CHARACTERISTICS OF A WESTINGHOUSE 810 SQUARE FOOT SURFACE CONDENSER AS INSTALLED IN THE UNIVERSITY OF TENNESSEE POWER PLANT

by

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The Westinghouse condenser operating in connection with a 500 K.W. G.E. turbine as installed in the University of Tennessee Power Plant has never operated according to the manufacturers specifications. As material on the actual operating characteristics of condensers is scarce, the trouble as yet has not been found. In an attempt to locate the cause of the low efficiency, the authors have made a series of tests showing the characteristics of the condenser under actual operating conditions.

The authors hope that this material will be found beneficial to those who are interested.

Turbines are coming to be the prime movers of the world. In order for them to be efficient, their auxiliaries have to be developed with them. The main auxiliary and the controlling factor of turbine operation is the condenser. Condensers were not so much needed in the days of steam engines, for the steam was at low pressures, the engine speeds were low; therefore they could exhaust into the atmosphere very satisfactorily. Another reason would be that the steam engine could handle a low vacuum due to the limited size of the piston.

The general principle of the condenser is the reducing of the back pressure and the recovering of the condensate. The reduction of the back pressure is its prime object. Under some conditions the recovery of the condensate is considered important. The condenser tends to draw the steam from the turbine. Saturated steam is confined in a closed tank, and the heat is abstracted resulting in a drop of temperature and pressure, and at the same time condensing part of the vapor. The amount of heat abstracted will depend mainly on the temperature of the cooling water. The greater the amount of heat abstracted, the greater the amount of vapor condensed, and the lower the pressure.

Atmospheric pressure varies according to the elevation above sea level. For ordinary conditions, this pressure is considered as 14.7 pounds per square inch. The condenser trys to lower this pressure which would be acting against the exhaust if the engine or turbine were exhausting into the atmosphere. The present day turbines would be exhausting at about 16 pounds per square inch if it were not operating condensing. By the use of condensers, the turbines can exhaust with a vacuum up to the first stage. "The improvement of turbine design has made it possible to utilize efficiently the high vacuum that is now practical to obtain; therefore, vacua that were originally considered high are entirely inadequate." [1]

Condensers may be classified as jet, barometric, and surface, all of which work on the same thermodynamic principles.

The general functions of the surface condenser has been previously described. The exhaust steam from the turbine enters the condenser at the top and passes downward through, or over, a series of tubes. Cooling water is passing through the tubes constantly at a high velocity. The water enters the tubes at the bottom of the condenser and makes several passes through the condenser before it is removed at the top. The cooling water never comes in contact with the steam in this type of condenser. "The steam passages between the tubes are proportional to minimize the resistance and the pressure drop from the steam entrance opening to the air off take opening. The condenser shell is usually made of cast iron." [2] Condensation of the steam takes place as it flows down through the tube nest from top to bottom. As the steam condenses, the temperature is lowered along with a lowered pressure. When steam is changed to water, "it has reduced its required space about sixteen hundred times."[3] The air can not get into the closed tank to take the place vacated by the steam, thereby creating

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1. Westinghouse Steam Condensers and Auxilliaries. I.B. 5483

2. Ibid.

3. Handbook of Steam Engineering, by Biggs and Woolrich.

a partial vacuum. The lower the temperature and pressure of the condensate, the higher the vacuum.

Air is naturally present in the steam---can not be condensed at ordinary temperature---more is created by the condensation, and provision for its removal is made possible by the use of air pumps and air ejectors.

The exhaust steam from the unit remains separate from the cooling water; therefore it is possible to measure the condensate or the steam consumption of the unit without the possibility of an error. This can not be done using any other type of condenser.

"The efficiency of the surface condenser, determined by the quanity of steam it will condense per hour per square foot of cooling surface, when using a given quanity of cooling water, depends principally upon the following factors:

(1) Velocity of the water in the tubes.

(2) Terminal difference in temperature between the water and steam.

(3) Efficient steam distribution over the cooling surface.

(4) Absence of air in the condenser.

(5) Freedom of the tubes from flooding by condensate.

(6) Condition of the tube surface."[1]

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1. Condensers and Auxiliaries, C.H.Wheeler Mfg. Co.

The purpose of the work was to determine the reasons why the condenser was not operating according to the manufacturers specifications which guaranteed 27 inch vacuum when using 75 degree fahrenheit cooling water and a throttle pressure of 175 pounds per square inch.

The theory of the condenser, as has been discussed, is the lowering of the exhaust pressure of the steam by lowering the temperature. There are several considerations involved.

"High vacuum requires that the latent heat of the steam shall be efficiently transmitted to the cooling water at a low temperature, that frictional resistance to the flow of steam through the tube bank be small, and that the air entrained in the steam and entering through leaks be effectively removed so that its influence on the vacuum is negligible."[1]

The frictional resistance ia a matter of design and so it nedd not be considered in this discussion.

Air in a condenser has two harmful effects on the vacuum. It increases the pressure according to the Law of Partial Pressures, and if present in quanity, it surrounds the tubes, and impairs the heat transfer.

The equipment provided for removing the air was examined and was found to be in good condition. This equipment was designed to remove more air than should be present in the steam, so the lines were checked for leaks. Only one leak was found, and the vacuum did not improve very much when this was repaired, so this

1. Westinghouse Condensers and Auxiliaries. I.B. 5483

phase of the work was dropped.

The transmission of heat from the steam to the cooling water involves many factors. Heat transfer varies with the temperature difference between the steam and the cooling water, the material, thickness, size, shape, and cleanliness of the tubes, the velocity of the water through the tubes, the percentage of air on the steam side of the tubes, the extent of the water blanketing on the steam side of the tubes, and the viscosity of the cooling water.

Most of these are beyond the control of the experimenter. The material, size, and shape of the tubes are matters of the designer. As the cooling water pump was driven by a constant speed induction motor, the velocity of the water going through the tubes could not be varied. The air did not seem to be causing the trouble, nor did water blanketing, as the condensate pump is of more than sufficient capacity to handle the condensate. This leaves only the cleanliness of the tubes and the temperature difference as influencing factors.

The tubes of the condenser were cleaned, and though this gave an increased vacuum, the performance was still unsatisfactory. Therefore, the remainder of the work done was aimed toward investigating the actual heat transfer, especially as affected by the temperature difference between the cooling water and the steam.

A number of experiments were made taking the temperature of the actual cooling water entering and leaving the condenser, the temperature of the steam as it left the turbine, the vacuum in the condenser, the load on the turbine and the amount of cooling water being pumped through the condenser. The amount of steam condensed was also measured, but this was found to be useless in the calculations of the heat transfer.

From these readings, the heat transfer of the condenser in B.T.U. per square foot per hour per degree difference of temperature was calculated.

These values show some rather strange results. It would seem that at low loads, the vacuum would be higher, but the tests show that with an increased load the temperature of the cooling water remains constant, a higher vacuum was obtained, and a much greater temperature drop in the spray pond.

Sample Calculations:

(1) Inlet temperature of cooling water--------76 deg,F.--- t_1

(2) Outlet temperature of cooling water-------94 " " --- t_2

(3) Temperature of the steam-----------------120 " " --- t_s

(4) Logarithmic mean temperature difference-------

$$\frac{t_2 - t_1}{\log_e \dfrac{t_s - t_1}{t_s - t_2}}$$

Log. mean temp. diff.-- $\dfrac{94 - 76}{\log_e \dfrac{120 - 76}{120 - 94}}$ --- $\dfrac{18}{\log_e 1.69}$ -- $\underline{34.4}$ deg.F

(5) U-- coefficient of heat transfer----

$$\frac{\text{Total heat transfered per hour}}{\text{Area x mean temp. difference}}$$

U-- $\dfrac{233,623\ (94 - 76)}{810 \times 34.4}$ ---- $\underline{151}$ B.T.U. per hour per square foot per dĕgree difference in temperature.

Pump Calibration:

Run	Time (in aec.)	Q (cu. ft.)	Q (#/hr.)
1.	230	240	233,500
2.	230	240	233,500

230 : 3600 :: 240 : X

X....$\underline{3756}$ cu. ft. per hour

3756 x 62.5.....$\underline{233,500}$ pounds per hour

The results of the experiment are clearly shown by the following tables:

With the cooling water 76 deg. far. to 80 deg. far.

Load in K.W.	Temp. drop in spray pond	Vacuum in in. Hg	Heat transfer rate in B.T.U.
118	16 deg.F.	25.4	118
185	18 "	25.6	157
190	18 "	25.8	151
230	17 "	26.75	240
390	28 "	25.8	283
420	31 "	24.9	292

With varying loads.

Inlet temp. of cooling water	Temp. drop in spray pond	Vacuum in in. Hg	Heat transfer rate in B.T.U.
92 deg.F.	15 deg.F.	22.4	89.3
91 "	15 "	20.5	72.3
84 "	22 "	25.3	200.0
82.5 "	16 "	25.4	118.0
80 "	31 "	24.9	292.0
78 "	28 "	25.8	283.0

It is seen from the tables that the vacuum varies with the temperature of the cooling water. However, the efficiency of the spray pond varies with the load which is placed on the turbine. With cooling water at 80 deg. far. and a load of 420 K.W., the vacuum was 24.9 inches of mercury, the temperature drop in the spray pond is 31 deg, far., and the heat transfer rate is 292 B.T.U. per hour per square foot per temperature difference in degrees. With the cooling water at 76 deg. far. and a load of 185 K.W., the vacuum was 25.6 inches of mercury, the temperature drop in the spray pond was only 18 deg. far., and the heat transfer was 157 B.T.U. per hour per square foot per degree difference in temperature.

In order to obtain efficient operation, it seems that the load must be incresed and the temperature of the cooling water must be lowered. The future development of the University will increase the load which the turbine carries; so this will naturally take care of itself. It was suggested that a pipe line be laid into the spray pond so as to supply it with water from the city mains, but an experiment made with a temporary line supplying city water showed that the expense would not be justified by the small gain in vacuum. To increase the size of the spray pond seems to be the only practicable way to lower the temperature of the cooling water, as the present installation will not cool the water below atmospheric temperature.

The inadequacy of the spray pond is seen from the test runs shown by table number one. The temperature of the air surrounding the spray pond was 57 degree and 56 degrees while the humidity was 27 and 21 (U.S.Weather Bureau). Under these conditions, which should tend toward very good evaporation with consequent low temperature, the temperature of the cooling water was 91 and 80 degrees fahrenheit respectively. This gives a temperature difference between the atmosphere and the cooling water of 34 and 24 degrees fahrenheit. According to the "Power Plant Engineering Handbook", a well designed spray pond, with a humidity from 20 to 30 percent, should cool the water as much as 8 degrees below the atmospheric temperature. The spray pond, therefore, is very inefficient.

The only improvement which can be made with present equipment is to keep the condenser tubes clean as possible. The spray pond should be enlarged about twice the present size so that more evaporation can be obtained, and the gradually incresing load will tend to cause more efficient operation.

BIBLIOGRAPHY

1. MECHANICAL ENGINEERING LABORATORY PRACTICE, by Charles F. Shoop and George L. Tuve. First Edition.

2. PRINCIPLES OF THERMODYNAMICS, by G. A. Goodenough. Third Edition, revised.

3. Feedwater Condensers Cooling Equipment. Power Plant Engineering Handbook.

4. Westinghouse Steam Condensers and Auxiliaries, Instruction Book. Westinghouse Electric and Manufacturing Company. I. B. 5483.

5. The Cooling of Water for Power Plant Purposes, by Raymond Clayton Rush. Thesis 1928.

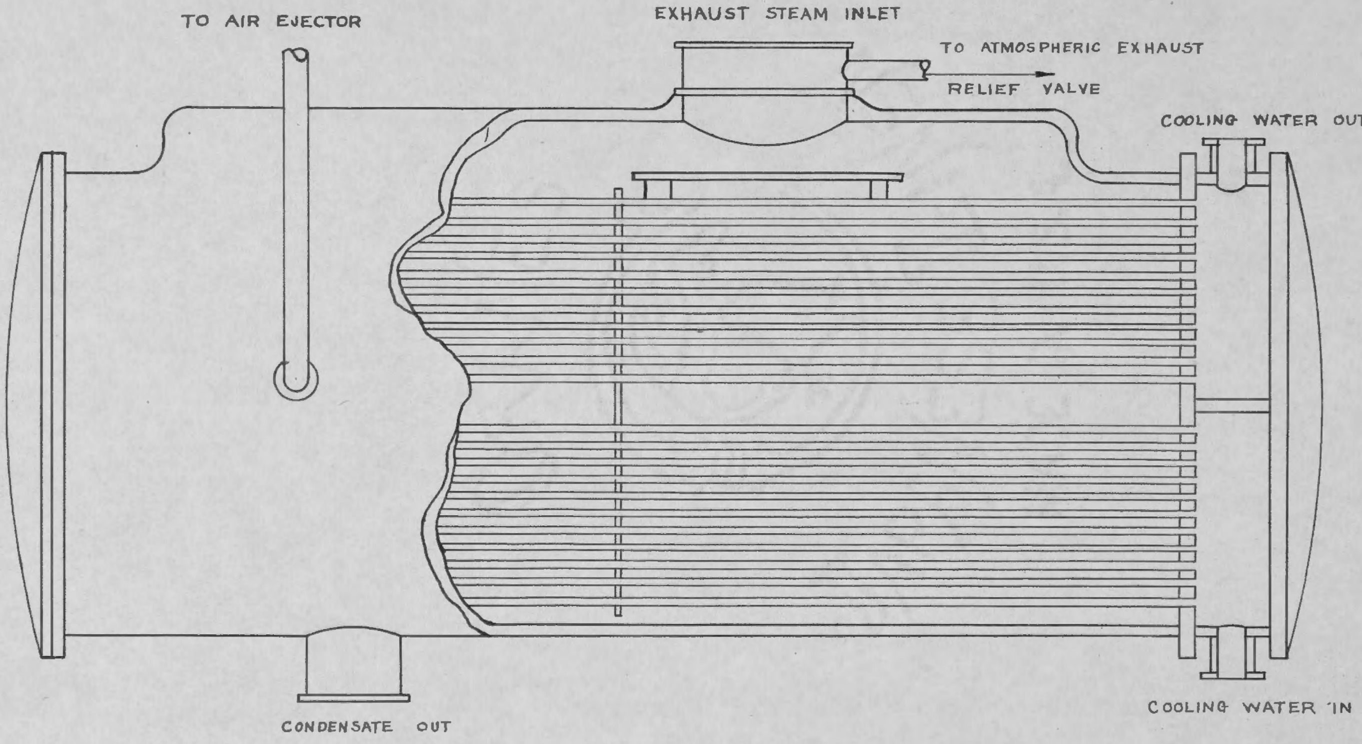
6. Condensers and Auxiliaries. C. H. Wheeler Mfg. Co., 1914.

Table No 1.

	Run No.1	Run No.2
Lenth of run	1 hour	1 hour
Av. temp. cooling water (in)	82.5 deg.F.	84 deg.F.
Av. temp. cooling water (out)	98.5 deg.F.	106 deg.F.
Av. temp. condensate .. deg.F.	125	128
Av. vacuum .. inches Hg	25.4	25.3
Pounds of cooling water cir.	233,500	233,500
Heat abstracted by cooling water ... B.T.U.	3,736,000	5,137,000
Water added through ejector	9,096 #	9,781 #
Av. temp. ejector water	70 deg.F.	65 deg.F.
Heat abstracted by ejector water ... B.T.U.	500,280	616,203
Pounds of steam condensed	12,950	14,923
Mean temperature difference	34 deg.F.	33 deg.F.
Heat transfer ... B.T.U./hr./deg. diff./square foot	136	198
Load in K.W.	118	200
Temperature of air	57 deg.F.	56 deg.F.
Humidity of air	27	21

Table No. 2

ITEM		Units	Run 1	Run 2	Run 3	Run 4	Run 5	Run 6	Run 7
Observed barometer	1	In. Hg	29.38	29.05	29.38	29.05	29.05	29.05	29.05
Corrected barometer	2	"	30.00	30.00	30.00	30.00	30.00	30.00	30.00
Observed vacuum at inlet	3	"	22.40	24.90	20.50	25.60	25.80	26.75	25.80
Corrected vacuum at inlet	4	"	23.02	25.65	21.12	26.35	26.55	27.50	26.55
Absolute pressure at inlet	5	"	6.98	4.35	8.88	3.65	3.45	2.50	3.45
Steam temp. corresponding to No.5	6	Deg. F	146.80	128.50	157.00	122	120	108	120
Observed steam temperature	7	"	146.00	127	156	119	118.5	106	119
Cooling water temperature (in)	8	"	92.00	80	91	76	76	78	78
Cooling water temperature (out)	9	"	107.00	110	106	94	94	95	106
Cooling water temperature rise	10	"	15.00	31	15	18	18	17	28
Temperature difference	11	"	39.80	17.5	51	28	26	13	14
Arithmetical mean temp. diff.	12	"	46.50	33	57.5	34	32	19.5	27
Logarithmetical mean temp. diff.	13	"	46.20	30.5	57.2	33.1	34.4	20.4	28.4
Load on turbine	14	K. W.	195.00	420	200	185	190	230	390
Cooling water per hour	15	Pounds	222,872	233,623	222,872	233,623	233,623	233,623	233,623
Heat transfer based on No.12	16	B.T.U./hr./sq. ft./deg. diff.	88.7	270	71.8	152	162	251	298
Heat transfer based on No. 13	17		89.3	292	72.3	157	151	240	283



WESTINGHOUSE TWO-PASS SURFACE CONDENSER

FIG. I

Figure No. 2

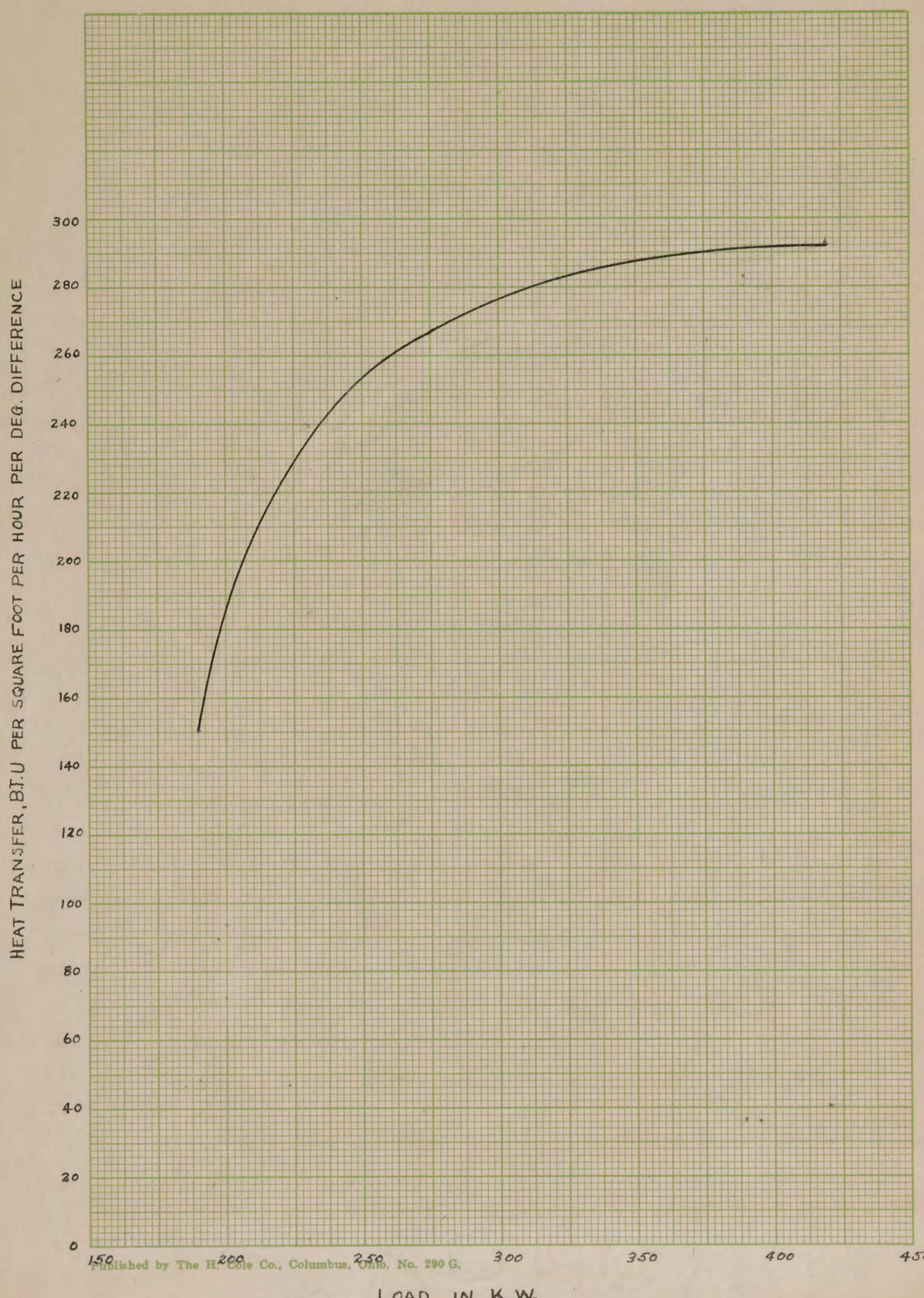
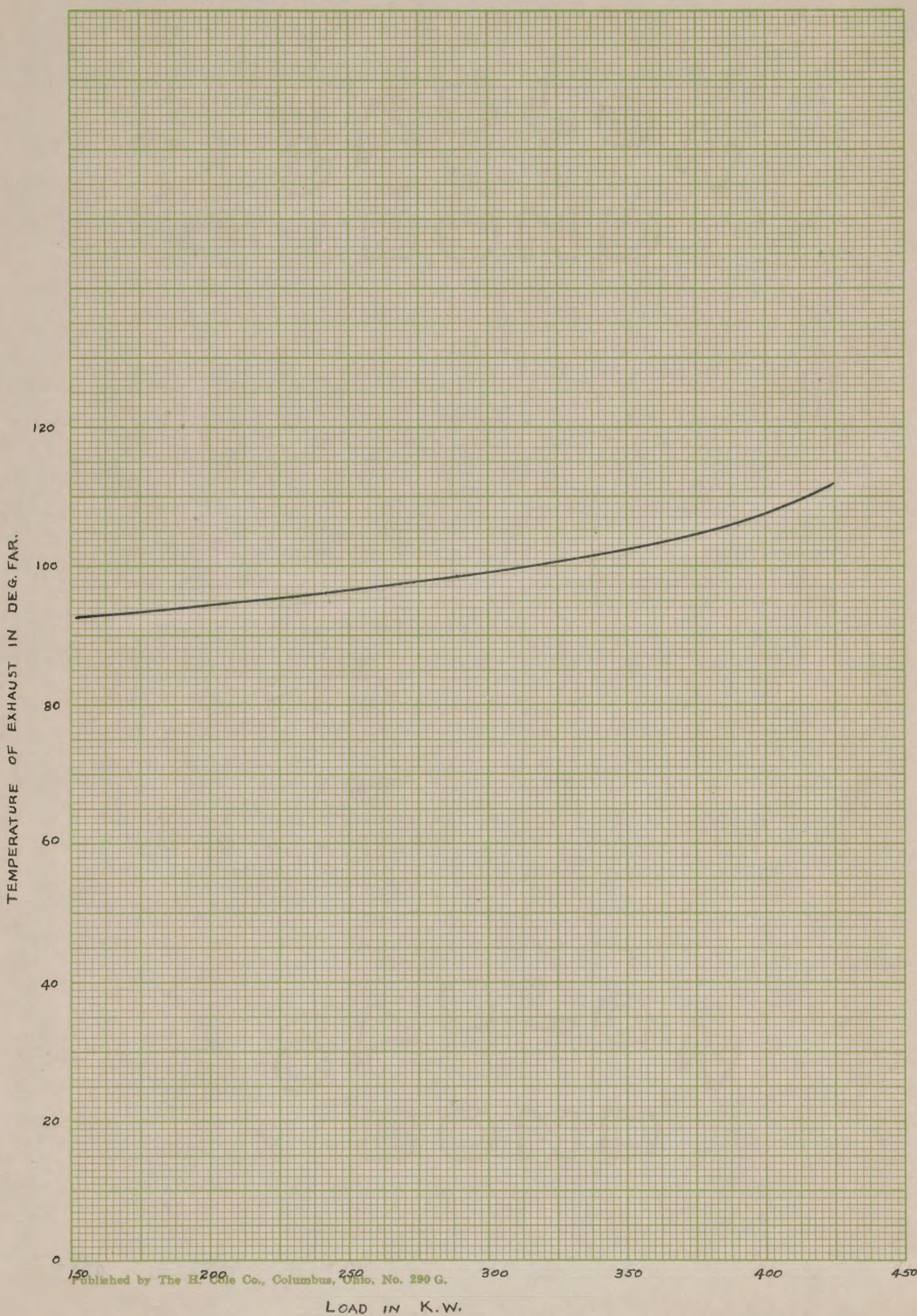
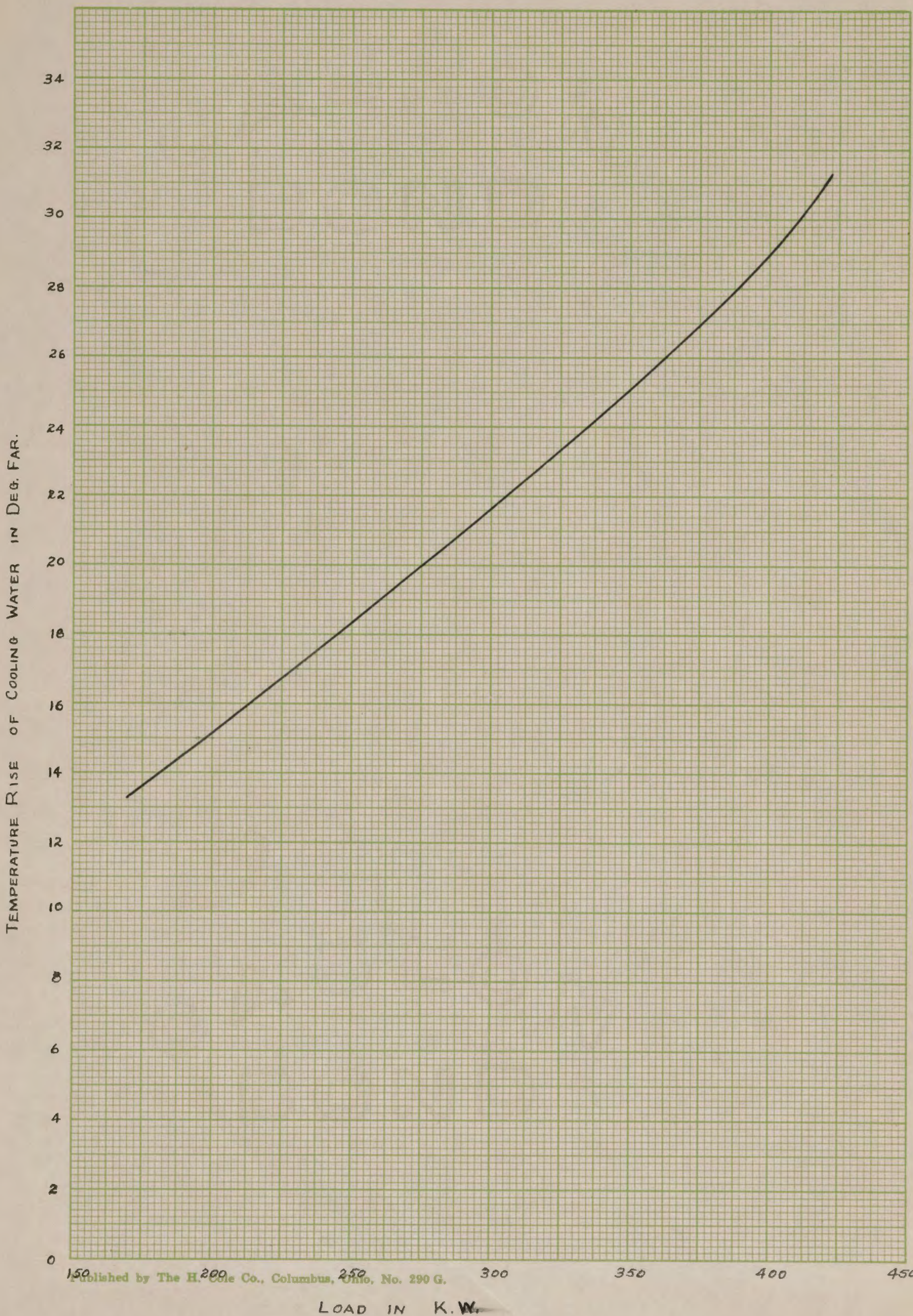

HEAT TRANSFER, B.T.U PER SQUARE FOOT PER HOUR PER DEG. DIFFERENCE
300
280
260
240
220
200
180
160
140
120
100
80
60
40
20
0
150
200
250
300
350
400
450
Published by The H. Cole Co., Columbus, Ohio. No. 290 G.
LOAD IN K.W.

Figure No. 3

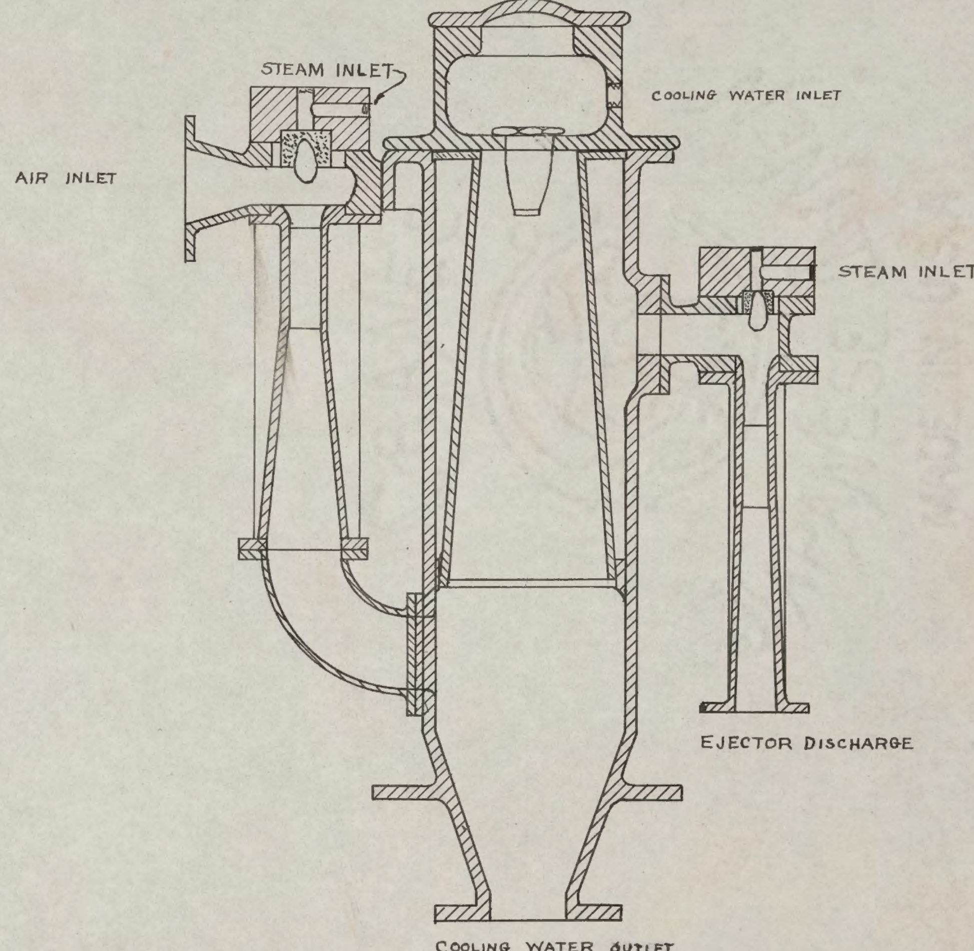


120
100
80
60
40
20
0
TEMPERATURE OF EXHAUST IN DEG. FAR.
150
200
250
300
350
400
450
LOAD IN K.W.
Published by The H. Cole Co., Columbus, Ohio, No. 290 G.

Figure No. 4



Temperature Rise of Cooling Water in Deg. Far.
34
32
30
28
26
24
22
20
18
16
14
12
10
8
6
4
2
0
150
200
250
300
350
400
450
Published by The H. Cole Co., Columbus, Ohio, No. 290 G.
Load in K.W.



CROSS SECTION THROUGH AIR EJECTOR WITH JET TYPE
INTERMEDIATE CONDENSER

The foregoing thesis is hereby approved as a creditable study of an engineering subject, carried out and presented in a manner sufficiently satisfactory to warrent 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 therin, but approves the thesis only for the purpose for which it is submitted.

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