must be defrosted at necessary intervals. Several methods of defrosting are outlined herewith.

1. When room temperature is below freezing.

a. Hot gas from compressor into coil.

With direct expansion ammonia the most common method is to pipe to the unit a so-called hot gas line. This is a line direct from the compressor prior to the condenser, which permits hot ammonia gas to be fed into the cooling unit, which then acts in turn as a condenser. Condensation and cooling of these gases thus cause defrosting of the unit. During this operation the coil and return system are subjected to the high side pressure of the compressor which may amount to from 150 to 200 pounds.

b. Hot brine inside coil.

With brine, if there is a source of hot brine (i.e., above freezing) it may be piped to the unit and pumped thru the coil, either returning thru its own special line or else being admitted to the general brine supply.

c. Steam jet under coil.

Where neither hot gas nor hot brine are feasible, but steam is available, it is possible to defrost by admitting a jet of steam under the coil.

The equipment for this purpose consists of two or more galvanized pipes, (depending on size of unit), one inch in diameter, each drilled with 30 5/32" holes spaced on two rows. York can supply these sprays properly positioned at the bottom of the coil and secured to the base of the unit. See Air-Conditioning Bulletin #1200.

(NOTE: ABOVE METHODS SUITABLE FOR ONE OR MORE UNITS IN A ROOM.)

When there is more than one unit in a room:

- d. Hot liquor in one unit, direct expansion in others.

Units can be so piped up with an inter-connecting liquid line that any single unit will have hot liquid ammonia pass thru the coil and then expand afterward in the remaining units. In other words, the total liquid ammonia supply for the room is passed thru one unit, whose expansion valve is either wide open or bypassed, and then on to the expansion valves of the remaining units. By this arrangement the coil of the first unit is in series with the liquor supply line, and defrosting of this coil is accomplished by means of the warm liquor.

This method can also be used when there is only one unit in the room, but where there are other existing coils into which the liquor can be expanded after passing thru the coil of the unit.

2. When room temperature is above freezing.

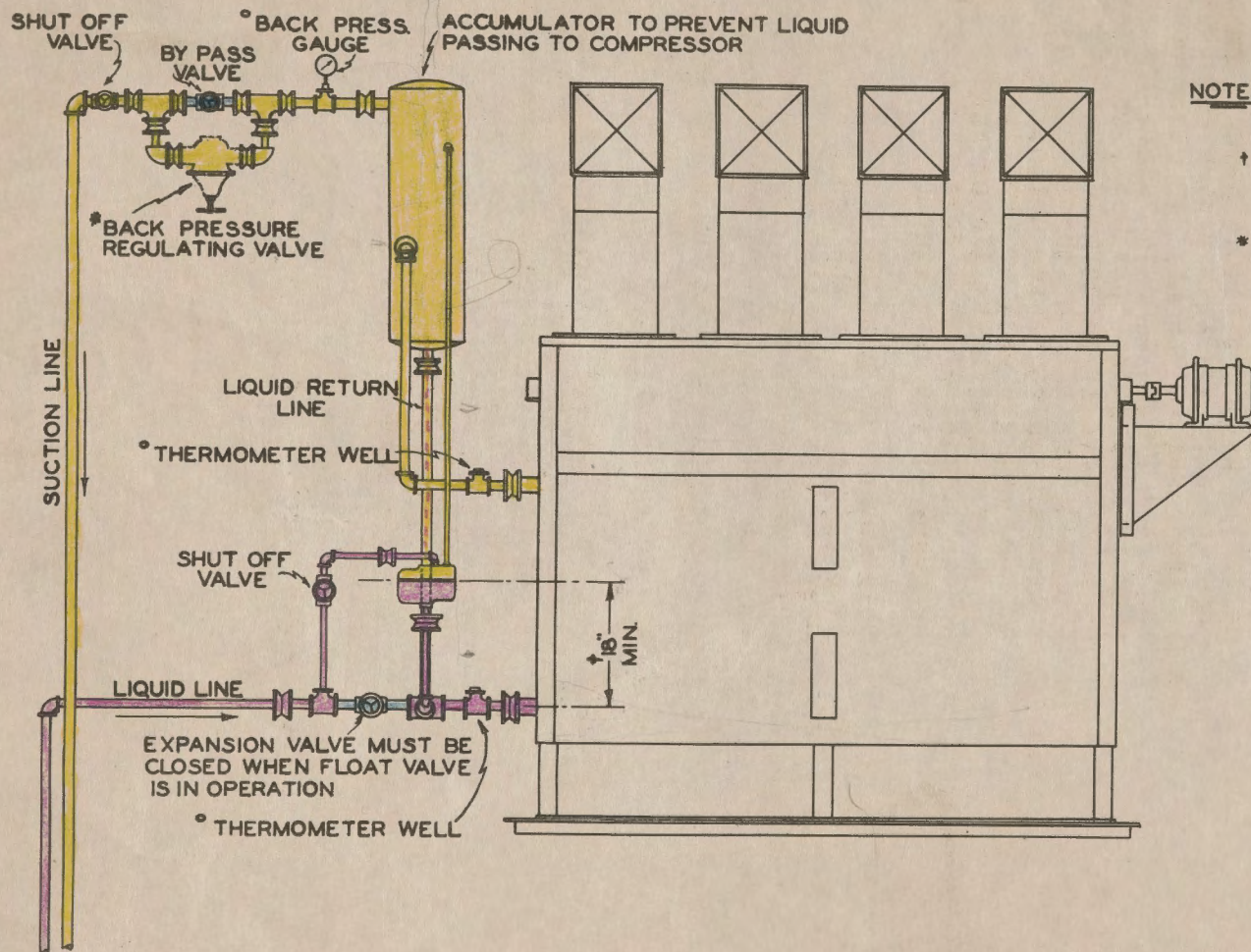
- a. Any of methods listed under (1).

- b. Shut off the refrigerant and let the fans run. However, the time for defrosting may require two or three hours, or even more, depending upon the room temperature and density of frost. If unlimited time is available, the fans may even be shut down. The higher the room temperature, the more rapid the defrosting.

3. The preferred method of defrosting is by means of hot gas as outlined in paragraph 1-A.

PIPING CONNECTIONS FOR YORK AIR COOLING UNIT

FLOODED AMMONIA SYSTEM



NOTES:-

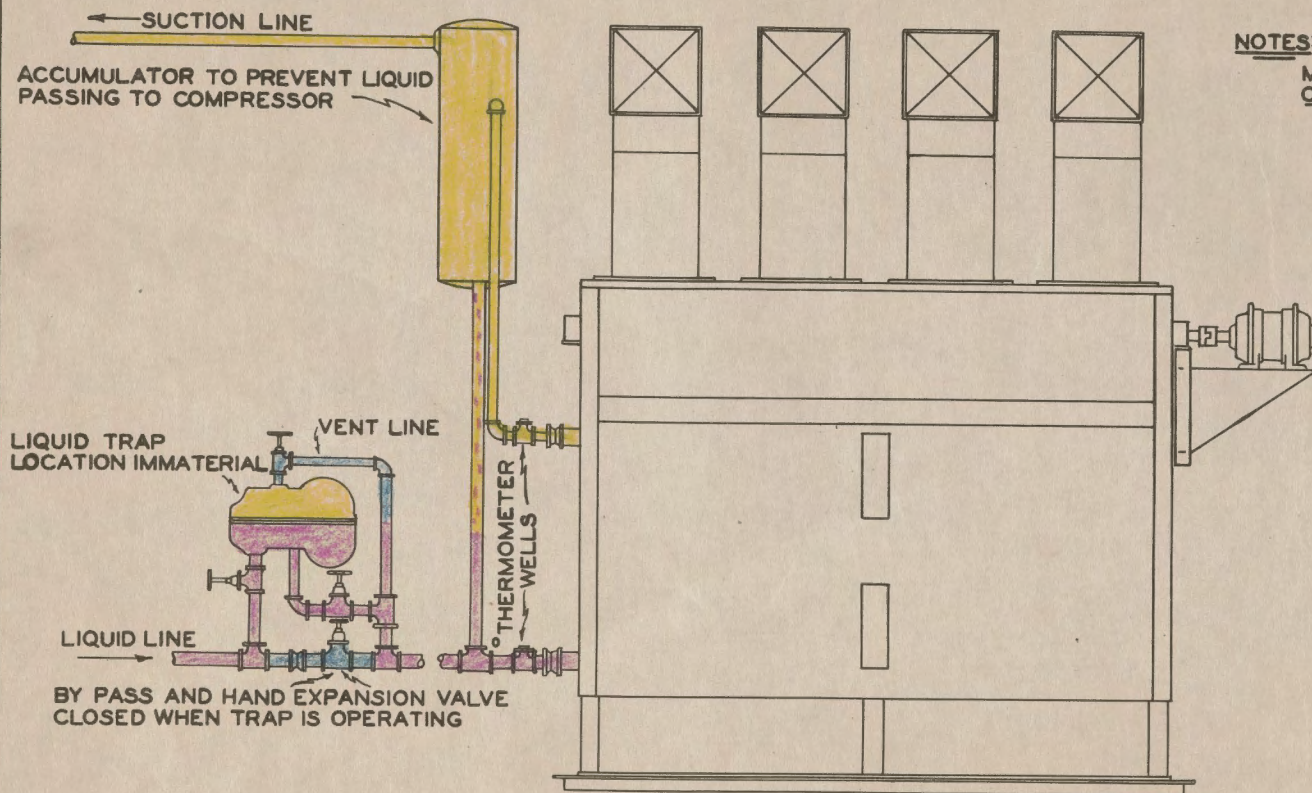
- FOR USE ON MULTIPLE OR SINGLE UNIT INSTALLATIONS.
- † LIQUID LEVEL TO BE AT LEAST 18" ABOVE CENTER LINE OF BOTTOM HEADER TO INSURE FULL FLOODED COILS WHEN IN OPERATION.
- * BACK PRESSURE REG. VALVE REQUIRED ONLY ON MULTIPLE SYSTEM WHERE IT IS DESIRED TO CONTROL THE BACK PRESSURE ON EACH UNIT INDIVIDUALLY.
- ° OPTIONAL

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PIPING CONNECTIONS
FOR
YORK AIR COOLING UNIT

FLOODED AMMONIA SYSTEM



NOTES:-

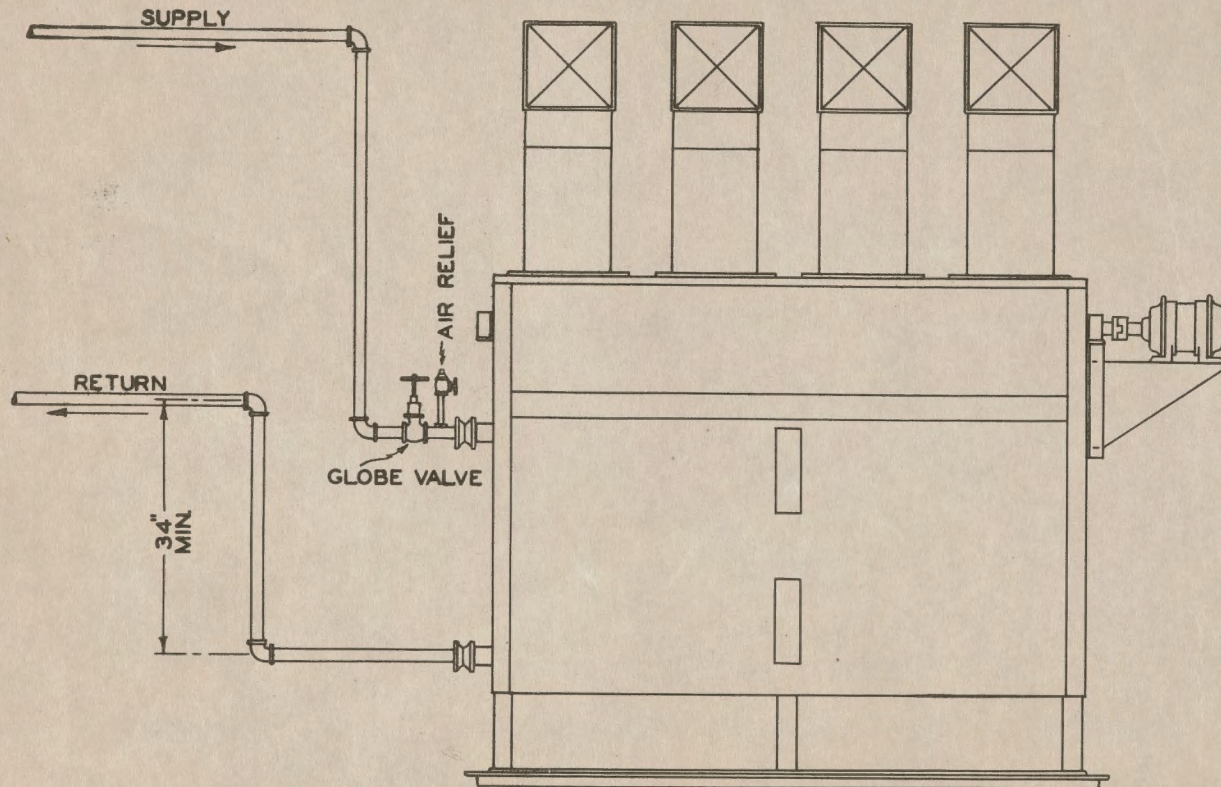
MAY BE USED WHEN ONE UNIT IS
ONLY LOAD ON COMPRESSOR.
° OPTIONAL

YORK HEATING & VENTILATING CORP.

PHILADELPHIA PA.

PIPING CONNECTIONS
FOR
YORK AIR COOLING UNIT

BRINE SYSTEM



YORK HEATING & VENTILATING CORP.
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YORK H.&V. CORP. S.D. 2572
DATE 1-25-30 R.B.D.

APPROVAL.

The foregoing thesis is hereby approved as a creditable study of an engineering subject, carried out and presented in a manner sufficiently satisfactory to warrant its acceptance as a prerequisite to the degree for which it has been submitted. It is to be understood that by this approval the undersigned does not necessarily endorse or approve any statement made, opinions expressed, or conclusions drawn therein, but approves the thesis only for the purpose for which it is submitted.

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