A DETERMINATION OF THE EXISTING WATER RATES FOR THE TWO 500 K.W. TURBINES INSTALLED AT THE UNIVERSITY OF TENNESSEE

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by:

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I. AN HISTORICAL REVIEW OF THE CENTRAL POWER STATION AT THE UNIVERSITY OF TENNESSEE

Along with the natural growth and expansion of the University of Tennessee, the heating, lighting, and power demands increased until it was clearly seen that a central power plant located near the campus was an essential feature of the expanding building program. In view of this, money was solicited for several years until the appropriation in 1924 allowed the plant to be constructed. The actual construction represented an achievement of a long standing dream of the Dean of Engineering of the University, and, when completed, furnished the University with a modern, cleanly designed building which offered an additional feature in the 200 foot, tapered, tile-construction smokestack, one of the tallest in Knoxville.

The present installment takes care of the light, heat, and power requirements of all the buildings on the campus, besides furnishing steam heat and electricity to many of the University buildings located several blocks away. In addition, the electrical load at the Agricultural group of buildings, some ten or twelve blocks away, is carried by the plant.

One of the most interesting features of the 24 hour, 7 day per week service is the heat distribution and condensate return system. This consists of the network of approximately two miles of underground piping extending over the campus and to the surrounding University build-

ings. Two main steam lines, six and four inches respectively, carry steam from a valve in the basement of the power house, which reduces the steam from 175 pounds to 80 pounds, to the buildings served. In the basement of the individual buildings are other reducing valves which lower the steam in the line to 75 or as low as 50 pounds depending upon the intended uses. All used steam condenses and is returned to the power house at a loss of about one third the distributing volume, being helped back by a suction pump.

The normal electric load on the plant varies from about 200 K.W. to 300 K.W. per hour and amounts to approximately 60,000 K.W. hours per month. This power is supplied by means of two 500 K.W. steam turbine-generator sets capable of a 2300 volt potential. One is a Westinghouse turbine, installed in 1926, and served by a 1100-foot, Elliot-Ehrheart, surface condenser. The other is a General Electric, installed in 1928, and served by a Westinghouse, 900-foot, surface condenser. There were two 350 horse power, water-tube, Walsh and Weidner boilers installed to deliver steam at 175 pounds to the first unit, and two years later, two additional 350 horse-power boilers made by Casey and Hedges to supply steam at the same pressure for the second unit.

The water used for condensing purposes may be taken from the city's line or it may be supplied from a small creek flowing adjacent to the University property. A small

dam and overflow weir has been constructed to back the creek water up to about a four or five foot head, and this in turn is pumped to the power-house spray pond. One of the worst difficulties found in the plant is the problem of maintaining a high vacuum of about 27 or 28 inches, which at the present is seldom reached. The circulating water pumps handling the condensate do not seem to be able to handle enough cooling water and new water is added constantly. When utilizing the creek water, however, much sediment and suspended matter are carried through the condenser tubes with the consequence that the condenser gets dirty very soon after placing it on the line. This condition is accentuated by a rain as the water at such times is quite muddy. A fast sand-and-gravel filter is included in the plant's equipment, but due to design or lack of care is not very helpful.

As a summation, the University Power Plant has easily demonstrated the adviseability of maintaining a central power station, and serves not only the demands for heat, light, and power, but also for demonstration to students, and as a practical laboratory for valuable observations or experimentation.

II. PURPOSE

This thesis, submitted for acceptance as partial fulfillment of requirements for the degree of Bachelor of Science in Mechanical Engineering, has for its purpose the determination of the present water rates of the two 500 K.W. steam turbines installed in the Central Power Station at the University of Tennessee. In order to make the results more effective, the paper will also present the theoretical rates of the turbines and show a comparison between these rates and the present water rates as determined.

III. THEORY INVOLVED

There should be apparatus and means available to accurately determine the steam consumption, or water rate, of installed turbines in any power-house at practically any time desired. This provision should be foreseen during the design of the plant layout and properly cared for by the correct installation of meters and measuring devices according to the individual manufacturer's instructions. Then, as is often the case, tests may be made to see how nearly the performance at the time of the tests approaches the theoretical rates guaranteed by the makers for new equipment. It is the purpose of most of these tests to show the steam consumption required to develop a unit of output, whether expressed in terms of horse power or in kilowatts per hour.

Briefly, the path of the steam from the boilers out through the exhaust essentially involves: flow through the main steam line to the turbine, passage through the valves and nozzles, or stationary blades, and, after expansion through the various stages, out to the exhaust. For testing, it is desireable to have a surface condenser so that the condensate may be kept separate and weighed in large tanks on platform scales. A number of observations may be made for varying load conditions, and the steam consumption per unit of output at these loads plotted against the output composes a water-rate curve.

If the total flow of water (or steam) for a unit of time is plotted as ordinate and the corresponding load as abscissae, the resulting curve will be a WILLANS LINE. These rates do not offer actual comparisons unless the cycles involved are similar or if they are not computed for the same conditions. For example, a prime mover may have a water rate of 30.0 pounds per K.W. hour at a back pressure of 19.7 inches vacuum and a consumption of 26.6 pounds per K.W. hour at a vacuum of 24 inches. This consumption would be less for a higher back pressure because of the direct proportion of water rate to the efficiency of the machine and thus proportional to the available heat of the steam during adiabatic expansion.[1]

In any actual test there are several important observations to be made in order to have sufficient data for the calculation of the desired rate. These may well be:[2]

1. Pressure of the steam supplied to the turbine
2. Speed of rotation of the turbine shaft, usually in revolutions per minute
3. Measurements of power with a Prony or water brake; or with electrical instruments in case the currents, line voltages, or watts are used

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1. Goodenough, G. A. Principles of Thermodynamics, pp. 203, 176-178
   Gebhart, G. E. Steam Power Plant Engineering, pp. 972-978
2. Moyer, J. A. Steam Turbines, pp. 389-390

4. Weight, or measurement by volume of the condensed steam discharged from the condenser. This amount is very difficult to obtain unless the condenser is equipped with a surface condenser. The accuracy of a test often depends largely upon the way the leakage steam , escaping through pipes, pumps, and valves, is cared for. The amount of all these leaks should be added to the total condensate.
5. Temperature of the steam as it enters the turbine. If the temperature is higher than that due to the pressure of saturated steam(found in steam tables) the steam is superheated. If the temperature is lower than that of saturated steam, then the quality, or percentage moisture in the steam, must be ascertained. Tests are generally more satisfactory when made on turbines using slightly superheated steam. If the moisture content is more than 4%, the quality is very difficult to determine.[1]
6. Vacuum, or back pressure of the steam at the exhaust.

One of the most valuable tests made on steam turbines is that in which only the load is varied while other conditions remain constant, or as nearly constant as can be obtained. For each separate test a different load will be main-

1. Ibid.

tained, and each test will allow a calculation for a separate point on the final curve.

It may be timely to mantion here that if any accuracy is to be expected, it is only natural that all instruments to be used will have to be calibrated . This calibration should be made before and after each test, in order to correct for any errors that may ensue within the instruments themselves.

Another important test of the performance of turbines is made when both the speed and the power are varied, other conditions remaining constant. This type of test shows at what speed the turbine gives the greatest output or economy. For complete information regarding the testing of turbines and other prime movers, including codes of testing as adopted by the American Society of Mechanical Engineers, see Moyer's POWER PLANT TESTING, 3rd edition, pp. 354-386.

IV. DESCRIPTION OF METHODS USED IN THE DETERMINATIONS FOUND IN THIS THESIS

The general methods for determining the water rates of steam turbines have been followed in the tests run on the two 500 K.W. turbines installed in the Central Power Station at the University of Tennessee. There were several varitions from the conditions normally yielding the best results, but an attempt has been made to ascertain, as nearly as possible, accurate results. For one thing, observations had to be made on the machines while a normal load was being carried, and this load varied somewhat during the duration of some of the separate tests. However, the variations were not of a nature to cause alarm, and the results are felt to be characteristic. In order to obtain sufficient data for calculations, the following observations were made for each load variant for each of the two turbines:

1. A measure of the total condensate for the run
2. The condensate temperature
3. The power output, or kilowatts per hour
4. The boiler pressure supplying the turbine nozzles
5. The back pressure on the turbine exhaust

The measure of the total condensate during the run was not obtained by a method of direct weighing into large tanks on platform scales, as suitable apparatus for this was not available, but was obtained volumetrically by means of a 3" Worthington Piston Pump Meter in tests on the Westinghouse

turbine. This pump was first calibrated under actual load conditions of rates of flow and condensate temperature. (SEE APPENDIX). The piston meter was utilized in preference to a Cochrane Intergrating, Indicating, and Recording steam flow meter because of two factors. First, the piping connections to the piston meter were convenient and did not cause excessive loss of head to the condensate pump, and after direct calibration at load conditions, could be assumed to accurately measure the condensate. Secondly, the metering element of the steam flow meter (orifice) and ensuing piping connections were not installed according to the manufacturer's recommendations. Readings were made every five minutes from both, and an examination of the curves plotted for both reveals that the rates according to the steam flow meter are below the theoretical.

In tests on the General Electric turbine, however, conditions are practically reversed. The steam flow meter orifice is located in accordance to instructions and as the piping connections are favorable, good results should be expected. On the other hand, the piston pump meter handles not only the condensate chargeable to the turbine, but also a variable amount of water used in a jet condenser on the first stages of the turbine. A correction for this jet water has been attempted and the amount used in the jet recorded with each test. Also, the jet water meter has been calibrated , but as the test conditions for the calibration were not similar to the actual

conditions, and the rates obtainable in use were not available during the calibration, little faith can be placed in the results. At best the corrections for the jet condenser water was only a fair estimate, and could not be relied upon. The curves for both methods again bear out these assertions.

Average values of the K.W. loads were used, and each test was carried on for an hour whenever the load did not vary too much. In some cases, a fairly constant load was obtained for a duration of forty five minutes and in others for only thirty minutes. The total flow of condensate per hour is plotted against the power output to give a WILLANS LINE for each turbine. As the back pressures were different for each test, corrections had to be made to some basic vacuum before comparisions could be made. In view of the fact that the theoretical water rates were already known for a vacuum of 24" of mercury, all results were corrected to this back pressure. In order to accomplish this correction, we assume the water rates of turbines to be directly proportional to the available heat in the steam entering the nozzles as determined by the initial pressure and the back pressure.[1] Thus by means of the ratio of the available heat for the particuliar vacuum to that for the base vacuum, we can compute the correction factors to be applied

1. This is a popular practice in dealing with water rates and when applied to steam turbines is proven theoretically in most books of thermodynamics when applied to steam turbines.

to the observed water rate which would exist at the base vacuum. The higher the vacuum maintained as back pressure, the lower will be the water rate, and with this in mind we can easily tell which way the corrections should be made. In order to correct a rate to a higher vacuum, the existing rate will have to be multiplied be a fraction less than unity in order to obtain the lesser rate that would exist for the higher vacuum. (SEE APPENDIX) By reading the total heats from a MOLLIER Chart, the correction factor can be made to include the correction for moisture content if the initial steam is not quite saturated or is slightly superheated.

Another correction has to be made, when the steam flow meter is used, for the flow meter constant when initial steam pressures are different that that for which the metering orifice is calibrated. This factor can be obtained from any other number of published curves, but is nearly always available from data included in the instruction book accompanying a flow meter. All of the above corrections would have to be considered whether computing the water rate per unit output directly or whether computing the total flow per hour from which curve the rate could be taken.

V. OBSERVATIONS AND DATA

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All Tests Made At The Central Power Station

At The University Of Tennessee

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Observations By: Penland,R.F.
Nicholson,D.M.

Measuring Devices:

1. Condensate Measured With A 3" Worthington Piston Pump Meter
2. Instantaneous Power Output Measured With An Indicating Wattmeter With A 1000 Watt Scale
3. Steam Flow Measured With An Indicating, Integrating, Recording Cochrane Steam Flow Meter
4. Condensate Temperature Measured With A Commercial, Mercury-Column Thermometer (212 Degree Scale-Fahr.)
5. The Boiler Pressured In pounds per square inch, Gage, On A Panel-Type Bourdon Pressure Gage
6. The Back Pressure , Or Vacuum, At Turbine Exhaust Was Measured With A Standard 30" Mercury Column Vacuum Gage

UNIT I

Westinghouse Turbine, Age 9 years
500 K.W. - 3600 RPM - 165# - Impulse And Reaction

Condensing By An Elliot-Ehrheart Surface Condenser Whose Circulating Pump Has A Capacity Of 1100 gpm

TEST #I 6/19/35

TIME	PISTON METER READING	STEAM FLOW METER READ'G	CONDENSATE TEMP. (°F)	INCHS VACUUM	LOAD K.W.	STEAM PRESS (lbs)
4:30	2968208	50958.5	133	23.6	195	170
:35	276	61.8	133	23.6	197	165
:40	346	65.2	133	23.6	195	170
:45	413	68.2	132	23.7	180	172
:50	480	71.2	131	23.8	182	168
:55	547	74.0	131	23.8	183	168
5:00	613	77.4	131	23.8	180	173
	Tot.405	Tot.18.9	Av132	A23.7	187.4	170

TEST #II 6/20/35

3:00	2958984	51671.8	141	22.7	218	172
:05	9060	675.3	141	22.7	220	170
:10	136	679.0	140	22.8	215	171
:15	212	682.2	140	22.8	215	170
:20	286	686.0	140	22.8	217	168
:25	362	689.4	140	22.8	217	171
:30	437	692.7	140	22.8	220	168
:35	514	696.0	140	22.7	220	172
:40	591	700.0	141	23.0	223	171
:45	667	703.0	140	23.1	195	168
:50	738	706.5	139	23.2	193	173
:55	809	709.8	138	23.2	187	170
4:00	879	713.0	138	22.9	185	174
	Tot. 895	Tot.41.2	A139.9	22.9	209.6	171

TEST #III 6/22/35

3:15	2960700	53241.2	124	24.9	128	173
:20	757	43.8	124	24.9	130	173
:25	814	46.3	125	24.9	125	173
:30	871	48.8	124	24.9	127	173
:35	926	51.1	125	24.9	125	173
:40	981	53.4	124	24.9	124	173
:45	1036	56.0	124	24.9	128	173
:50	090	58.1	124	24.9	128	173
:55	146	60.5	124	25.0	124	173
4:00	200	63.0	124	25.0	123	173
:05	254	65.4	123	25.0	121	173
:10	307	67.9	123	25.0	128	173
:15	363	70.2	124	24.9	138	173
	Tot. 663	Tot.29.0	A 124	A24.9	126.8	173

TEST IV 6/28/35

TIME	PISTON METER READING	STEAM FLOW METER READ'G	CONDENSATE TEMP. (°F)	INCHS VACUUM	LOAD K.W.	STEAM PRESS. (lbs)
5:00	2961569	57537.2	119	25.2	80	177
:05	622	39.3	119	25.3	80	162
:10	670	41.2	118	25.2	82	180
:15	720	43.3	118	25.2	83	172
:20	768	45.4	120	25.1	81	175
:25	817	47.6	120	25.1	81	185
:30	865	49.7	120	25.1	84	180
:35	914	51.7	120	25.2	80	174
:40	963	53.8	120	25.1	86	180
:45	2013	55.8	120	25.0	93	173
	Tot.444	Tot18.6	A 119.4	25.2	83	178

UNIT II

General Electric turbine, Age 7 years
500 K.W. - 3600 RPM - 175# - 6 stage - type L

Condensing By a Westinghouse Surface Condenser Of 810 square feet, Whose Circulating Pump Has A Capacity Of 820 gpm

Measuring Devices
The same Apparatus Was Used On This Unit As Was Employed In Unit I

TEST #I 7/ 3/35

TIME	WATER		STEAM FLOW METER READ'G	CONDENSATE TEMP. (°F)	INCHS VACUUM	LOAD K.W.	STEAM PRESSURE
	JET	PISTON METER					
2:05	15517	2963008	60718.5	134	24.0	220	178
:10	23	098	20.7	135	24.0	225	165
:15	28	184	23.0	136	24.0	220	174
:20	34	274	25.2	136	24.2	218	186
:25	40	360	27.4	135	24.2	219	178
:30	45	448	29.5	136	24.1	217	182
:35	51	535	31.7	136	24.1	221	185
:40	56	623	33.8	136	24.0	218	180
:45	62	710	36.0	136	24.0	218	182
:50	67	797	38.2	136	24.0	220	188
	Tot50	Tot.789	Tot19.7	135.8	24.1	219.7	180

TEST #II 7/ 7/35

Time	Water Jet	Water Piston Meter	Steam Flow Meter Read'g	Cond's. Temp (° F)	Inchs Vacuum	Load K.W.	Steam Pressure
3:45	21626.4	2964547	62306.3	134	22.9	148	183
:50	30.4	621	08.0	134	22.9	145	182
:55	36.5	695	10.7	133	22.9	142	181
4:00	40.6	769	11.7	137	22.9	141	179
:05	46.7	843	13.8	138	22.9	138	183
:10	51.8	916	15.8	136	23.0	138	183
:15	56.0	988	17.8	137	23.0	140	185
:20	61.1	5063	19.8	138	23.0	138	188
:25	66.2	136	21.7	138	22.9	141	181
:30	70.9	208	23.6	139	23.0	140	181
:35	75.6	279	25.6	139	23.0	138	185
:40	80.3	350	27.5	139	23.0	138	186
:45	85.1	421	29.2	139	23.1	137	183
	Tot58.7	Tot.874	Tot.22.9	A137	23.0	140.3	184

TEST #III 7/ 9/35

2:10	26854.0	3618	884.5	148	20.0	240	183
:15	63.7	740	887.2	150	20.0	244	180
:20	73.2	864	890.0	152	19.9	242	179
:25	82.8	991	892.6	153	19.8	242	186
:30	92.6	4116	895.2	152	20.3	240	185
:35	902.6	240	898.0	152	20.2	240	185
:40	15.5	370	900.6	152	20.0	243	183
:45	28.3	508	903.4	148	20.0	240	181
:50	40.3	643	906.2	140	20.1	255	181
:55	50.9	781	909.2	142	19.7	252	183
3:00	61.4	920	912.2	146	19.1	260	181
:05	72.0	5060	915.2	152	18.8	250	178
:10	82.5	200	918.2	154	18.4	252	177
	Tot128.5	Tot 1584	Tot33.7	A148	19.7	246.1	183

TEST #IV 7/16/35

4:30	46464	5640.0	8080.3	140	21.9	185	182
:35	481	775.6	83.0	141	22.6	180	188
:40	493	902.0	85.2	135	22.9	176	190
:45	506	6034.3	87.5	128	22.8	173	186
:50	521	172.5	89.8	121	22.7	170	181
:55	534	302.0	92.0	121	22.7	170	183
5:00	548	429.5	93.9	123	22.7	164	183
	Tot 84	Tot789.5	Tot13.6	A130	22.6	174	185

TEST #V 7/17/35

Time	Water		Steam Flow	Cond's Temp. (°F)	Inchs Vacuum	Load	Steam
	Jet	Piston Meter	Meter Read'g			K.W.	Pressure
5:30	47426	37580.0	68380.0	119	25.8	90	180
:35	434	656.5	381.5	119	25.8	92	179
:40	441	733.0	383.0	119	25.8	93	181
:45	449	811.0	384.6	119	25.9	95	185
:50	457	886.0	386.3	119	25.9	97	183
:55	466	969.5	388.1	119	25.7	110	183
6:00	476	8063.5	389.9	120	25.6	110	182
	Tot.50	Tot483.5	Tot. 9.9	A 119	25.8	98.1	182

VI. CALCULATIONS OF RESULTS

A tabled account of results as determined in this thesis is given below in TABLE I, and the curves plotted from this data are shown in FIG. I and FIG. II. As has been explained before, the WILLANS LINES (FIG. I) were plotted from the corrected values of condensate existing at individual line loads. The performance curves representing a comparision between the existing water rates and the theoretical rates have been plotted with values taken directly from the WILLANS LINES, but in order to show as example of complete results, the water rates per indicated K.W. output as computed from the test data and not taken from the WILLANS LINES are included in the last four columns.

K.W. Load	Test Vacuum	Correction Factor-24"	Test Total Flow	Corrected Total Flow	Test Piston Meter Rate	Corrected Piston Meter Rate	Test Steam Flow Meter Rate	Corrected Steam Flow Meter Rate
				WESTINGHOUSE				
83.0	25.2	1.038	5840	6050	60.7	63.0	54.7	56.8
126.8	24.9	1.028	6540	6720	51.5	52.9	41.9	43.1
187.4	23.7	0.990	7987	7900	42.6	42.2	37.0	36.6
				GENERAL ELECTRIC				
98.1	25.8	1.059	4750	5030	32.3	34.2	48.5	51.3
140.3	23.0	0.970	5500	5340	34.7	33.6	39.3	38.1
174.0	22.6	0.958	6530	6260	27.0	25.9	37.5	35.9
219.7	24.1	1.002	6310	6383	25.0	25.1	28.7	28.8
246.1	19.7	0.888	8090	7190	30.0	26.6	32.9	29.2

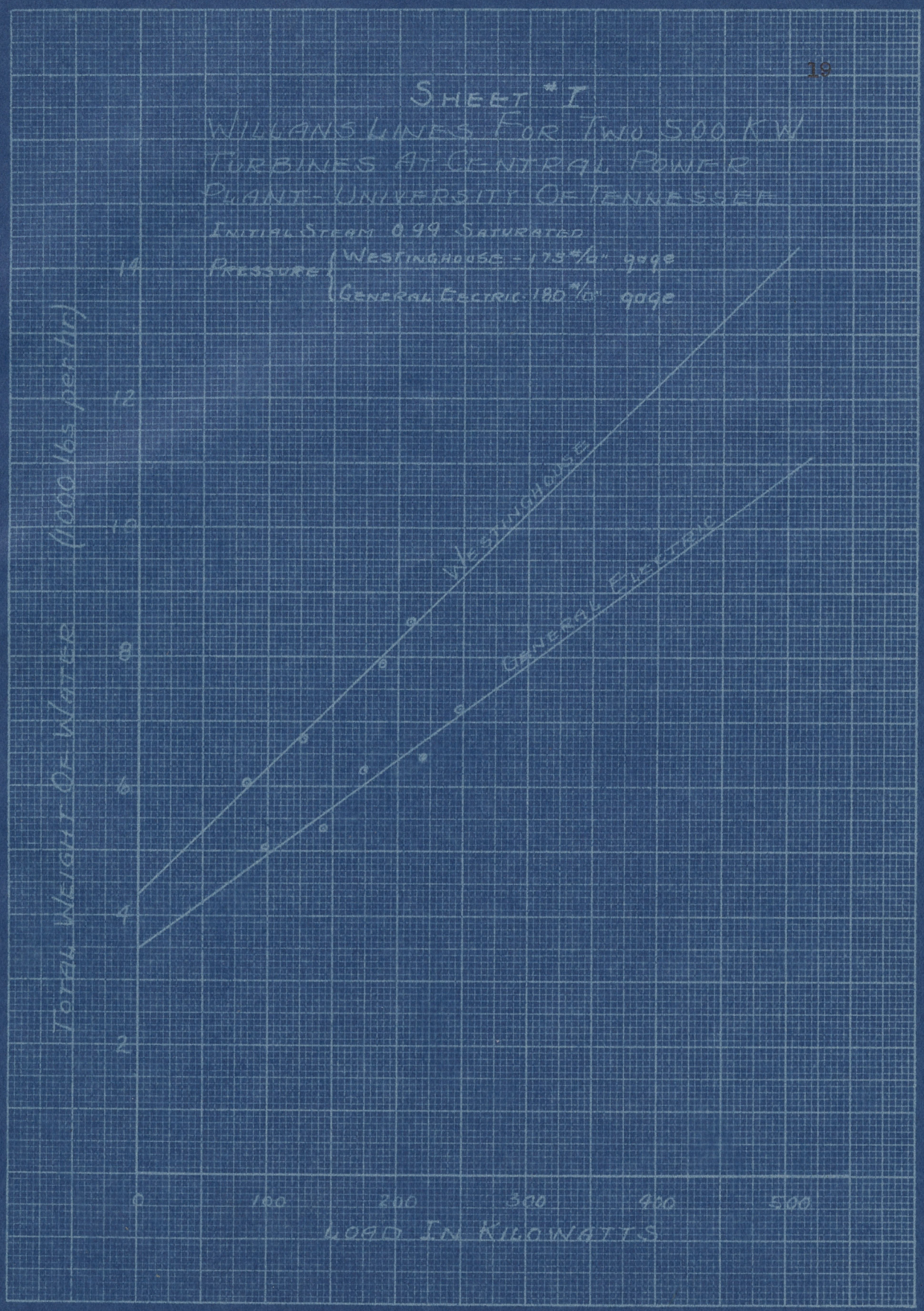
TABLE I

Sheet #I

Willans Lines For Two 500 KW Turbines At Central Power Plant - University Of Tennessee

Initial Steam 0.99 Saturated

Pressure { Westinghouse = 175 #/□" gage
General Electric: 180 #/□" gage



SAMPLE CALCULATIONS

UNIT #I - Westinghouse

Piston Meter Method:
Load 83 K.W. - TEST #IV - Duration Of Test 45 min's.

Total Condensate:
(444)(9.86)(4/3) = 5840 pounds
Correction Factor At Average Vacuum Of 25.2":
From Curve SHEET III,APPENDIX,= 1.038
Corrected Total Flow:
(1.038)(5840)= 6040 pounds ***

Individual Test Water Rate Per K.W. Hour:
$\frac{(444)(9.86)}{83} \times \frac{4}{3}$ = 60.7 #/K.W. Hour

Steam Flow Meter Method:
Using The Same Values As In Above(TEST IV)

Individual Test Water Rate Per K.W. Hour:
$180 \times \frac{1.260}{1.235} \times 18.6 \times \frac{4}{3} \times \frac{1}{83}$ = 54.7 pounds

$\frac{1.260}{1.235}$ Is A Pressure Correction From Cochrane Flow Meter Instruction Book, p. 20

Corrected Piston Meter Rate:
(1.038)(60.7)= 63.0 #/K.W. Hour
Corrected Flow Meter Rate:
(1.038)(54.7)= 56.8 #/K.W. Hour

UNIT #II - General Electric

Steam Flow Meter Method:
Load 140.3 K.W. - TEST #II - Test Duration 1 Hour

Orifice Meter Constant= 240
Negligible Pressure Correction
Total Flow Of Steam:
(240)(22.9)= 5500 pounds
Correction Factor From Curve In Appendix:
(Average Vacuum Of 23.0") f= 0.970
Corrected Total Flow:
(0.970)(5500)= 5340 pounds ***

Piston Meter Method:

Total Condensate:(874 units)
(874)(9.86)= 8620 pounds
Calibration Constant:
From Curve SHEET III, APPENDIX, Rate Of Flow Of
$\frac{58.7}{60.0}$= 0.98 cu. ft./min., C= 1.04
Total Flow Through Jet:
(1.04)(58.7)= 61.0 cu. ft./min.
At Condenser Temperature Of 136° F., The Density Of Water Is 61.45 lbs./cu. ft.
Therefore:
(61.45)(61.0)= 3755 pounds

Condensate Chargeable To The Turbine:
8620 - 3755= 4865 pounds *****
Corrected Total Flow Of Steam:
(0.970)(4865)= 4715 pounds
Individual Water Rate Per K.W. Hour:
4715/140.3= 33.6 lbs./K.W. Hour

Steam Flow Meter Method:

Individual Test Rate Per K.W. HOur:
5500/140.3= 39.3 lbs./K.W. Hour
Corrected Water Rate:
(39.30)(0.970)= 38.1 lbs/K.W. Hour

VII. CONCLUSIONS

By observation of the performance comparison curves, SHEET II, we can see thet the existing water rates for the two 500 K.W. turbines compare favorably with their theoretical rates. But we may examine the curves more closely.

For the Westinghouse turbine, the curves #3 and #4 apply to its theoretical and existing water rates respectively. From the two it appears that a greater variance from the theoretical is found when the loads on the machine are either high or low, or around 500 K.W. and 100 K.W.. At the 500 K.W. load, there is an increase in water rate of about 22% occuring after a period of nine years.[1] This is approximately 2.44% per year. At loads of about 100 K.W., the percentage increase in rate is about 1.50% per year, and around loads of 200 K.W. about 0.28% annually. A fair estimate for the range of the complete curve would be a yearly increase of about 1.00%.

According to tests of a large utility company, depreciation in efficiency caused by normal wear has been found to average about 0.78% per year.[2] These tests were undoubtedly run on large machines of high capacities, but the values afford a means of comparison and should apply very well to smaller ones also. As the efficiency of a turbine is inversely proportional to the water rate, the above water rate increase of

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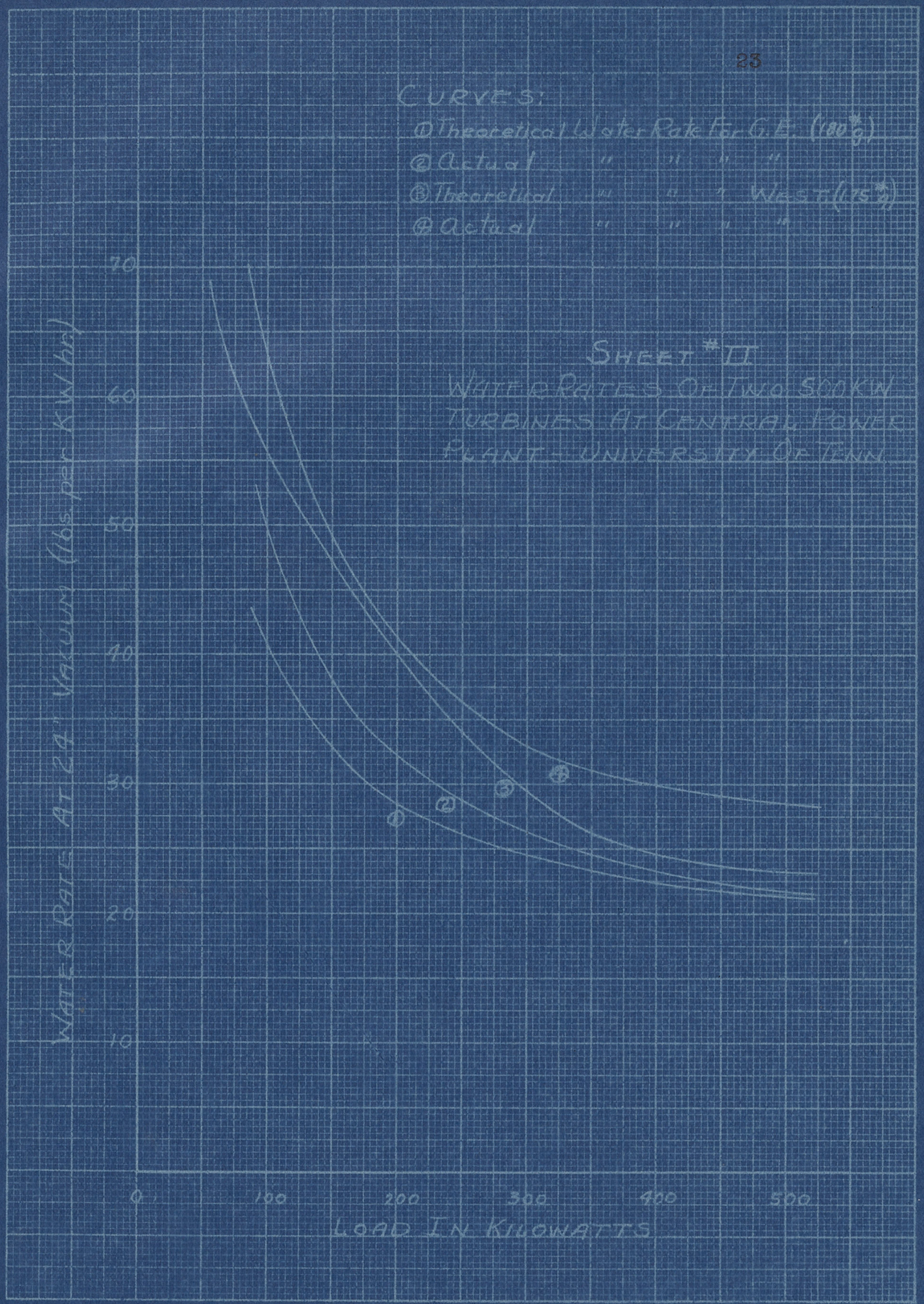
1. The turbine was installed in 1926
2. Reynolds, L.L. Obsolescence of Steam Turbines, Electrical Journal 31:221-223 June 1934

CURVES:
① Theoretical Water Rate For G.E. (180# gage)
② Actual " " " "
③ Theoretical " " " Waste (125# gage)
④ Actual " " " "



SHEET # II
WATER RATES OF TWO 500 KW TURBINES AT CENTRAL POWER PLANT - UNIVERSITY OF TENN.

1.00% per year compares favorably with the 1.00% deficiency found to be an average value for many turbines.

On examination of the curves #1 and #2, on the performance comparison sheet, applying to the General Electric turbine, we see a generally closer adherance. This might be expected, as the G.E. is not as old as the Westinghouse, being installed in 1928. The largest differences are found in the rates at low loads of around 100 or 200 K.W.. As the rated capacity of the turbine-alternator is approached, the variance becomes smaller and smaller.

At 100 K.W. the water rate increase is about 22.5% for seven years, or 3.2% per year. At 250 K.W., the increase is about 1.4% per year, and at 500 K.W. about 0.33% per year. If anything, these percentages are a little high when compared with the expected efficiency decrease mentioned above, but represent fair values.

The depreciation of steam turbines due to normal wear is caused by increased clearances of the blades and dummies resulting from rubs on blades, erosion on the cylinders and spindles, roughness of blades resulting from erosion and corrosion, leakage of inlet valves, deposits of scale on the blades, and many other factors. It is of interest to note, however, that by reseating valves, replacing blades, and generally overhauling a turbine, the efficiency can be restored to within 0.25% of its original efficiency for each year of service.[1]

1. Ibid.

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2. Reynolds, L. L., "Obsolescence of Steam Turbines", Electrical Journal, June 1934

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PAMPHLETS

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IX. APPENDIX

(A) Determination Of The Vacuum Correction Factor Curve

It is common practice in dealing with steam turbine thermodynamics to assume that the cycle of operation is the Rankine heat cycle[1] and that therefore the efficiency of the turbine may be expressed as the percentage of available heat converted into work. The heat converted into work is assumed to be the difference between that contained in the steam at initial temperature,or pressure, and that contained in the vapor at exhaust conditions, as determined by means of any available table of specific heats or a Mollier diagram. It will then be clearly seen that as the vacuum,or back pressure, at which the turbine is exhausting steam determines the amount of heat converted into work, it will also determine the efficiency and is inversely proportional to the water rate. The lower the back pressure, the more heat will be theoretically converted into work, and the higher will be the efficiency of the machine. Therefore, in order to compare the water rates for any specific back pressure, some particular "vacuum" will have to be selected as the standard, and all rates existing at the other pressures will have to be corrected to the rate that would exist at the standard back pressure.

From a steam table and accompanying Mollier diagram by MARKS & DAVIS, the total heat in a pound of steam at 175 lbs. gage (190 lbs. absolute)[2] and 99% saturated is found to be

1. Unless, of course, steam is bled from the stages.
2. The average initial pressure supplied to the Westinghouse turbine.

1188 B.t.u.

At 190 lbs. abs., 99% saturated, TOTAL HEAT= 1188 B.t.u.

AVAILABLE HEAT At 20" vacuum, 1188 - 942 = 246 "

" " " 22" " 1188 - 928 = 260 "

" " " 24" " 1188 - 912 = 276 "

" " " 26" " 1188 - 894 = 294 "

" " " 28" " 1188 - 859 = 329 "

If,as is done in this thesis, 24" vacuum should be selected as the standard back pressure, the water rate existing at other vacuums should be multiplied by a correction factor which is the ratio of the available heat at the existing vacuum to the available heat at the standard vacuum. Thus, for an existing vacuum of 20", the water rate at that vacuum should be multiplied by the fraction 246/276, or 0.892. By this method, the correction factor for all vacuums from 19" to 28" have been calculated and the results plotted in graphical form as is shown in curve SHEET III. The data for this curve are given below.

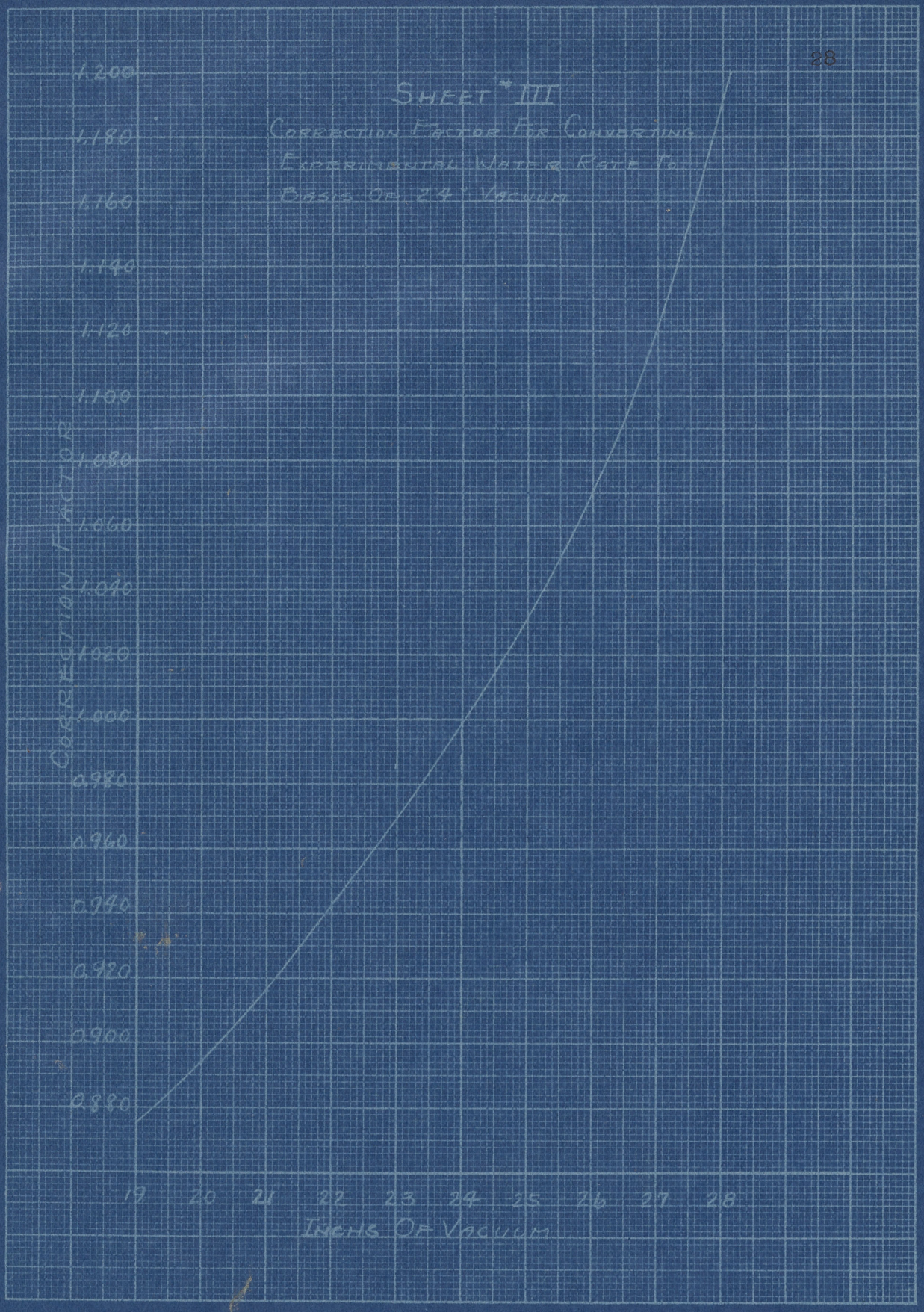
Vacuum	20"	22"	24"	25"	26"	27"	28"
Fraction	$\frac{246}{276}$	$\frac{260}{276}$	$\frac{276}{276}$	$\frac{285}{276}$	$\frac{294}{276}$	$\frac{310}{276}$	$\frac{329}{276}$
Factor	0.892	0.942	1.000	1.032	1.065	1.122	1.192

(B) Calibration Of Cochrane Mechanical Steam Flow Meter

When dealing with any particular flow meter, it is always advisable to refer to the instructions included with the instrument at the time of purchase. As is the case for Cochrane Flow Meters, a complete pamphlet is furnished

Sheet * III

Correction Factor For Converting Experimental Water Rate To Basis Of 24" Vacuum



containing all possible information on the installation, adjustment, and calibration of the meter. Definite instructions on many points are included which should be followed carefully in order to assure accurate results.

Assuming that the metering orifice and piping connections have been installed correctly, it is a simple matter to calibrate the flow meter. With the equalizing valve at the top and back of the meter open so that the pressure from the high and low(upstream and downstream) lines are equalized in a small chamber, the U-tube mercury tubes should balance each other and the indicating and recording arms should rest on zero. These arms may be adjusted by means of set screws if no coinciding exactly with the zero positions. A calibrating weight, furnished with the meter, should be placed on a pin at the upper end of the left(or low pressure) cylinder and the U-tube allowed to come to rest. With the unbalanced effect of this weight, the meter should indicate and record 20% of the meter capacity, if standard. Also, the right hand dial of the counting train, or totalizer, should make one revolution in thirty minutes. A screw adjustment varies the stroke of the clock, or counting, arm and the calibration may be effected without difficulty.

This standard calibration was made on the steam flow meter used in the tests, and an error of less than one percent obtained.

(C) Calibration Of 3" Worthington Piston Water Meter

The piston meter is generally considered to be the best type for measuring the flow of hot water, and if calibrated at the test conditions (temperatures) is within the limits of experimental error. The available apparatus was not sufficient to measure and weigh the condensate in large tanks directly, but the piston meter was calibrated in pounds per unit read from the dial at actual load conditions. Below is given a list of the individual tests. These results show a straight line calibration curve and a constant output of 9.86 pounds per meter unit.

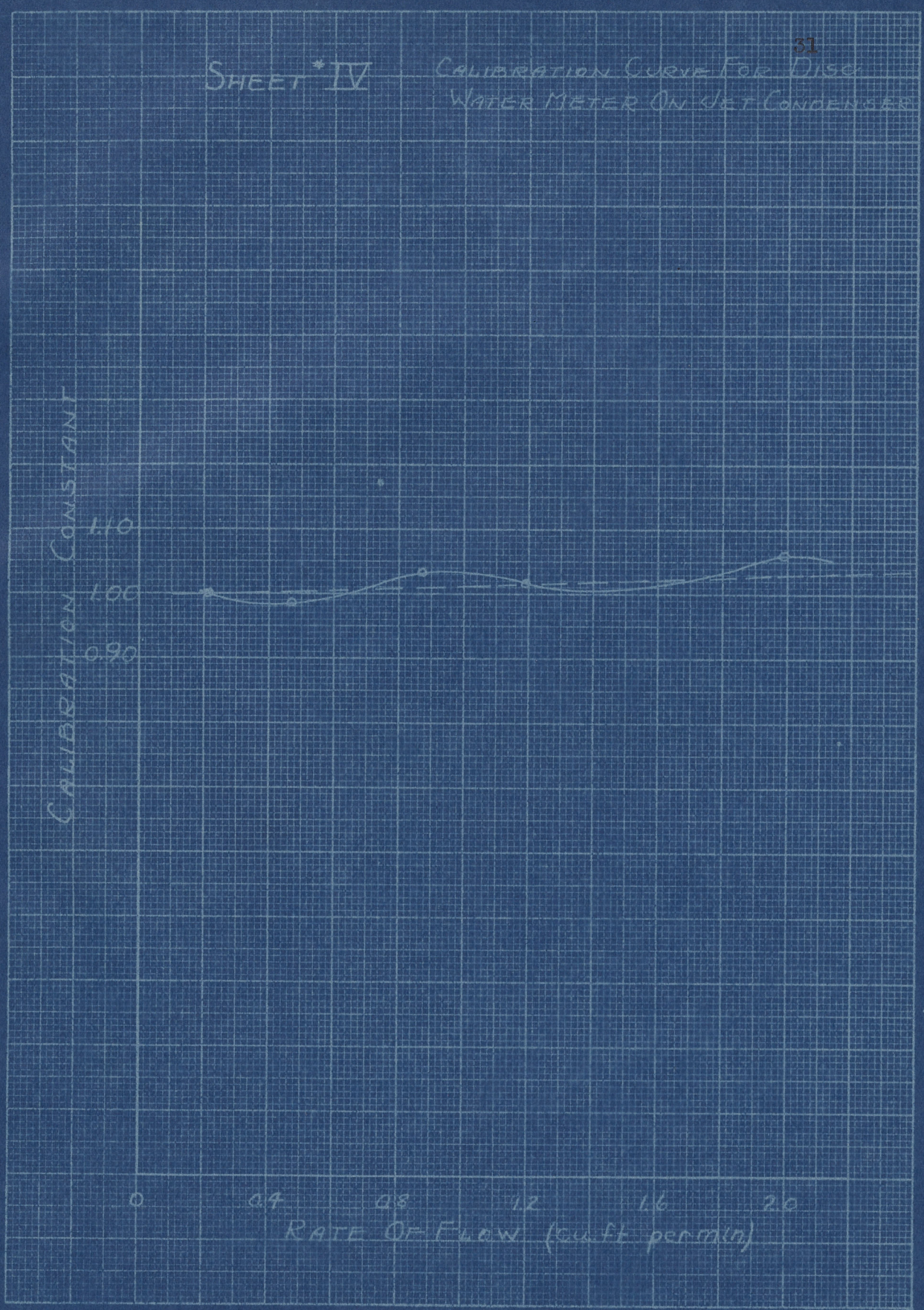
K.W. Load	Temp Cond's.	Meter At Beginning	Meter At End	Meter Units	Wt. Of Water	lbs. per Meter Unit
76	118	474.0	510.0	36.0	350	9.72
120	126	960.0	995.5	35.5	"	9.86
126	124	592.5	628.0	35.5	"	9.86
198	133	082.5	118.0	35.5	"	9.86
220	138	002.0	037.5	35.5	"	9.86
250	156	475.0	510.0	35.0	"	10.00
						9.86

(D) Calibration Of Jet Water Meter (Disc type)

This is a Pittsburg water meter and measures the amount of water used to condense the steam from the first two stages in the turbine (General Electric). In an attempt to calibrate it, the meter was connected onto the city water line and the water flowing through it was caught and weighed on a small platform scale. However, the rates of flow that could be obtained were not as high as those commonly met with in practice, which is probably explained by the fact that in

SHEET * IV

CALIBRATION CURVE FOR DISC WATER METER ON JET CONDENSER



actual use the meter discharged into a region of less than atmospherical pressure. Therefore, the calibration as made in air will not apply very well to the conditions existing in actual tests. The data obtained is given below and the calibration curve is shown on the preceding page. Instantaneous rates and the average rates are shown by the two curves included.

Nº Run	Meter At Beginning	Meter At End	Metered cu.ft.	Weight Of Discharge	Time Of Discharge min.s	Rate Of Discharge cu.ft./min	Discharge cu.ft.	Per Cent Error
1	421.35	422.15	0.80	50	3'- 40"	0.214	50	0.00
2	422.94	424.52	1.58	100	3'- 20"	0.475	98.75	1.25
3	425.37	426.99	1.62	100	1'- 22"	1.190	101.10	1.10
4	434.88	437.00	2.12	125	1'- 04"	1.990	132.50	6.00
5	440.55	442.20	1.65	100	1'- 55"	0.861	103.10	3.10

(E) Calibration Of The Indicating Wattmeter On the Instrument Panel

The indicating wattmeters on the instrument panel were found to check admirably with the totalizing wattmeters, and the check for one of them is given below.

Taking readings every five minutes from the indicating meter, the general average for one hour was compared with the amount recorded on the totalizing wattmeter.

Output= 202 K.W. In 29min.s 45 sec.s On TOTAL meter

" = " " " 1785 sec.s= 0.056 K.W./sec.

Output for one Hour, (3600)(0.056)= 201.8 K.W. **

Indicating Wattmeter:

Output(Averaged for one hour)= 201.6 K.W.

$$\%\text{Error} = \frac{201.8 - 201.6}{201.8} = 0.099\%$$ Negligible ***

* APPROVAL SHEET *

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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