No. 1299*d*

DISCUSSION ON STEAM TURBINES

J. H. Lawrence.[1] The results given by Mr. Naphtaly in Table 1 under the heading B.t.u. per kw-hr. vary considerably from those which I have calculated, and give the turbine a much lower heat consumption than I think it should be credited with. I have tried various methods and have used both the old and the new steam tables but have not been able to check the figures given in the table. Table 2 shows the revised results.

TABLE 2 REVISED SUMMARY OF TESTS ON 10,000-KW. TURBINE

Marks and Davis Tables

Run	Load, Kw.	Steam Pressure, Lb. Gage	Super-heat, Deg. Fahr.	Vacuum, 30 in. Barometer	Water per Kw-hr.	B.t.u. per Kw-hr.	Rankine Cycle, Ratio	Thermal Efficiency, Per cent
A	7972	171	58	28.28	14.581	17035	66.8	20.04
B	8563	168	59	28.18	14.427	16830	68.2	20.27
C	8198	169	60	28.10	14.596	17017	67.6	20.05
D	9173	167	59	27.90	14.572	16926	68.9	20.15
E	5333	173	54	28.34	15.655	18274	61.8	18.67
F	8148	167	60	26.16	15.855	18093	69.5	18.86
G	5401	174	56	26.16	17.611	20075	62.2	16.99

The heat consumption of a turbine is the difference between the heat content of the entering steam and the heat of the liquid in the exhaust steam, multiplied by the pounds of steam per kilowatt-hour. Accordingly, for run *B*, with 168 lb. gage pressure, 59 deg. superheat, 28.18 in. vacuum and 14.427 lb. steam, we get a heat consumption of (1232.7–66.1) 14.427 or 16,830 B.t.u. per kw-hr. This result is almost 12 per cent higher than that given by Mr. Naphtaly. The corresponding thermal efficiency is 20.27 and not 22.65 per cent.

Mr. Naphtaly's figures for Rankine cycle ratio check very closely with those which I have calculated.

[1] Student Mem. Am. Soc. M. E., Cornell University.

Presented at the Annual Meeting, New York 1910, of The American Society of Mechanical Engineers.

D. S. Jacobus. It would be interesting to have the views of some gas engine advocates in respect to the B.t.u. consumption obtained in the tests as compared with what they would expect under similar conditions for a gas-engine plant. The figures would indicate that the B.t.u. consumption per kw-hr. is low as was also the case in the tests of the oil-burning plant at Redondo, Cal., the results of which have already been reported to the Society. In these tests a kilowatt-hour was turned out at the switchboard with a consumption of about 25,000 B.t.u., based on the total heat of combustion of the fuel burned, all the oil burned for miscellaneous purposes about the plant being included, as well as that for carrying the boilers over a $4\frac{1}{2}$ hr. lay-over period.

Chas. H. Parker[1] and I. E. Moultrop. Referring to Mr. Varney's paper, it is interesting to learn that the steam turbine has operated satisfactorily when compared with a large number of water power stations and its service has been more satisfactory in regulation and maintenance than that obtained from either steam engines or gas engines. The engineers of the Pacific Gas and Electric Company are to be congratulated on the speed with which a 9000-kw. steam generating station was completely built and also on the very low cost of this station per kilowatt of capacity. If Mr. Varney will state briefly in his closure how this very low cost was obtained it will add much interest to his paper.

The author states that the test showed the water rate of the turbine to be inside the manufacturer's guarantee, but it would seem that the actual performance was not especially creditable to a modern Curtis turbine of 9000-kw. maximum capacity. Of course, the vacuum during the test was not especially good and with injection water of 51 deg. to 52 deg. fahr., it should be possible to reduce the absolute back-pressure reported in the test by about 1 in., which would improve very much the water rate of the turbine.

The superheat of 70 deg. to 75 deg. on the maximum load test is also rather low for the best results. Experience has proved that 150 deg. fahr. of superheat is safe for superheaters and piping and this increased quantity would better the water rate some 6 per cent.

Either or both of these changes might have increased the investment by a small amount. However, a gain of about $1\frac{3}{4}$ lb. in the water rate per kw-hr. would seem to be easily obtainable and would justify

[1] Boston Edison Co.

a considerable investment charge. It is not usual for a maximum-load test to show better results than a normal-load test. The General Electric Company guarantee usually calls for about ½-lb. poorer water rate at maximum than at normal load.

Mr. Naphtaly has contributed some very valuable data upon the Westinghouse double-flow turbine, a type of machine about which very little information has been published. The unit is large, having a normal rating of 10,000 kw. and an overload capacity of 50 per cent. As the rise of temperature on the generator is only 60 deg. cent. after a 24-hours' run at 1½ load, this unit might well be given a rating of 15,000 kw. continuous capacity. It seems from the test that this unit is very efficient, especially at the low superheat and comparatively poor vacuum used during this test.

The vacuum correction curve is very instructive, as it shows a very small gain from an increase of vacuum, thereby differing widely from an impulse turbine. The shape of the vacuum correction curve would indicate that the effect of increased vacuum at overloads would be very slight.

It is very unfortunate that the requirements of the station would not permit the test being carried to full overload of the turbine, as it would be interesting to see how the opening of the secondary steam valves would affect line of total water consumption, and a test for effect of change in vacuum at overload would give a third point on the curve that would be most instructive.

A 3500-kw. horizontal Curtis turbine in a plant of the Great Western Power Company at Oakland, Cal., was tested at 3464 kw. load, 194.35 lb. pressure at boilers, 127.4 deg. superheat, 1.17 absolute back-pressure, inches of mercury. The water rate was 13.62 lb. per kw-hr. For the size of the unit this seems very good.

At the L Street Station of the Boston Edison Company, a 5000-kw., 5-stage, Curtis turbine unit tested in 1907 showed the following:

Load, kw	2558	5195	7526
Steam pressure, lb. gage	173.0	173.7	169.7
Superheat, deg. fahr	149	142	134
Vacuum absolute back-pressure, in. mercury	1.13	1.18	1.35
Water per kw-hr., lb	15.24	13.57	13.73

This same machine has been tested once a month since being installed and the best record of actual water rate came with the highest superheat and best vacuum as follows: 5095 kw. load, 170.4 lb. steam pressure at throttle, 179.1 deg. fahr. superheat, 0.52 in. of mercury back-pressure, 12.71 lb. of water per kw-hr.

The water rate of this test, corrected to standard conditions, is slightly poorer than in the test made at the L Street Station, so that any possible error in the latter test is in the direction of making the water rate too poor rather than too good. These figures seem to show conclusively the advantage of high superheat and vacuum, at least on the impulse type of turbine. It is the overall efficiency of the plant for which engineers should strive and this is a question of the greatest output for a pound of coal. Of course the investment needed to secure the increased efficiency must be justified by the saving in operating.

A glance at the correction curves used with the various units spoken of is most interesting, as they suggest the possibilities of the various kinds of turbines under different conditions of superheat and vacuum:

Type	G. E. Vert.	West. D'ble-Flow	G. E. Vert.	G. E. Vert.
Company	Pac. G. & E.	City Elec. Co.	N.Y. Edison	Boston
Station	Oakland	San Francisco	New York	L Street
Maximum capacity of unit, kw	9000			
Normal capacity of unit, kw		10,000	8000	5000
Corrections for pressure, 1 per cent for every	10 lb.	20 lb.		10 lb.
Superheat, 1 per cent for every	12½ deg.	10 deg.	12 deg.	12 deg.
Vacuum, normal load, per in.	1.08 lb.	0.45 lb.	0.985 lb.	1.15 lb.
Vacuum, ½ load, per in.		0.91 lb.	1.35 lb.	1.60 lb.

These figures would indicate that the impulse turbine received more benefit from an increased pressure than the combined impulse-reaction machine, which is very curious, to say the least, as the high-pressure elements are of the same type. The benefit from an increase in superheat is marked in both the types, with the advantage slightly in favor of the double-flow type. The benefit from an increase in vacuum is very much less in the double-flow combined impulse-reaction turbine than in the straight impulse turbine, especially at the heavier loads, which is 42 per cent as much at full load and 62 per cent at ½ load.

The designer should provide for a moderately high superheat and as good a vacuum as the inlet water temperature will warrant, and then the operating engineer should continually maintain the condensing apparatus in such order as to get the best vacuum obtainable with the apparatus furnished him. This is true with both types of turbines discussed, but is especially true of the straight-impulse turbine.

Geo. A. Orrok. The Society is to be congratulated on the papers of Messrs. Naphtaly and Varney, which place before the membership actual tests of the most modern types of power-generating apparatus. It is interesting in this connection to compare these results with those of other turbine installations, published tests of which have been available during the last few years. I have had made such a table, recalculating the tests on the same basis, using the new Marks and Davis steam tables. I have neglected the Rankine cycle efficiencies, since in most cases the efficiency of the generator and the mechanical efficiency of the turbine are not given, and cannot be easily obtained.

TABLE 3 ECONOMY TESTS OF STEAM TURBINES

Marks and Davis Steam Tables

Turbine	Load, Kw.	Steam Pressure, Lb. Abs.	Super-heat	Vacuum	Steam, Lb. per Kw-hr.	B.t.u. per Kw-hr.
Rummelsburg, A. E. G.	4179.6	179	289.3	29.19	11.95	15665
Carville, Parsons	5164.1	207	125.2	29	13.15	16122
Chicago, Curtis	10816	190	147	29.47	12.9	16206
Berlin Moabit, A. E. G.	3150	185	225	28.5	13	16352
Boston, Curtis	5195	179.5	142	28.8	13.52	16578
City Electric, San Francisco, Westinghouse	8563	183	59	28.18	14.427	16850
N. Y. E., Westinghouse	9830	192.2	96	27.31	15.15	17778
Brown-Boveri, Rhenish Westfalen	5128	176	193.7	28.47	14.32	17790
N. Y. E., Curtis	8880	192.5	108.5	28.1	15.05	17940
Pacific Curtis, Oakland	8775	194	72.95	28.03	15.95	18735

It is much simpler to compare the various machines on the basis of the B. t.u. in the steam at the throttle valve, and to credit the turbine which that portion of the heat turned back into the feed water system at the condenser temperature. This fact is what power producers are interested in, as the B.t.u. so calculated represents what they are paying for in coal.

In making up Table 3 I have used in the actual test result, pounds of steam per kilowatt-hour, vacuum, superheat and steam pressure not corrected to the contract conditions.

E. D. Dreyfus. The experience obtained from over 2000 steam turbines in service should now reveal the operating conditions and design which realizes the highest final efficiency. Uniformity of operating conditions has already become evident, and very probably a general accepted type will eventually be prevalent. These facts have already been the subject of both favorable comment and noteworthy action in this direction.

Mr. Naphtaly is to be commended both for having selected moderate working conditions by means of which he obtains the highest ultimate economy and for having endorsed an increase in rotative speeds which obviously conduces to higher efficiencies.

Fig. 15 demonstrates that 100 deg. fahr. is about the upper limit of superheating that may be profitably employed. In constructing this diagram, care has been taken to make all of the assumptions favor high superheat as much as possible, so that I believe it presents the most optimistic view of the subject.

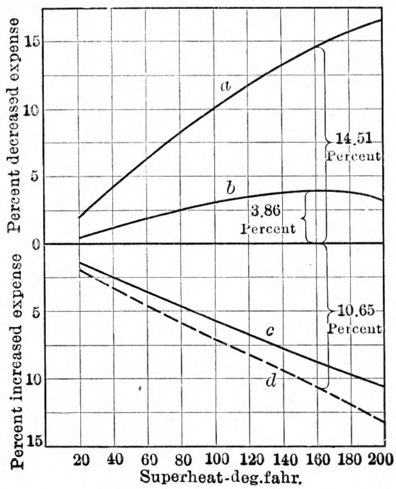

FIG. 15 INCREASE OF EXPENSE WITH INCREASE OF SUPERHEAT

Full load operation 175 lb. steam pressure, 210 deg. feed water temperature, C_p values from Knoblauch & Jakob. Investment and maintenance charge pro-rated assuming equal capital and fuel expense.

a Per cent decrease in turbine water rate
b Net economic gain not allowing for increased radiation losses
c Increased heat addition to steam
d Total increased expense, including investment and maintenance charges

While the water rates which Mr. Naphtaly obtained with his turbine (14.57 lb. as tested, and 13.88 lb. as corrected) somewhat exceed other steam consumptions that have been recorded, there has, nevertheless, been established a new mark in the efficiency of conversion of the available thermal energy between the throttle and exhaust pressures into electrical energy at the switchboard: viz., 69 per cent as referred to the Rankine cycle.

Very few of us realize, initially, the true relative effect of different operating conditions. Any changes in steam pressure, vacuum or superheat, as affecting the available energy, is shown at a glance in a temperature-entropy diagram as reproduced in Fig. 16. For example, with initial conditions of 190 lb. and 100 deg. fahr. superheat, ½ lb. reduction in pressure at the exhaust end from 28 to 29 in. vacuum, increases the available energy as much as the addition of over 100 lb. at the inlet of the first stage.

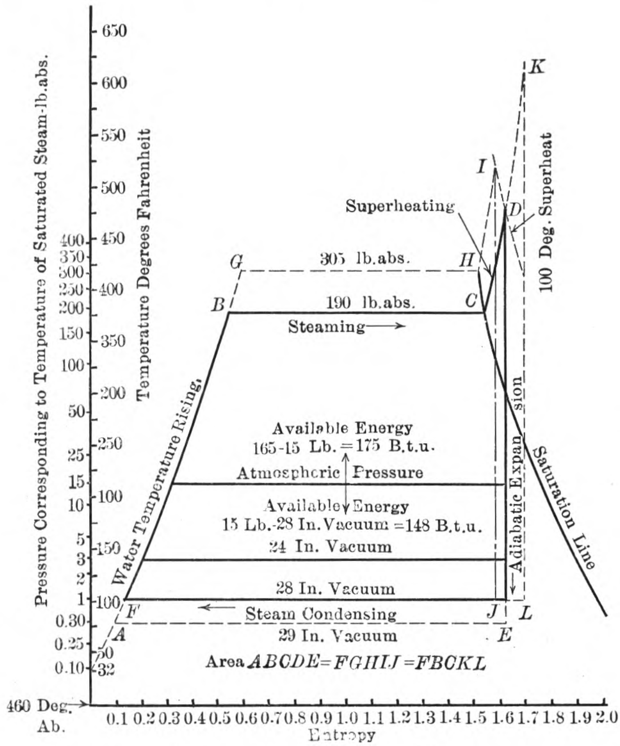

FIG. 16 TEMPERATURE-ENTROPY DIAGRAM

The above efficiency, as already noted, has not been equalled in any other published test. It is to be remembered that this percentage includes all armature and windage losses of the generator, as well as the field losses which are commonly omitted. Table 4 gives the best American and European turbine performances that are on record.

Although some show lower steam consumption, due either to a higher superheat or to vacuum, or to both, in every instance the efficiency ratio falls considerably below that of the City Electric turbine, which is truly complimentary to American development. The extrapolated result of 71 per cent at maximum load on the primary valve stands out even more prominently.

TABLE 4 STEAM CONSUMPTION AND EFFICIENCIES OF STEAM TURBINES

	Type	Name	Load, Kw.	R.p.m.	Boiler Pressure, Lb. per Sq. In. Gage.	Superheat, Deg. Fahr.	Vacuum, In. Mercury, 30 In. Standard	Water Rate, Lb. per Kw-Hr.	Rankine Cycle Efficiency*	Authority
AMERICAN	Reaction Westinghouse	New York Edison.........	9870	750	177.0	97.0	27.31	15.0	67.3	Operating Co.
		Brooklyn Rapid Transit..	11466	750	177.0	106.0	28.15	14.45	65.72	Operating Co.
		City Electric Co., San Francisco, Cal...........	9173	1800	167.0	59.0	27.9	14.57	69.0	J. G. White & Co. Tests
	Impulse Curtis	New York Edison.........	8880		177.8	108.5	28.1	15.05	63.1	Geo. A. Orrok
		Commonwealth Edison....	10186	750	176.0	147.0	29.47	12.9	62.1	Consulting engineers
		Pacific Gas & Elec.........	8775	750	179.3	72.95	28.03	15.95	61.0	F. H. Varney
EUROPEAN	Reaction	C. A. Parsons, Carville...	5059	1200	194	92	29.27	13.35	62.9	Z. d.V.d.I.† 1907, p. 1122
		C. A. Parsons, Chelsea....	6000	1000	190	140	28.00	14.09	65.8	Test result
		Brown-Boveri, Frankfurt.	3522	1360	142	136	28.92	13.68	65.8	Z.d.V.d.I. 1908, p. 516
	Impulse	Escher Wyss Co., Essen...	5118	1025	146	158	27.63	15.16	65.1	Z.d.V.d.I. 1908, p. 1436
		Escher Wyss Co., Turin...	3540	1485	172	105	28.29	15.10	64.4	Z.d.V.d.I. 1908, p. 1437
		A. E. G. Moabit, Berlin....	3169	1500	170	217	29.19	12.74	6[illegible].5	Z.d.V.d.I. 1907, p. 386
		A. E. G. Rummelsburg...	4239	1497	173	285	29.19	11.90	65.1	Z.d.V.d.I. 1909, p. 762
		D. Howden & Co., Manchester..................	7520	1000	184	133	27.55	14.50	67.0	The London Engineer, 1909, p. 462
		D. Howden & Co., Manchester..................	6383	1000	188	137	27.44	14.30	67.6	The London Engineer, 1909, p. 462

*Peabody's Steam Tables, 1909 Edition, used in calculating Rankine cycle efficiency on kw-hr. basis.
†Zeitschrift des Vereins deutscher Ingenieure.

It is believed that the efficiency ratio consistently improves until 100 deg. fahr. superheat is exceeded, when further delay of the dew point of occurrence of saturation is offset by the greater amount of heat in 1 lb. of steam passing out to the exhaust. Hence it is not improbable that the turbine would have accomplished a slightly better record with additional superheating, 60 deg. being the average throughout the test. The effect of vacuum is more indefinite and is subject to the areas and angles provided in the final stages of the turbine.

It is sometimes indiscriminately stated that the higher the vacuum the better the overall plant results. This is plainly a narrow perspective as the power consumption of the condenser auxiliaries is thus entirely ignored. Condenser temperatures fall off very rapidly with the increase of vacuum, as is shown by the standard steam tables, and the corresponding capacity and power required by the air and circulating pumps becomes proportionately greater. There is consequently a

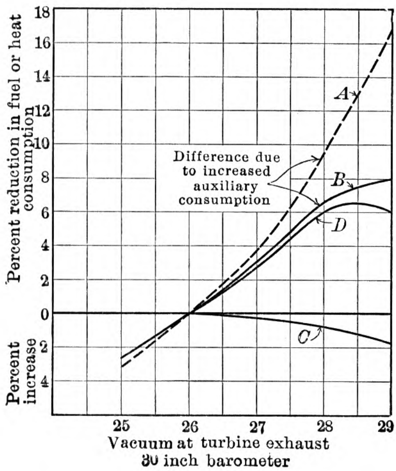

FIG. 17 RELATIVE VALUE OF DIFFERENT VACUA ON ULTIMATE PLANT ECONOMY BASES

200-kw. turbine using surface condensers supplied with average injection water at 55 deg. fahr. Steam conditions 175 lb., 100 deg. superheat. Most economic arrangement of auxiliaries selected for each vacuum to give best heat balance. Relation between investment and fuel economy pro-rated on the assumption of the fixed charges being one-half of fuel expense burning $3 coal.

A = Actual reduction at the turbine

B = Net reduction in plant fuel consumption

C = Equated cost (greater fixed charges and maintenance) of obtaining the higher vacuum

D = Final plant improvement

point where the improvement in the turbine fails to exceed the increased power demanded by the auxiliaries, and in the latitudes we are accustomed to, we generally find the most economical yearly average vacuum varies between 28 in. and 28.5 in. (referred to 30 in. barometer), depending upon the source of water supply.

This is apparently the upper economic limit when the falling off in vacuum with periodic fouling of tubes is considered. Moreover, the deductions were based on uniform efficiency ratios of the turbines for all vacua, which evidently encourage a higher limit. It is to be noted that all the station auxiliaries were accounted for in deriving the best heat balance. Although larger units than those treated in Fig. 17 may have slightly increased advantages owing to a better arrangement of auxiliaries, they do not tend to produce improved plant economies beyond the limits already expressed when all modifying conditions are borne in mind. For this reason, the order given in Table 3 is not an equitable reckoning, since the vacuum factors are disregarded.

I am at variance in opinion with Mr. Moultrop in regard to the merits attached to the effect of change in vacuum on the economy of the different types of turbines. I do not believe the manner in which he has presented this phase portrays the essential considerations. It is patent that the reaction and impulse turbines may be readily designed for the same efficiency for any given vacuum whatever, virtually placing them on a parity in so far as realizing on a fixed high vacuum is concerned. Regarding fluctuations in operating conditions, his comparison correctly shows that the straight-impulse machine with the fewer rows of blades both gains and loses most with changing vacuum, and consequently is more sensitive to such variations and will more rapidly decline in efficiency if the auxiliary station equipment is not maintained up to the original standard.

Doubling the rotative speed reduces the steam consumption from 4 per cent to 6 per cent, depending upon the size. In 1000-kw. turbines of 3600 and 1800 r.p.m. under U. S. Government tests, a difference of 4.5 per cent was shown. Large turbines, such as those under discussion, would probably show somewhat better results.

While fuel economy is an important factor of the total cost of power, investment and operating expenses must also be reckoned with. Fortunately I have been able to obtain what I believe may be considered conservative installations and monthly operating costs from Mr. Naphtaly's plant. These results plainly indicate efficient management, both in construction and subsequent operation. The entire station consists of two 2500-kw., and one 6000-kw. turbine, in addition to the 10,000-kw. unit. A typical monthly payroll was $3565; oil, waste, supplies, water and maintenance material averaged for three months, $769; the corresponding oil-fuel consumption, for the entire station 20,800 bbl., and the output, 4,908,800 kw. The smaller units increase the amount of wages, and therefore only three-fourths, or less, is strictly chargeable on the base of the use of large units.

Considering the new addition provided by the installation of the 10,000-kw. turbine, the cost to the City Electric Company may be conservatively quoted at $339,413 or $33.94 per kw., including building, foundations, boilers, piping conduits, oil storage and burning equipment, auxiliaries and electrical apparatus—in fact, everything from the cost of real estate up to and including the switching gear.

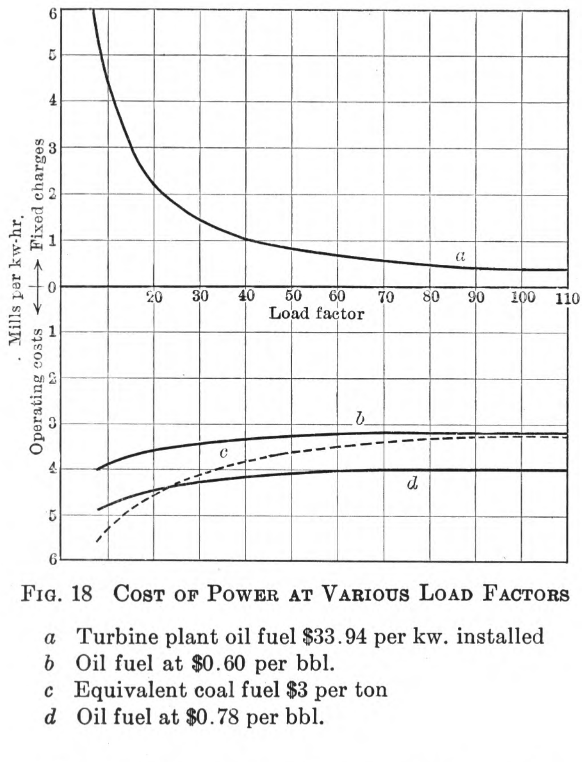

FIG. 18 COST OF POWER AT VARIOUS LOAD FACTORS

a Turbine plant oil fuel $33.94 per kw. installed
b Oil fuel at $0.60 per bbl.
c Equivalent coal fuel $3 per ton
d Oil fuel at $0.78 per bbl.

The City Electric expense per unit generated at various load factors, is similarly included in Fig. 18—four mills, total, with the low oil cost, and five mills with a higher oil cost, which is certainly excellent. Oil burning naturally affects the investment by the elimination of bunkers and coal-handling apparatus with corresponding heavy foundations and also entails less boiler-room labor. A reduction (Fig. 18) to eastern conditions has, therefore, been carefully made on the basis of 14,500 B.t.u. coal at $3 per ton delivered, fully accounting for the various items involved. The investment would hardly be increased more than 30 per cent at the utmost for coal-handling machinery and bunkers.

The low total cost of a complete-expansion turbine, excluding investment and cost of labor, settles the question definitely as to the advisability of introducing a combined engine-turbine unit in a new station to accomplish a small saving in fuel.

Another feature of interest refers to the floor space occupied. By employing the higher speed of 1800 r.p.m., instead of 900, the unit has been reduced in length about 17 per cent. Mr. Naphtaly developed his plans providing for two rows of turbines in the engine room, having them alternately faced with respect to one another. The 10,000-kw. turbines of the type described demand an engine-room floor space of only 0.18 sq. ft. per kw. of normal capacity installed, which figure it is thought also constitutes a new record.

R. J. S. Pigott. It is fortunate that manufacturers and owners of turbines are becoming more broad-minded about publishing data on tests. Unfortunately, many of these are rendered less valuable by the omission of full data, particularly the bases of computation. A case in point is the computation of B.t.u. per kw-hr. and efficiencies (Table 1). These figures could not be checked from the data in the paper.

In Par. 17 Mr. Naphtaly compares his machine, which is really 15,000-kw. maximum rating, with the combined unit at the Interborough Rapid Transit Company's 59th Street Power Station. His comparison is based on 60 deg. superheat on his machine against saturated steam at the 59th Street Station. The thermal superiority of the combined unit is 8.5 per cent on this basis. There is nothing, however, to prevent the use of superheat on the combined unit. A fair basis is using superheat in both cases.

With 60 deg. superheat, the 59th Street unit water rate at best load would be 12.3 lb. per kw. instead of 13.2 lb., which brings the advantage of the combined unit to 13.5 per cent, about the same as given in the paper[1] by Mr. Stott and myself on this unit. But taking the conditions just as they stand, if the units are operated on normal load the saving on coal and water by the more economical combined unit, even on saturated steam, averages approximately $80 per day. The extra attendance is $8 per day; the additional fixed charge $12 per day; and the maintenance charge $7 or $8 per day over that

[1] Trans. Am. Soc. M. E., vol. 32, p. 69.

required by the turbine installation. This still leaves a very substantial margin saved by the combined unit. These figures are based on a cost of $65 per kw. maximum rating for the high-pressure turbine system, and $76 for the combined unit, a somewhat conservative figure. In spite of this considerable advantage of the low-pressure turbine system, it is highly improbable that new plants of the combined type will be built, since the high-pressure turbine is improving almost daily in efficiency.

It would have been interesting if Mr. Varney had itemized his cost of $51 per kw., which is evidently based on maximum rating. When the lack of coal-handling apparatus and absence of land charge is taken into consideration, the total cost figures nearly up to the average cost of other turbine plants.

W. L. R. Emmet. The results reported in Mr. Varney's paper are about 1 lb. per kw-hr. worse than the results which Mr. Moultrop has reported on his Boston turbine, which is identical with the machine reported in Mr. Varney's paper, two or three of which have been tested with uniform results. The machine in question was tested under station conditions after having been in operation about a year and the possibilities of considerable errors in such tests are very well known.

Fig. 2 of Mr. Naphtaly's paper shows a cross-section of the Westinghouse turbine on which the test was made. The peculiarity of this machine, as would naturally be expected at such a high speed, is the small area for the discharge steam at the low-pressure end. These low-pressure blades are carried by a drum which is under considerable strain, due to its own weight, and the load which the buckets impose. Such strains must be conservative to prevent loosening of the drum, which would result in unsteady running. To avoid this the discharge area has been made very small in order to reduce the height of the blades and a rough calculation indicates that with the turbine at a load of 9000 kw. the steam was issuing from the final blades at a speed of about 1400 ft. per sec., which is equivalent to a loss of about 9 per cent of the total energy of the steam. This condition is not only indicated by the dimensions, but is shown by the rate of improvement of vacuum reported. When these facts are considered, the supposition that this machine would give a very much better efficiency at a greater load is easily seen to be fallacious. This machine when operated on good vacuum should be rated at 7000 kw. instead of 10,000 kw. and with a suitable proportioning of the upper part of the turbine, it should give very good economy at such a load

The economy reported is extremely good in all the tests and it seems probable that many of the readings are fairly correct, although I am inclined to doubt the one that gives 69 per cent at heavy load because the dimensions do not seem to render it possible. It may be that this test is as much favorable to this machine as the other one was unfavorable to the Curtis.

The question of the value of vacuum is a very important one to all users of most turbines. It is easy to build machines suited to low vacuum, and if no regard is paid to efficiency, the ratings may be increased indefinitely, thereby reducing the cost.

If the dimensions are proportioned conscientiously for high vacuum, many difficulties are encountered with high speed. Mr. Varney's machine gained over 6 per cent in water rate for a variation of 1 in. of vacuum when operated at full load, while Mr. Naphtaly's machine gained 3 per cent. Any turbine can be proportioned for a high degree of Rankine efficiency with low vacuum, but as the efficiency curve falls we fail to get a full compensation for the falling temperature of the condensed water. In most good turbines the limit of desirable vacuum lies above 29 in. Mr. Naphtaly's machine, at 28 in. vacuum and 8000 kw., should be a highly efficient unit, but with higher vacuum it should be relatively inefficient and its efficiency would also decline rapidly with high loads if the vacuum is high.

It is often stated that records of turbine tests in Europe are better than those in this country and that the Curtis machines made by the A. E. G. are better than American Curtis machines. In point of fact, the A. E. G. turbines at Berlin, which have been tested over and over again, run about a quarter of a pound worse than a five-stage 3000-kw. and a six-stage 3000-kw. machine under exactly the same conditions in this country.

I am not a strong advocate of superheat because I believe the gain in actual fuel is small, but if superheat is to be employed, my experience shows that a great deal can be used. Some machines are endangered by superheat while others are not. The Curtis turbine is in the latter class and superheat as high as 200 deg. may be used with safety.

W. L. Waters. The history of the commercial steam turbine unit for power station work has been one of gradually increasing speeds, the advantages of high rotational speed being improved steam economy and reduced floor space. This has resulted in a continual demand from the steam turbine builders for generators of higher speed to be direct connected to the turbines; and the degree to which this increase in speed has been carried can be seen from the fact that a standard 2500 kw., 60-cycle unit which five years ago ran at 1200 r.p.m., today operates at 3600 r.p.m. The difficulties in the design of high-speed alternating-current generators are due to the increased centrifugal stresses resulting from the higher speed and to the danger of excessive vibration caused by the relative motion of the component parts of the rotating field magnet structure. As Mr. Emmett states, it is possible to adopt a comparatively cheap and simple mechanical design for a slow-speed unit and to manufacture from standard material with average workmanship. With high-speed units, on the other hand, it is necessary to introduce greater accuracy and refinement in the design and to use high-grade material and skilled workmanship in the manufacture, in order to obtain the same factor of safety and operating characteristics as can be obtained in the corresponding low-speed units.

It has been the general experience that when these precautions are taken, high-speed units are as satisfactory if not more so, from an operating standpoint, than the corresponding low-speed ones; and a 3000-kw. 3600-r.p.m. generator carefully designed and built, or one of 10,000 kw. at 1800 r.p.m., is a better operating machine and has a higher factor of safety than a corresponding unit of one-half or one-third the speed.

Mr. Naphtaly's paper is of interest because it describes one of the first of the large modern high-speed steam turbine units and gives tests made by independent engineers of high standing. Since the results of these tests are so satisfactory and the unit has been so successfully operated, it is probable that the old standard slow-speed machine will be superseded by considerably higher speed units similar to the 10,000-kw. turbine at the City Electric Company at San Francisco.

Francis Hodgkinson. Inasmuch as I was identified with the designing and building of the turbine which is the subject of Mr. Naphtaly's paper, and also had the pleasure of being present when the tests were made, I can therefore reply to some of the points brought up in the discussion.

Although it is true that the tests were not scientific in the sense that the losses were all segregated and the like, I wish, however, to emphasize strongly the fact that no stone was left unturned either by the owners or by the engineers conducting them, in the effort to insure accuracy of the tests.

I am in accord with Mr. Moultrop in deploring the fact that loads heavier than 9000 kw. could not have been carried during these tests.

Regarding the efficiencies referred to in Mr. Orrok's discussion, viz., the efficiency ratio, or Rankine Cycle efficiency, as it is variously called, and the other which Mr. Orrok prefers, viz., the B.t.u. measured in the steam at the throttle above water at the temperature corresponding to the pressure within the condenser, I agree with him that the latter is perhaps the more expressive for the user of the turbine to employ. However, it lacks particular significance, for it takes no cognizance of the power consumed by the auxiliaries. Large condensers requiring considerable power might be employed, giving a high vacuum, when a lower number of B.t.u. per kw-hr. would be shown for the turbine, which in cases might be entirely offset by the large amount of power taken to operate the condenser. It is, of course, understood that as long as the exhaust steam from the auxiliaries is condensed in a feed heater, the power taken by the auxiliaries is of little moment. However, carrying this condenser consideration to extremes, the condenser pumps might absorb as much energy as the main unit, when surely a low B.t.u. rate per kw-hr. would be of but little use to the power plant user.

For the builder and others interested in comparing the performance of one turbine with that of another, I find the efficiency ratio (the ratio of the energy actually obtained from a given amount of steam expanding between certain pressure limits and what is theoretically obtained from an ideal engine) the most convenient. This efficiency should certainly include the mechanical losses of the turbine and there is no reason why it should not also include the losses of the generator.

Comparing the turbine performance on an efficiency ratio basis, a broad view of the operating conditions must still be taken, i.e., equally good turbines may not have quite so high an efficiency ratio at a very high vacuum as for a moderate vacuum. The efficiency ratio is also varied by the amount of superheat, as already pointed out. Both Messrs. Emmet and Moultrop have called attention to the low vacuum correction on this machine as compared with the type of impulse turbines with which they are familiar. The vacuum correction for any type of machine is a variable factor, depending entirely upon the proportioning of the low-pressure blading, and the reaction type of turbine may be designed to give as high a vacuum correction as any other type. This is exemplified in Table 4, in the case of the Carville (Parsons) turbine, where an efficiency ratio of 62 per cent was obtained with the turbine carrying 29.29-in. vacuum and but 90 deg. superheat.

The wisdom of designing commercial turbines for very high vacua would seem to me questionable, since in average power plant operation, 28 in., or certainly 29 in., vacuum is exceptional. Mr. Dreyfus raised the point and expressed the opinion that 28 in. to 28.5 in. average vacuum during the year is somewhere near the limit which may profitably be employed. There is certainly a limit somewhere, dependent upon the average temperature of the cooling water available and the head to which it has to be pumped, since any steam used to operate the pumps, the exhaust from which would be in excess of that required for heating the feed water, would be a direct loss chargeable to the main unit. These are matters depending upon the local conditions and must be individually reckoned with.

Regarding the proportioning of the blading in Mr. Naphtaly's machine, Mr. Emmet has undertaken to scale a photographic reproduction of a drawing, from which he attempts to calculate the terminal steam velocity in the turbine. In this, I think he is taking a great deal for granted, for the actual figures are, in fact, considerably lower than he states. The steam velocities at the low-pressure end of this turbine, however, have a higher ratio to bucket velocity than has been the usual practice, and this fact Mr. Emmet very properly points out as accounting for the low vacuum correction. Had this turbine been designed with low-pressure drums of larger diameter, as it could have been without encountering the mechanical difficulty of loosening the drums, as Mr. Emmet fears, these losses, due to the steam velocity issuing from the final blades, which he says amount to about 9 per cent of the total energy of the steam, would have largely been avoided. Then, it would certainly be interesting to inquire, in view of the results shown by the tests, what would have been the performance of this turbine with material reduction in these alleged losses. However, with the low vacuum corrections determined from the tests themselves, correcting run *D* in Mr. Naphtaly's tests to the best vacuum condition of the L Street Station of the Boston Edison Company, as quoted by Mr. Moultrop, viz., 0.52 in. absolute mercury pressure, the City Electric machine would give 12.12 lb. of steam per kw-hr., taking the superheat correction of 1 per cent for each 10 deg., and 12.4 lb. per kw-hr. if we take the superheat correction of 1 per cent for each 12 deg., the load being 9173 kw., which is less than the normal rating of the turbine.

Similarly, the City Electric turbine reduced to the condition cited for the Great Western Power Company's turbine at Oakland, Cal., gives 13.3 and 13.5 lb. per kw-hr. for each superheat correction, respectively.

Attention should be called to one point, viz., the number of tests conducted on this turbine, which when plotted, show a general agreement. I think no tests should be regarded as entirely reliable without checking the total amounts of steam in accordance with the Willans right line law. If the tests as carried out on this turbine are correct with 11,000-kw. load, the performance would assuredly be in accordance with Fig. 4. At the time the tests were carried out, observations were made during certain swings of load up to 10,000 kw. and above, to determine the pressure drop through the admission valves, and it was found that the turbine was easily capable of carrying 11,000 kw. on the primary valve with 175-lb. gage pressure at the throttle.

CLOSURES

Sam. L. Naphtaly. The B.t.u. per kw-hr. given in Table 1 was figured on the basis of feed water at 210 deg. fahr., arbitrarily reasoning that a credit is to be allowed in the steam plant due to the return of heat from the auxiliaries through the feed-water heater, while on the other hand, no similar operation exists in the gas plant. Inadvertently, the denoting of the B.t.u. per kw-hr. above 210 deg. fahr. was omitted in transcribing the tables. This column was included only incidentally, but for a true comparison of gas and steam plants, it is obvious that all factors in their operation must be duly considered.

In considering steam stations only, manifestly the heat directly chargeable to the turbine should be measured above the condenser discharge water temperature, or more precisely, exhaust temperature, to avoid being affected by any inefficiency of the condenser. Evidently, it is not always safe to adopt the last mentioned method, since the merits of different turbine installations may thus be confused in a manner such that the plant producing the best net overall results, including the auxiliary power, would not be ranked accordingly. The turbine with unnecessarily large and possibly wasteful auxiliaries, or one which had had the advantage of being tested in winter over that of another tested in summer (and correspondingly influenced by a better vacuum), may very likely be unjustly accredited with a better record. One discussor gave this phase some consideration graphically.

Mr. Emmet remarked that the tests which I have reported are more favorable to the turbine I am operating than like tests of the Curtis turbine at Oakland. An inspection of the two different test logs, although plotted to a slightly different scale, will reveal that the conditions were, for all practical purposes, virtually the same, resulting in a difference of 1 lb. in favor of the Westinghouse unit.

Referring to the comparison which Mr. Pigott discusses, authorities seem to differ appreciably as to the value of superheat in the reciprocating engine. I believe some engine experts assert 6 per cent or possibly 7 per cent improvement in the economy of the engine when steam is superheated to 100 deg. fahr., which is much less than Mr. Pigott's figures. Moreover, it does not appear consistent to correct the Interborough combined unit performance for superheat, and on the other hand, vary the City Electric turbine economy to the vacuum obtaining during the Interborough plant tests. Had all corrections been made in the same direction, i.e., for the City Electric vacuum, as well as superheat, as seems most logical in this discussion, the results would agree practically with the smaller difference noted.

It may be of interest to note that this turbine has since been operating with practically the same degree of efficiency as on the tests. While we have not taken the actual water rate, we have kept a very close record of the oil consumption of the plant during the hours this turbine operated, and from that we draw the conclusion as suggested above.

Further, and in order to clear up a question which arose in the discussion, concerning the capacity of the turbine, I would say that the plant was called upon during the winter to put out upwards of 12,000 kw. The inlet pressure recorded was practically in accordance with Fig. 4, in that with 175-gage pressure and at 11,100 kw., the drop through the admission valve was as plotted. The auxiliary valve opened, however, on loads above 11,100 kw.

F. H. Varney. Replying to Mr. Moultrop, the total cost of the station is itemized as follows:

		Per Cent of Total Cost
Total cost per kw	$51	
Building		13.45
9000-kw. turbine		32.09
Eight 755 h.p. boilers, steam pipe and boiler room auxiliaries		35.50
15,000-sq. ft. condenser and auxiliaries		6.45
Circulating water system		8.27
Electrical equipment		3.04
Fuel oil storage		1.20
		100.00

Regarding the vacuum, I would say that our specifications called for 28-in. vacuum, and that the dry vacuum pump barely came within the guarantee. We are about to install a pump of larger capacity, inasmuch as all other conditions are adequate for a better vacuum.

The matter of more superheat is a more difficult proposition and on account of the doubtful total gain in efficiency, we are inclined to let that remain as it now is.

In reply to Mr. Emmet, I would say that when the turbine was erected some foreign material was accidentally left on two of the bucket wheels, damaging the buckets of the third and fifth stages, and that the manufacturers estimated before the test was run that new buckets in the place of the damaged ones would increase the water rate 1 lb. per kw. The results of the test, however, were given as they were made, without regard to what should have been the water rate of the turbine.