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CHARACTERISTICS OF MODERN HYDRAULIC TURBINES.*

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With Discussion by Messrs. Lewis F. Moody, John C. Parker, C. M. Allen, E. Kuichling, Daniel W. Mead, S. J. Zowski, and Chester W. Larner.

The design and selection of turbines for hydraulic power-plants were at one time—and that within comparatively recent years—processes of an extremely uncertain nature. They partook quite largely of experimentation, much of which, having no sound basis of scientific knowledge, degenerated into the purest guesswork. During late years, however, the various problems involved have been the subject of much careful and intelligent study, and the situation has become decidedly better, although there is still room for vast improvement.

One of the chief obstacles in the path of progress at the present time is the scarcity of published data on the performance of turbines of recent construction. A large quantity of exceedingly valuable information has been accumulated by the manufacturers, but very little of it is available to the public in published form, or, in fact, in any form whatever. The publication of tests seems to be an idea altogether abhorrent to the majority of turbine builders in America. It is with this policy of reticence that the writer is inclined to take issue, not only because it is of an unprofessional character, but because

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Mr. Barnes.

extreme temperatures with freezing conditions. The character of the concrete should be perfect as to density and integrity. This was possible under the system of casting used, as it afforded full opportunity for inspection and curing, and the writer believes such conditions were fulfilled in the work described.

The slab reinforcing bars were carried from girder to girder, with additional cantilever bars in cross direction laid over the track beams. In Plate XXIV, the dots representing the section of the slab bars were reproduced as dashes, hence the criticism justly applied by Mr. Hawkesworth.

No width of slab, in addition to that of the girders and beams, was considered in the calculations. The moment was taken to be that due to a distributed load of two-thirds of the locomotive weight (24 tons) over the two track beams, 7 by 20 in., spanning between the girders, 10 ft. from center to center.

The writer desires to acknowledge the suggestion of this type of pile protection, by Mr. F. A. Koetitz, in a paper before the Technical Society of the Pacific Coast, in April, 1906. He has recently learned that, since this date, this form of protection has been patented by Mr. Koetitz, who has kindly favored the writer with some photographs of work, embodying this protective feature, executed by the Pacific Construction Company, in San Francisco, of which Mr. Koetitz is Vice-President. The writer regrets that he did not have the benefit of that company's experience when the Bocas wharf was being built, for the production of the shells was the most difficult part of the work.

The writer also regrets that some discussion of the stability of concrete in sea-water has not been brought out by this paper. He feels indebted to the counsel of William B. Fuller, M. Am. Soc. C. E., that density be attained as the greatest protection against the action of sea-water, and is pleased to record that close observation of the shells at the date of writing (January, 1910) reveals no outward sign of deterioration of the concrete after the lapse of 27 months. Observations on other concrete structures in the tropics, which have shown some disintegration under the action of sea waves, emphasize the importance and efficacy of density in such locations.

It is greatly to be desired that additional information be obtained on this most important feature in the use of concrete in sea water.

it defeats the very commercial ends which, according to some peculiar method of mistaken reasoning, it is supposed to further. It is only by a broad-minded policy of publicity on the part of the manufacturers, that the Engineering Profession at large can become acquainted with the possibilities of turbine construction, and thereby profit in the selection of this apparatus by the progress which is being made in the development of the art.

The publication of data relative to results which have been accomplished does not necessitate the publication of any information concerning the details of design on which these results depend. Any manufacturer is justified in protecting for his own use any points of superiority in design which he may be able to devise. His success depends on them, and he could not reasonably be expected to publish them for the benefit of his competitors; but the results which can be accomplished by his machines in operation are quite a different matter. This information is of vital importance to all engineers interested in the development of water-power, and it is in pursuance of this conviction that the writer has prepared this paper.

In the evolution of turbine design in America, probably no factor has played a more important part than the Holyoke testing flume. Up to the present time, more than 1 800 turbines have been tested there, and these tests have undoubtedly had a very important influence on the advancement of the art. The fact that all wheels (within certain limits of capacity)* are tested there under essentially equivalent

* The Holyoke flume is best adapted to the testing of wheels from 27 to 42 in. in diameter, approximately. At the time the flume was built, wheels were of much lower capacity, and it was not designed to handle the large volume of water discharged by high-speed wheels 45 in. in diameter or greater. The writer names 42 in. as the limit, not because he believes that it is impossible to get good results with a wheel of larger size, but because no engineer of judgment would think of installing a larger wheel to operate under similar conditions, and dependence cannot be placed on such tests. If the conditions are less favorable than would be found in an ordinary plant, they are certainly unsuitable for a test where the object is to determine the best performance of which the wheel is capable. The main defect in design which affects the tests of large wheels is the shallowness of the tail-race. The floor of the flume, on which the draft-tube of the wheel stands for test, is only 5 ft. above the flat bottom of the tail-race. The draft-tube of a 49 or 50-in. wheel, unless it is very short, will be about 7 ft. in diameter at the lower end. The back pressure due to a column of water 7 ft. in diameter impinging on the flat bottom of the tail-race, 5 ft. from the end of the draft-tube, can be easily imagined. It has a throttling effect upon the discharge which cuts down the effective head, but does not show on the gauge, and hence the efficiency, as shown by the test, suffers. The loss does not seem to be altogether proportionate to the quantity discharged, but, apparently, is affected very largely by the relation of the draft-tube diameter to the depth of the tail-race in which the column has to turn to flow toward the weir. A 7-ft. column of water cannot readily be turned through an angle of 90° in a space of 5 ft. without considerable disturbance. The writer on one occasion tested a wheel which discharged 275 cu. ft. per sec. and found that the tail-water level, as determined by pressure readings from the draft-tube, was actually 1 ft. higher than the level in the well where the head gauge floats. The head, as shown by the gauge, was about 10 ft., whereas the real head on the wheel was only about 9 ft. Naturally, it is useless to expect good efficiency under such conditions. This is an extreme case, but wheels of smaller size, down to about 42 in. in diameter, undoubtedly suffer in the same way, although to a less extent.

conditions, adds greatly to the value of the results, because it renders them directly comparable. Tests made in place are subject to the effects of so many local conditions that one scarcely knows when it is fair to make direct comparisons, but Holyoke tests are not subject to this uncertainty, because the conditions are known and constant.

Further than this, the writer believes that it is not going too far to state that the Holyoke tests are more accurate than a majority of those made elsewhere, and are as accurate as the best of them. There is no other device for measuring power which is as generally reliable as the friction brake, when accurately adjusted. At the Holyoke flume the power of all wheels is determined by friction brakes, of which there are several sizes. The water is measured, according to the Francis formula, by a sharp-crested weir of adjustable width. The width of the weir is adjusted according to the discharge of the wheel, in order to bring the head on the weir within the limits of the experiments on which the Francis formula is based. This is in accordance with the best practice elsewhere. Furthermore, the arrangements for taking simultaneous readings, and the system of testing in general, are much superior to the conditions which often obtain in the case of tests in place.

These facts, together with the unquestionable impartiality of the tests, have given them a position of generally accepted reliability. Certain imputations to the contrary have occasionally emanated from Europe, but these are best answered by the fact that, in not a few cases, European engineers who have transplanted their activities to America have failed conspicuously to attain in the Holyoke flume the good results with which they were accredited at home.

Holyoke Tests versus Tests in Operation.—It should not be thought, from the foregoing remarks, that the writer regards a Holyoke test as necessarily an exact determination of the duty which a turbine will perform after installation. It is scarcely ever that, although, by the exercise of proper judgment, it may be a reasonably reliable indication of what may be expected in regular operation.

Tests are of interest to the Engineering Profession at large for two reasons: first, in order to determine what results have been accomplished by a machine which has been built and installed; second, as evidence of what a builder has accomplished in the past, on which to base speculations as to what he can accomplish in the case of a proposed installation. In the first case, a test in place is generally regarded as the most satisfactory, but, in the second case, the writer finds a great diversity of opinion regarding the relative merits of Holyoke tests and tests of units in regular operation. It is conceded by all, of course, that an accurate test in operation is the only exact method of determining the performance of the turbine under working conditions, and it is further conceded by most engineers that the tests which they themselves make are accurate. However, they are not so credulous in regard to the tests made by others. The apparatus used for tests at the site is usually improvised for the occasion, is handled, to a certain extent, by unpracticed hands, and the opportunities for error are many. For this reason some engineers place more confidence in Holyoke tests because they know that they are correct and that they furnish a reliable basis on which to estimate the results which may be expected after installation.

On the other hand, some assert that wheels are tested at Holyoke under ideal conditions which cannot be obtained in practice; that the test shows merely the efficiency of the runner, and a lot of other generalities, none of which has in general any sound foundation of fact. There are certain conditions under which a Holyoke test is more favorable to the wheel than a test in place, and there are other conditions under which it is decidedly the opposite. As a matter of fact, the results of any test, whether made at Holyoke or at the site, cannot be relied on to indicate the performance of a prospective installation under different conditions of head, or speed, or capacity, without very careful study of the influence which these changed conditions will have on the efficiency of the wheel. For example, to infer that, because a manufacturer has obtained certain efficiencies with a wheel developing 10 000 h.p. under 150 ft. head at 200 rev. per min., he can obtain the same results with a wheel developing 10 000 h.p. under 100 ft. head at 300 rev. per min. would be entirely unwarranted and erroneous. Such comparisons, however, are often made, sometimes to the disadvantage of the builder, and frequently to the delusion of the purchaser.

In order to elucidate this matter, the factors which affect a comparison between wheels operating under different conditions will be analyzed. For this purpose the losses of energy in the turbine may be divided as follows:

(a)—Hydraulic loss in the casing, from the penstock flange to the entrance to the guides,

(b)—Hydraulic loss in the guides and runner,

(c)—Hydraulic loss due to leakage around the runner,

(d)—Hydraulic loss in the draft-tube,

(e)—Mechanical loss due to the friction of the revolving parts.

In respect to loss (a), the advantage is usually in favor of the Holyoke test. It is ordinarily impracticable to test a runner at Holyoke in the casing in which it is to be permanently installed, and this fact must be taken into consideration. As installed in the flume, the wheel draws the water to the guide openings smoothly and with minimum disturbance, from a state of comparative rest. Naturally, the efficiency is higher under those conditions than it will be when the wheel is installed in a usually all too constricted casing in which the water approaches the guides with whirls and eddies due to obstructions and restricted areas. There is no form of casing for a turbine that can be made within the limits of reasonable proportions, except the spiral, which does not cause considerable loss of efficiency. The loss in a properly designed spiral casing is relatively small, because it accelerates the velocity gradually and delivers the water to the guides without abrupt change of direction. A test of a wheel in an open flume can be fairly applied to a turbine of the spiral-casing type with very little allowance for additional loss, but a considerable deduction should be made if the wheel is to be installed in a cylindrical casing.

Loss (a) is dependent largely on the velocity in the casing, and is related to the head only in that for a given velocity the percentage of loss decreases as the head increases.

Loss (b) is independent of the head, provided the speed is allowed to vary as $\sqrt{h}$, but is materially affected by the type of runner and the design of the guides. The results obtained with one type of runner cannot be assumed to apply to another type. This point is fully treated in the paper.

Loss (c) is likewise independent of the head, and is as great at Holyoke as in any other case. Both the leakage around the runner and the discharge through it vary as the $\sqrt{h}$, and hence the ratio of one to the other (or the percentage of loss) does not vary at all.

Loss (*d*), in the case of small wheels, is probably as low at Holyoke as elsewhere. With large wheels, however, this loss is excessive.*

Loss (*e*) varies with the head, decreasing as the latter increases, because the power of the wheel increases very much faster than the friction loss due to the weight of the revolving parts. There seems to be a general impression abroad that in testing wheels at Holyoke the mechanical friction is so far minimized as to be practically eliminated. It is true that effort is made to reduce the friction load, but it is not eliminated, by any means, and, although the percentage of loss is less than it would be for the same wheel running in horizontal bearings under the same head, it is more, and sometimes a great deal more, than it would be for the same wheel operating in a horizontal position under a higher head. In fact, it is necessary to minimize the friction in order to show results which will be representative of what the wheel can accomplish under heads of from 30 to 50 or 60 ft. For wheels to operate under higher heads, a Holyoke test is, almost without exception, a disadvantage to the wheel. This fact can be readily demonstrated. Consider, for example, one of the wheels referred to later, Test No. 1799. This wheel developed 112 h.p. under 17.25 ft. head. Suppose it is desired to install a pair of these wheels on a horizontal shaft under 80 ft. head. Will the efficiency be more or less than it was in the testing flume? Under this heading, of course, the influence of mechanical friction only will be considered.

This wheel when tested had four vertical guide-bearings, one ball thrust-bearing, and a stuffing-box, in contact with the shaft. The weight on the ball-bearing, including the unbalanced thrust of the runner, was about 7 000 lb. The pressure on the upper guide-bearing under the brake was 1 000 lb., due to the pull of the load. Under 80 ft. head, two of these wheels would develop

$$\frac{112 \times 2 \times \sqrt{80^3}}{\sqrt{17.25^3}} = 2\,240 \text{ h.p.},$$

which is twenty times as much power as was developed under test. The weight of the revolving parts would be about 5 000 lb.

Now, making the impossibly liberal assumption that under test the friction of the three lower guide-bearings, the stuffing-box, and the ball-bearing was absolutely nothing, there still remains the upper guide-

* See foot-note on p. 307.

bearing, which is known, beyond question, to carry a load of 1 000 lb. Since the bearings of the proposed unit will carry a weight of 5 000 lb., the friction will be about five times as great (neglecting the influence of speed, which would modify this statement slightly).

The power developed, however, is twenty times as great, and hence the percentage of energy lost is only one-quarter as much as when the wheel was tested at Holyoke. If the friction under test was 2%, then the friction in operation will be only 0.5%, or a difference of 1.5% in favor of the latter. Under yet higher heads, the wheel in operation has a continually increasing advantage from this source.

In spite of these facts, the writer has known engineers of experience to make the broad statement that friction is so completely eliminated from Holyoke tests that they may be considered to represent purely ideal results which cannot be obtained in practice, and this in reference to heads of even 200 or 300 ft. Such statements will not stand under analysis. Under high heads, exactly the reverse is true. In fact, the writer knows of no case so ideal or so favorable to high efficiency as that of a large reaction unit operating under a high head. In such a case, not only is the mechanical friction reduced to a minimum effect, but the casing usually takes the spiral form, which minimizes the casing loss. The wheel usually has individual draft-tubes, which minimize the draft-tube loss, and the design of the runner approaches the purely inward-flow Francis type, the characteristic of which is high efficiency. The design of the wheel throughout is the most efficient which can be devised.

Scope of Tests.—Tests of turbines after installation are usually made for the purpose of determining whether a specific set of conditions has been complied with satisfactorily. These tests are usually limited to observations at various gate openings under the head and speed for which the turbine was designed. Where the power is measured electrically, the difficulty of determining the efficiency of the generator at varying speeds precludes the possibility of a comprehensive test. Such tests may fulfil their function, which is to determine the performance of a guaranty, but they add very little to the knowledge of turbine characteristics in general.

It is customary practice to design a turbine to perform a certain duty under a certain head, and either to test it under that head, or else under whatever head is available, and, from the data thus obtained, compute the performance of the wheel under the head for which it was designed. Excepting high heads, there are few installations, however, where the conditions are by any means constant. A test made at constant speed under one head is of little value in determining the performance of the wheel at the same speed under another head. It is of even less value in determining the general characteristics of the wheel, and, as will be demonstrated later, these characteristics are of vital importance in considering its adaptability to a given service. The characteristic curves of a wheel cannot be plotted from a test at constant speed. It is essential that the speed should vary through a considerable range. In this respect, the Holyoke tests are excellent, and, but for the fact that they are so seldom published, they would furnish a vast amount of useful information.

Selection of Wheels.—The writer is of the opinion that more trouble has resulted from the misapplication of turbines to conditions for which they are not suited, than from any other of the many mistakes which have been made in the equipment of water-power plants. There are numerous instances all over the country where a wheel which would give good results if run at the proper speed is being run at some other speed and giving correspondingly poor results. The cause of these mistakes is not far to seek. It lies chiefly in the failure to analyze the tests of these wheels and thus determine whether they are properly adapted to the existing conditions.

Comparatively few engineers realize to what an extent the performances of different types of turbines differ. The divergence is wide, and the speed selected determines largely the type of runner which must be used. In spite of this, the speed is usually determined from a consideration of generator economy alone, without regard to its effect on the economy of the turbine. Such a course is short-sighted and ill-considered, to say the least. It results in high-speed wheels being adopted where lower-speed wheels are more suitable for many reasons. Frequently, much care and ingenuity are devoted to shackling the manufacturer with rigid guaranties for apparatus which should never be used at all for the purpose intended. Premiums are even paid to secure results which, economically considered, would be expensive if obtained gratuitously.

There is a more and more insistent demand for high-speed wheels, without a proper consideration of the limited conditions to which they are adapted and the undesirable attributes which all of them possess. The high-speed wheel, as will be explained later, has its legitimate field, but in many plants where it is being used to-day it is by no means the most economical type.

Nor has the manufacturer failed of his share of responsibility in the misuse of his wheels. Because of the long-established practice in America of standardizing runners and building lines of stock sizes, there has been too much effort directed toward adapting the plant to the wheels, rather than the wheels to the plant. This unfortunate practice has been due largely to the "cut and try" methods of the McCormick school of turbine designers, which, until recent years, dominated the turbine business in America. Each builder, having by tedious experimentation evolved his standard patterns, was accustomed to select from his stock the wheel which, according to his judgment, would most closely approximate the desired results. Frequently, the approximation was a very remote one, not more, probably, because of the rigidity of his standards than because of his meager knowledge of their characteristics. It is only within the last few years that manufacturers have learned to analyze the tests of their wheels intelligently, and to predict with accuracy what they will do under a variety of conditions. It is still more recently that designing of special wheels to suit individual requirements has come into general practice.

The writer believes that the special wheel is to be the rule in the future, rather than the exception. This should not be taken to mean that every new wheel will necessarily be an entirely new and original design. Such a procedure would be of too experimental a nature, particularly in the case of mixed-flow wheels, which are not as completely amenable to mathematical analysis as are their more simple prototypes, the Francis wheels. New designs will consist rather of modifications of and improvements on wheels which have been tested and found to be efficient, and, as such, will be more certain of success than if based wholly on theoretical considerations. The important point to be observed is that the wheel should be exactly suited to the conditions, in order that the best results may be obtained. There should be no approximation.

Description of Tests.—The purpose of this paper is not to present a study of the various problems which arise in the selection of water-wheels, but to illustrate, by means of tests, the individual charac-

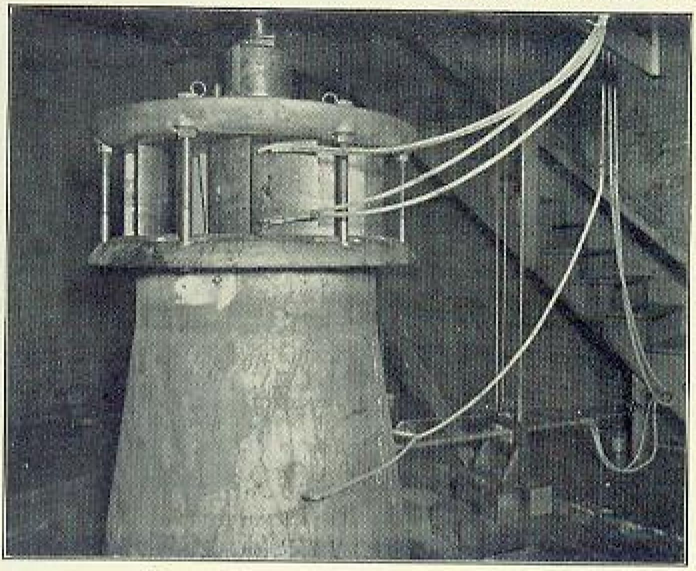


FIG. 1.—WHEEL IN FLUME READY FOR A TEST.

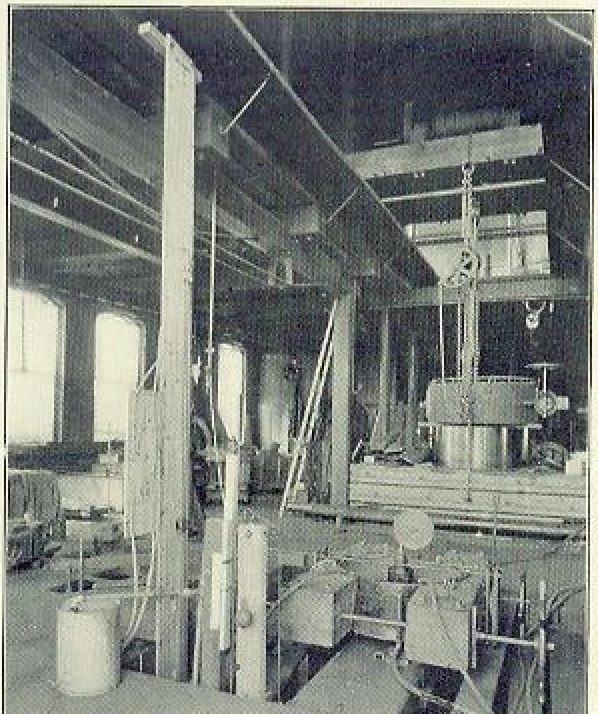


FIG. 2.—OPERATING FLOOR, SHOWING BRAKE, GAUGE-BOARD, ETC.

teristics of several different types, and to indicate in a general way the conditions of service to which each is best adapted. The conclusions drawn from this study are based, not merely on a consideration of these tests alone, but also of all the other American and European tests which have been accessible.

The tests which have been used for illustration were made recently at the Holyoke testing flume on a series of five specially designed runners. They are reproduced from the original reports in Tables 1 to 5, inclusive. These runners range from the low-capacity Francis type wheel, No. 1795, adapted particularly to high heads, to the large-capacity high-speed wheels, Nos. 1796 and 1797, adapted particularly to low heads. The average maximum efficiency of the five wheels is 88.06 per cent. This is believed to represent the best results thus far attained in the practice of the art. Tests Nos. 1799 and 1800 are especially notable in that they show higher efficiencies than any other reliable published records in either American or European practice. Test No. 1799, showing an efficiency of 90.43% at three-quarters-gate, is probably very close to the maximum obtainable economy.

The five wheels were successively tested on a vertical shaft in the same casing. The gates were of the balanced-wicket type. All the runners were of cast iron, made in one piece from cores.

Fig. 1 is a sectional view of the arrangement for testing one of the larger wheels. The vertical thrust was carried on a ball-bearing placed on top of the crown-plate. This arrangement was necessitated by the variation in height of the runners. It was not adopted to minimize the mechanical friction, and it is not believed that the friction was any less in the case of these tests than is usual with wheels of the same size. The usual arrangement is to carry the thrust on lignum vitæ step-bearings, but the invariable practice of all manufacturers is to turn off the face of the block until a small button about 2 in. in diameter is left for bearing. While running on this button the friction is practically nothing, although it takes more of a pull to start than is required with a ball-bearing.

Fig. 2 shows the arrangement for wheel No. 1795, with an interior wooden draft-tube. This inner tube was not used with any of the other wheels.

Fig. 1, Plate XXXI, is a photograph of the complete apparatus erected in the flume ready for a test. This view shows an arrangement

TABLE 1.—TESTS OF A 32-INCH R. H. WELLMAN-SEAVER-MORGAN COMPANY TURBINE WHEEL, No. 1795.

Date, February 18th and 19th, 1909. Case No. 1794.

Wheel supported by ball-bearing step. Swing-gate. Conical draft-tube.

Number of experiment.	Proportional Part of: Percentage of full opening of speed-gate.	Proportional Part of: Percentage of full discharge of wheel.	Head acting on wheel, in feet.	Duration of experiment, in minutes.	Revolutions of wheel per minute.	Quantity of water discharged by wheel, in cubic feet per second.	Horse power developed by wheel.	Percentage of efficiency of wheel.
69	1.000	1.024	17.23	4	112.50	34.45	46.81	69.59
67	1.000	1.009	17.50	3	152.67	34.22	55.80	82.16
68	1.000	1.008	17.57	2	168.40	34.25	57.31	83.98
66	1.000	1.005	17.46	3	164.00	34.05	56.90	84.40
65	1.000	0.992	17.47	3	172.67	33.60	55.92	84.00
64	1.000	0.979	17.48	3	181.33	33.17	54.53	82.93
63	1.000	0.965	17.46	4	189.25	32.67	52.53	81.21
62	1.000	0.946	17.47	4	218.00	32.06	49.27	77.57
61	1.000	0.686	17.59	4	251.50	23.39	34.91	74.81
60	1.000	0.631	17.86	4	298.00	21.62	......	......
59	0.969	0.983	17.50	3	113.67	31.63	45.23	72.01
58	0.880	0.984	17.52	3	135.67	31.70	50.84	80.72
57	0.880	0.981	17.54	3	140.33	31.60	52.80	84.01
56	0.880	0.925	17.58	3	153.67	31.40	53.32	85.41
55	0.880	0.922	17.44	4	156.50	31.21	52.85	85.62
54	0.880	0.916	17.45	3	160.33	31.00	52.66	85.84
53	0.880	0.909	17.46	3	164.00	30.79	52.35	85.87
52	0.880	0.903	17.47	4	168.00	30.59	52.07	85.92
51	0.880	0.898	17.48	4	172.75	30.42	51.95	86.14
50	0.880	0.893	17.48	4	176.75	30.25	51.52	85.91
49	0.880	0.886	17.49	4	184.00	30.05	51.08	85.69
48	0.880	0.864	17.53	4	206.50	29.31	47.77	81.98
47	0.880	0.795	17.59	4	244.75	27.02	38.97	63.02
46	0.880	0.572	17.91	4	269.75	19.61	......	......
45	0.741	0.809	17.57	4	106.00	27.48	39.23	71.65
44	0.741	0.801	17.57	4	130.50	27.20	45.18	83.35
43	0.741	0.791	17.55	3	152.00	26.85	45.71	85.53
41	0.741	0.784	17.57	4	159.75	26.63	45.82	86.35
40	0.741	0.780	17.60	4	166.25	26.52	46.15	87.18
42	0.741	0.777	17.56	4	171.50	26.40	46.02	87.53
39	0.741	0.770	17.64	4	180.75	26.22	45.90	87.08
38	0.741	0.758	17.67	4	191.50	25.82	44.30	85.61
37	0.741	0.727	17.73	3	210.67	24.81	38.96	78.16
36	0.741	0.693	17.79	3	231.33	23.68	32.11	67.20
35	0.741	0.509	18.09	2	269.40	17.31	......	......
34	0.593	0.669	17.84	3	101.00	22.80	32.71	70.63
33	0.593	0.660	17.84	3	122.67	22.61	35.32	79.40
31	0.593	0.644	17.86	3	145.00	22.07	37.57	84.01
30	0.593	0.642	17.85	3	151.33	21.98	37.81	84.97
32	0.593	0.637	17.88	4	158.00	21.83	38.01	85.87
29	0.593	0.632	17.91	4	168.75	21.68	37.88	86.02
28	0.593	0.622	17.91	3	173.00	21.33	36.82	84.98
27	0.593	0.606	17.99	3	188.00	20.82	34.79	81.90
26	0.593	0.537	18.16	4	230.00	18.54	22.11	57.92
25	0.593	0.406	18.65	3	281.00	13.98	......	......
24	0.444	0.505	17.91	3	91.00	17.32	23.92	67.90
22	0.444	0.489	17.92	2	130.00	16.78	27.06	79.36
21	0.444	0.486	17.94	3	136.00	16.70	27.68	81.48
23	0.444	0.480	17.92	4	130.25	16.66	27.70	81.82
20	0.444	0.481	17.94	3	145.33	16.50	27.57	82.12
19	0.444	0.472	17.94	4	153.00	16.21	25.90	81.56
18	0.444	0.465	17.95	5	161.40	15.98	26.13	80.34
17	0.444	0.450	17.99	4	178.00	15.46	24.71	78.32
16	0.444	0.402	18.04	4	231.50	13.85	16.07	56.70

TABLE 1.—(*Continued.*)

Number of experiment.	Proportional Part of:		Head acting on wheel, in feet.	Duration of experiment, in minutes.	Revolutions of wheel per minute.	Quantity of water discharged by wheel, in cubic feet per second.	Horse-power developed by wheel.	Percentage of efficiency of wheel.
	Percentage of full opening of speed-gate.	Percentage of full discharge of wheel.						
15	0.444	0.325	18.17	4	281.25	11.22	......	......
14	0.296	0.325	18.19	3	89.67	11.22	15.35	66.32
13	0.296	0.320	18.21	3	107.00	11.06	16.58	72.60
11	0.296	0.317	18.23	3	114.67	10.97	16.98	74.85
12	0.296	0.316	18.22	3	118.33	10.94	16.97	75.07
10	0.296	0.313	18.24	3	122.67	10.82	17.08	76.07
9	0.296	0.308	18.25	4	130.75	10.68	16.94	76.62
8	0.296	0.304	18.26	4	137.75	10.54	16.57	75.91
7	0.296	0.300	18.30	4	146.50	10.41	16.27	75.29
6	0.296	0.295	18.31	4	154.75	10.23	15.75	74.15
5	0.296	0.291	18.32	4	164.00	10.10	15.17	72.31
4	0.296	0.283	18.33	5	181.20	9.82	14.25	69.81
3	0.296	0.276	18.36	4	191.25	9.58	13.27	66.53
2	0.296	0.266	18.33	4	208.00	9.23	11.55	60.18
1	0.296	0.225	18.37	4	261.00	7.83	......	......

NOTE.—For Experiments Nos. 1, 15, 25, 35, 46 and 60, the jacket was loose.

During the above experiments, the weight of the dynamometer, and of that portion of the shaft which was above the lowest coupling was 1 300 lb.

With the flume empty, a strain of 1.0 lb., applied at a distance of 2,4 ft. from the center of the shaft, sufficed to start the wheel.

of Pitot tubes for measuring the velocity at the top and bottom of the guides and in the draft-tube. The latter consisted of a movable tube, with velocity and pressure points, introduced through a gland in the side of the draft-tube about 2 ft. below the bottom of the runner. This tube was arranged to move horizontally across the diameter of the draft-tube and also to revolve about its own axis. By means of these two movements, the velocity at any distance from the center of the shaft could be determined, and also the angle of whirl. The latter would necessarily be the angle of the velocity point which gave the highest reading. Both movements of the tube were controlled from the floor above, and, by means of indicators, the exact position of the point was at all times known.

Fig. 2, Plate XXXI, is a photograph of the operating floor, showing the brake for measuring the power of the wheel. At the left is the gauge-board and auxiliary apparatus connected to the Pitot tubes below. The water columns were drawn up into view by means of a vacuum. The arrangement of this apparatus is shown diagrammatically in Fig. 3. The position of the columns in the glass tubes was controlled

TABLE 2.—TESTS OF A 28-INCH R. H. WELLMAN-SEAVER-MORGAN COMPANY TURBINE WHEEL, NO. 1796.

Date, February 25th, 1909.

Wheel supported by ball-bearing step. Swing-gate. Conical draft-tube.

Number of experiment.	Proportional Part of: Percentage of full opening of speed-gate.	Proportional Part of: Percentage of full discharge of wheel.	Head acting on wheel, in feet.	Duration of experiment, in minutes.	Revolutions of wheel per minute.	Quantity of water discharged by wheel, in cubic feet per second.	Horse-power developed by wheel.	Percentage of efficiency of wheel.
95	1.077	0.971	17.11	4	158.00	97.00	125.20	66.52
94	1.077	1.017	16.97	3	190.67	101.16	147.66	75.84
93	1.077	1.036	16.94	3	224.33	102.98	156.38	79.04
92	1.077	1.053	16.89	3	239.33	104.50	159.58	79.72
91	1.077	1.061	16.87	3	247.33	105.22	161.17	80.06
89	1.077	1.068	16.81	3	253.67	105.70	161.45	80.12
90	1.077	1.072	16.80	3	259.00	106.08	161.71	80.01
88	1.077	1.079	16.82	4	267.50	106.82	162.15	79.58
87	1.077	1.026	17.05	3	294.67	102.27	125.03	63.23
86	1.000	0.913	17.28	3	147.00	91.65	120.20	66.98
85	1.000	0.957	17.16	2	190.50	95.70	144.34	77.50
84	1.000	0.972	17.14	3	211.67	97.10	152.09	80.89
83	1.000	0.981	17.13	3	225.00	98.03	156.85	82.36
82	1.000	0.990	17.11	3	232.67	98.90	157.96	82.31
80	1.000	0.996	17.09	4	240.25	99.43	160.19	83.13
81	1.000	1.003	17.07	3	247.33	100.07	161.17	83.19
79	1.000	1.004	17.07	4	252.25	100.14	160.55	82.82
78	1.000	1.001	17.13	4	259.00	100.00	157.00	80.82
77	1.000	0.983	17.22	3	268.33	98.43	146.39	76.15
76	1.000	0.911	17.47	4	293.50	91.96	106.75	58.59
106	0.923	0.868	17.32	4	143.25	87.24	115.49	67.39
105	0.923	0.899	17.24	5	176.00	90.12	135.49	76.99
104	0.923	0.920	17.15	4	201.00	91.96	146.21	81.74
101	0.923	0.931	16.98	5	213.20	92.59	148.62	83.66
100	0.923	0.936	16.98	6	220.67	92.96	150.48	84.31
99	0.923	0.942	16.91	4	227.25	93.51	151.52	84.50
102	0.923	0.945	16.93	4	232.00	93.82	151.88	84.31
103	0.923	0.945	17.04	4	235.50	94.14	152.74	83.96
98	0.923	0.945	16.98	4	237.75	93.82	151.32	84.00
97	0.923	0.924	17.02	4	254.50	92.04	138.84	78.15
96	0.923	0.823	17.25	4	288.25	82.47	87.36	54.15
42	0.923	0.870	17.17	3	146.33	87.02	116.20	68.57
41	0.923	0.895	17.10	4	170.00	89.37	130.87	75.51
40	0.923	0.921	17.04	4	202.75	91.80	146.25	82.44
39	0.923	0.928	16.99	3	208.67	92.35	147.99	83.17
35	0.924	0.932	17.03	4	216.25	92.81	150.75	84.10
38	0.923	0.937	16.97	4	220.00	93.20	151.36	84.38
36	0.924	0.939	17.01	4	223.75	93.44	152.58	84.05
34	0.923	0.940	17.02	4	226.25	93.66	150.86	83.45
37	0.923	0.944	16.97	3	238.33	93.90	151.69	83.94
33	0.923	0.921	17.13	4	256.25	92.05	139.80	78.18
32	0.923	0.823	17.27	3	288.00	82.54	87.29	53.99
31	0.923	0.730	17.50	4	334.75	73.75	......	......
74	0.846	0.824	17.46	3	158.67	83.15	120.23	73.02
75	0.846	0.836	17.46	3	175.67	84.35	129.91	77.78
72	0.846	0.861	17.34	5	202.90	86.50	143.40	84.30
70	0.846	0.865	17.33	4	209.00	86.95	145.09	85.25
71	0.846	0.868	17.33	3	215.00	87.24	147.27	85.80
73	0.846	0.870	17.34	4	219.25	87.47	148.19	86.15
69	0.846	0.869	17.32	4	221.25	87.32	147.53	86.01
68	0.846	0.866	17.33	4	227.75	87.02	144.96	84.76
67	0.846	0.858	17.36	4	231.75	86.25	140.48	82.78
66	0.846	0.845	17.39	4	243.75	85.11	132.98	79.22
65	0.846	0.828	17.44	4	256.50	83.44	124.39	75.37

TABLE 2.—(*Continued.*)

Number of experiment.	Proportional Part of:		Head acting on wheel, in feet.	Duration of experiment, in minutes.	Revolutions of wheel per minute.	Quantity of water discharged by wheel, in cubic feet per second.	Horse-power developed by wheel.	Percentage of efficiency of wheel.
	Percentage of full opening of speed-gate.	Percentage of full discharge of wheel.						
64	0.846	0.754	17.59	3	282.00	76.31	85.47	56.15
30	0.769	0.750	17.38	3	141.00	75.52	102.56	68.90
26	0.769	0.766	17.35	3	166.00	77.02	115.72	76.36
27	0.769	0.779	17.32	3	183.00	78.24	124.24	80.84
25	0.769	0.789	17.31	3	194.00	79.25	129.36	83.15
29	0.769	0.793	17.25	4	200.75	79.48	131.42	84.52
28	0.769	0.792	17.26	4	206.00	79.40	131.11	84.36
24	0.769	0.773	17.38	4	226.25	77.82	123.43	80.47
23	0.769	0.735	17.49	3	251.33	74.17	106.64	72.49
22	0.769	0.690	17.56	3	269.67	69.82	81.73	58.78
21	0.769	0.623	17.68	4	323.75	63.27	......	......
20	0.615	0.617	17.91	4	189.50	63.07	88.79	69.31
16	0.615	0.627	17.79	3	158.33	63.81	95.97	74.55
17	0.615	0.634	17.80	3	171.33	64.55	101.26	77.71
15	0.615	0.638	17.73	4	179.50	64.82	103.37	79.31
18	0.615	0.636	17.77	4	183.00	64.74	103.16	79.07
19	0.615	0.634	17.80	2	188.00	64.61	102.56	78.64
14	0.615	0.627	17.72	4	194.50	63.74	100.21	78.24
13	0.615	0.596	17.77	3	218.67	60.60	92.78	75.97
12	0.615	0.563	17.79	4	243.00	57.29	73.65	63.72
11	0.615	0.519	17.92	4	312.00	53.00	......	......
10	0.462	0.452	17.34	3	117.33	45.40	56.90	63.73
5	0.462	0.453	17.02	3	136.0C	45.05	61.88	70.17
7	0.462	0.461	17.08	4	146.75	46.04	64.94	72.82
6	0.462	0.462	17.05	4	152.00	46.04	66.34	74.52
4	0.462	0.462	16.92	4	155.25	45.90	65.88	74.79
9	0.462	0.459	17.16	3	162.00	45.94	66.78	74.69
8	0.462	0.457	17.15	3	166.67	45.69	65.67	73.90
3	0.462	0.451	16.98	4	172.25	44.83	62.65	72.57
2	0.462	0.432	17.05	4	217.50	43.04	52.74	63.37
1	0.462	0.404	17.18	5	282.40	40.40	......	......

NOTE.—During the above experiments, the weight of the dynamometer, and of that portion of the shaft which was above the lowest coupling, was 2 600 lb.

With the flume empty, a strain of 0.5 lb., applied at a distance of 3.2 ft. from the center of the shaft, sufficed to start the wheel.

by three-way cocks. By connecting a set of tubes with the vacuum tank, the columns were drawn above the floor level, after which, by closing the cock, they were held in a stationary position. If drawn too high they could be dropped back by opening the cock to the atmosphere. After the columns were once adjusted and the cocks closed, they would stand indefinitely without attention.

This apparatus for the measurement of velocities has been described at some length for the benefit of those who may be interested in the same line of experiments. Lack of space precludes the publication of any of the data obtained, although they are of great interest to the designer, and throw much light on some very obscure points in the theory of turbine design.

TABLE 3.—TESTS OF A 30-INCH R. H. WELLMAN-SEAVER-MORGAN COMPANY TURBINE WHEEL, NO. 1797.

Date, February 26th and 27th, 1909.

Wheel supported by ball-bearing step. Swing-gate. Conical draft-tube.

Number of experiment.	Proportional Part of: Percentage of full opening of speed-gate.	Proportional Part of: Percentage of full discharge of wheel.	Head acting on wheel, in feet.	Duration of experiment, in minutes.	Revolutions of wheel per minute.	Quantity of water discharged by wheel, in cubic feet per second.	Horse-power developed by wheel.	Percentage of efficiency of wheel.
82	1.000	0.942	17.06	3	149.00	94.89	126.41	68.85
81	1.000	0.950	16.99	3	160.33	95.52	131.16	71.26
80	1.000	0.963	17.00	4	190.50	96.85	144.30	77.28
75	1.000	0.977	17.02	4	210.50	98.34	150.52	79.30
74*	1.000	0.984	16.95	4	220.00	98.81	153.31	80.72
76	1.000	0.992	17.01	4	227.25	99.75	155.61	80.87
78	1.000	0.994	16.94	4	229.50	99.75	155.76	81.28
77	1.000	0.997	16.98	4	231.25	100.14	155.55	80.66
79	1.000	1.002	16.90	3	239.67	100.45	156.85	81.47
73	1.000	1.003	16.94	4	254.75	100.70	154.37	79.80
72	1.000	0.871	17.13	3	287.00	87.93	86.96	50.91
71	1.000	0.728	17.44	3	317.67	74.17	......	.....
52	0.923	0.875	17.08	3	131.33	88.14	111.42	65.26
51	0.923	0.896	16.95	3	156.33	89.97	127.89	73.95
50	0.923	0.913	16.93	4	188.25	91.66	142.59	81.02
44	0.923	0.920	16.89	4	201.25	92.20	146.34	82.86
45	0.923	0.927	16.85	4	211.50	92.81	148.67	83.89
46	0.923	0.931	16.86	4	217.00	93.20	149.91	84.12
49	0.923	0.934	16.83	4	220.50	93.42	150.99	84.68
47	0.923	0.936	16.83	4	223.75	93.60	151.86	85.00
48	0.923	0.937	16.83	3	226.33	93.74	150.87	84.32
43	0.923	0.929	16.88	4	241.75	93.13	146.49	82.17
42	0.923	0.788	17.21	4	276.25	79.75	88.70	58.77
41	0.923	0.670	17.44	4	312.50	68.26	......	.....
70	0.846	0.868	17.07	4	192.75	87.47	142.50	84.15
67	0.846	0.875	17.10	4	205.00	88.22	147.83	86.41
68	0.846	0.876	17.06	4	209.50	88.22	148.53	87.02
66	0.846	0.876	17.11	4	212.00	88.37	149.02	86.91
69	0.846	0.877	17.05	3	213.00	88.30	148.43	86.94
65	0.846	0.876	17.13	5	213.80	88.37	147.70	86.03
64	0.846	0.875	17.13	5	215.80	88.30	146.46	85.38
63	0.808	0.810	17.33	4	143.50	82.24	114.78	71.01
62	0.808	0.834	17.25	3	177.00	84.44	135.14	81.81
61	0.808	0.843	17.24	5	194.80	85.35	142.83	85.59
60	0.808	0.847	17.21	3	201.00	85.73	144.94	86.02
59	0.808	0.848	17.19	3	206.00	85.73	146.05	87.39
58	0.808	0.847	17.19	4	208.25	85.65	145.12	86.91
57	0.808	0.845	17.19	3	210.00	85.41	148.80	86.36
56	0.808	0.840	17.22	3	216.67	85.06	141.80	85.37
55	0.808	0.825	17.25	4	224.75	83.53	136.19	83.34
54	0.808	0.807	17.29	4	236.50	81.80	128.98	80.41
53	0.808	0.695	17.51	4	264.25	70.97	80.06	56.81
40	0.769	0.755	17.29	4	118.50	76.63	98.85	62.21
39	0.769	0.776	17.21	4	146.50	78.47	110.97	72.46
35	0.769	0.800	17.03	4	174.75	80.49	127.07	81.74
36	0.769	0.806	17.03	4	190.75	81.18	135.24	86.31
37	0.769	0.805	17.10	4	197.25	81.21	137.46	87.28
38	0.769	0.805	17.15	4	200.50	81.28	137.29	86.84
34	0.769	0.803	16.83	4	199.50	80.34	132.96	86.72
33	0.769	0.788	16.93	4	211.75	79.04	128.33	84.55
32	0.769	0.745	16.93	4	232.50	74.72	112.71	78.56
31	0.769	0.660	17.36	4	257.50	67.02	78.02	59.13
30	0.769	0.579	17.60	3	306.00	59.27	......	.....
29	0.615	0.610	17.69	3	100.67	62.60	69.54	55.37

TABLE 3.—(*Continued.*)

Number of experiment.	Proportional Part of: Percentage of full opening of speed gate.	Proportional Part of: Percentage of full discharge of wheel.	Head acting on wheel, in feet.	Duration of experiment, in minutes.	Revolutions of wheel, per minute.	Quantity of water discharged by wheel, in cubic feet per second.	Horse-power developed by wheel.	Percentage of efficiency of wheel.
28	0.615	0.621	17.64	3	127.67	63.60	85.10	66.80
25	0.615	0.648	17.60	3	167.38	65.76	104.44	79.67
27	0.615	0.644	17.57	4	175.50	65.83	107.41	81.80
24	0.615	0.648	17.61	4	178.00	65.76	107.86	82.13
26	0.615	0.641	17.59	3	179.67	65.55	106.70	81.60
23	0.615	0.482	17.96	3	296.67	49.84	......	.....
20	0.615	0.634	17.88	4	155.00	65.37	101.44	76.53
21	0.615	0.634	17.80	5	151.40	65.22	98.17	74.56
22	0.615	0.639	17.69	3	160.33	65.50	102.08	78.37
19	0.615	0.644	18.08	5	174.20	66.80	111.89	81.69
18	0.615	0.643	18.11	3	180.00	66.73	112.35	81.97
17	0.615	0.638	18.07	4	183.50	66.19	111.20	81.98
16	0.615	0.636	18.03	4	185.00	65.90	109.86	81.53
15	0.615	0.630	17.96	3	190.00	65.15	109.38	82.43
14	0.615	0.606	17.81	4	201.00	62.06	103.53	81.80
13	0.615	0.574	17.79	4	216.75	59.07	91.94	77.15
12	0.615	0.563	17.54	5	245.20	56.46	74.20	66.15
11	0.615	0.483	17.71	5	294.60	49.58	......	.....
8	0.462	0.475	17.82	3	104.33	48.95	58.16	58.80
7	0.462	0.483	17.75	4	139.75	49.58	71.98	72.12
6	0.462	0.482	17.76	3	147.00	49.51	73.98	74.14
5	0.462	0.481	17.80	4	156.75	49.45	75.90	76.12
9	0.462	0.477	17.93	2	162.00	49.22	76.57	76.51
10	0.462	0.472	17.97	3	167.00	48.78	75.50	76.35
4	0.462	0.464	17.88	5	178.00	47.80	73.64	75.97
3	0.462	0.436	18.05	4	216.50	45.13	65.60	71.01
2	0.462	0.415	18.08	4	246.25	43.02	44.77	50.75
1	0.462	0.381	18.27	4	280.50	39.75	......	.....

NOTE.—The jacket was loose for Experiments Nos. 1, 11, 23, 30, 41, and 71.

During the above experiments, the weight of the dynamometer, and of that portion of the shaft which was above the lowest coupling was 2 600 lb.

With the flume empty, a strain of 0.5 lb., applied at a distance of 3.2 ft. from the center of the shaft, sufficed to start the wheel.

Figs. 4 to 19, inclusive, are performance curves plotted from the test reports, on the common basis of 1 ft. head. The head at the testing flume fluctuates, and it is therefore necessary to reduce all readings to a common basis. With constant efficiency, the brake-horse-power varies as $\sqrt{h^3}$, and the speed and discharge as $\sqrt{h}$. This principle is true, theoretically, and has been repeatedly proven, experimentally, and all the computations have been made upon this basis.

Three sets of curves have been plotted for each of the five wheels. The first in order for each wheel is a set of brake-horse-power and efficiency curves for all gate openings plotted to speed. Each curve is designated by the gate opening which it represents. In addition to power and speed, the writer usually plots the discharge curves, but

TABLE 4.—Tests of a 31-inch R. H. Wellman-Seaver-Morgan Company Turbine Wheel, No. 1799.

Date, March 2d and 3d, 1909.

Wheel supported by ball-bearing step. Swing-gate. Conical draft-tube.

Number of experiment.	Proportional Part of: Percentage of full opening of speed-gate.	Proportional Part of: Percentage of full discharge of wheel.	Head acting on wheel, in feet.	Duration of experiment, in minutes.	Revolutions of wheel per minute.	Quantity of water discharged by wheel, in cubic feet per second.	Horse power developed by wheel.	Percentage of efficiency of wheel.
49	1,000	1,049	17.15	3	134,67	79,75	115,81	74,06
48	1,000	1,038	17.15	3	147,00	78,95	119,29	77,08
47	1.000	1,025	17.19	3	165,67	78,02	123,40	81.18
46	1,000	1,024	17.18	2	174,00	77,91	124,34	81.91
45	1,000	1,018	17.19	3	178,33	77,52	124,19	82,18
44	1,000	1,013	17.19	3	183,33	77,16	124.35	82,66
43	1.000	0,012	17.19	4	186,25	77,09	124.07	82,56
42	1.000	1.009	17.18	4	189,50	76,82	123,94	82,81
41	1.000	1,007	17.19	3	193,00	76.67	123,89	82,89
40	1.000	1,006	17.12	4	195,50	76,44	123,13	82,96
39	1.000	1,002	17.09	4	200,25	76.10	122,48	83,04
38	1.000	0,999	17.10	4	206,25	75,87	122,41	83,19
37	1.000	0.997	17.07	3	210,33	75,66	121,01	82,62
36	1.000	0,993	17.09	4	219,00	75,38	119.36	81,70
35	1.000	0,906	17.24	4	258,75	69,11	78,85	57,98
34	1.000	0,744	17.45	4	302,25	57,07	......	.....
32	0,883	0,909	17.18	3	129,33	69,23	101,03	74,90
31	0,883	0,904	17.20	4	154,25	68,82	111.16	82,81
33	0,883	0,901	17.19	4	165,00	68,60	113.91	85,18
30	0,883	0,896	17.20	3	172,33	68,25	114,80	86,23
29	0,883	0,892	17.21	4	181,50	67,97	116,51	87,82
27	0,883	0,888	17.25	3	188,00	67,77	117.27	88,45
26	0,883	0,886	17.30	4	194,00	67,71	117,49	88,44
28	0,883	0,885	17.23	4	197,25	67,43	117,06	88,85
25	0,883	0,883	17,31	4	201,75	67,49	116,07	87,61
24	0,883	0,857	17,28	4	213,25	65,42	109,77	85,62
23	0,883	0,818	17.31	4	227,00	62,53	96,23	78,39
22	0,883	0,761	17.42	4	244,25	58,32	73,96	64,19
21	0,883	0,620	17.65	4	291,75	47,83	......	.....
69	0,750	0,835	17,28	4	124,75	63,78	91.41	78,19
68	0,750	0,839	17.23	4	148,75	63,93	103,59	82,93
67	0,750	0.834	17.23	4	168,75	63,60	109,35	87,99
64	0,750	0,830	17.24	4	179,25	63,26	110,72	89.52
65	0,750	0,829	17.23	4	182,00	63,20	110,77	89,69
63	0,750	0,828	17.25	4	185,25	63,12	111.66	90,43
66	0,750	0,824	17.23	3	188,67	62,85	110.83	90.21
62	0,750	0,821	17.29	4	191,75	62,72	110.32	89,70
61	0,750	0,809	17.30	4	197,50	61,81	107,64	88,76
60	0,750	0,796	17.31	4	203,25	60,86	104,62	87.42
59	0,750	0.768	17.38	4	212,50	58,80	96.52	83,28
58	0,750	0,693	17.55	4	237,00	53,31	71,76	67,64
57	0,750	0,578	17.75	4	287,00	44,32	......	.....
20	0,667	0,726	17.27	4	106,25	55,43	70,78	65,20
19	0,667	0,737	17.26	3	148,67	56,25	90,93	82,59
18	0,667	0,739	17,25	4	161,00	56,39	95,55	88,02
17	0,667	0,738	17.24	4	168,50	56,31	96.94	88,05
16	0,667	0,733	17.27	3	173,67	55,92	96,75	88,35
15	0,667	0,722	17.30	4	179,25	55,17	95,52	88,25
14	0,667	0,699	17.34	4	189,50	53,48	91.81	87,30
13	0,667	0,671	17.39	4	201,75	51,37	85,52	84,42
11	0,667	0,508	17.63	4	280,75	39,17	......	.....
8	0.500	0.554	17.82	3	117,00	42,95	60,23	69,39
7	0.500	0.546	17.91	3	135,00	42,46	65,40	75,84
10	0,500	0,548	17.67	3	151,00	42,28	68,58	80,96

TABLE 4.—(*Continued.*)

Number of experiment.	Proportional Part of:		Head acting on wheel, in feet.	Duration of experiments, in minutes.	Revolutions of wheel per minute.	Quantity of water discharged by wheel, in cubic feet per second.	Horse-power developed by wheel.	Percentage of efficiency of wheel.
	Percentage of full opening of speed-gate.	Percentage of full discharge of wheel.						
6	0.500	0.548	18.05	4	157.50	42.72	71.54	81.80
9	0.500	0.547	17.71	4	157.00	42.28	69.41	81.73
5	0.500	0.539	18.12	5	167.60	42.11	71.05	82.10
4	0.500	0.512	18.13	4	187.00	40.02	67.95	82.58
3	0.500	0.488	18.18	4	213.00	38.21	58.05	73.68
2	0.500	0.400	18.07	4	232.00	35.82	42.15	57.26
1	0.500	0.402	18.20	3	275.00	31.50	.....	.....
52	0.333	0.302	18.18	3	96.00	23.35	31.38	59.68
55	0.333	0.361	18.01	3	117.33	28.10	39.08	68.09
51	0.333	0.348	18.19	4	133.00	27.25	40.27	71.64
54	0.333	0.347	18.21	3	139.67	27.30	40.60	72.28
53	0.333	0.340	18.22	3	148.67	26.62	39.15	71.18
56	0.333	0.333	18.06	4	148.00	26.00	37.64	70.69
60	0.333	0.316	18.26	3	201.67	24.82	36.61	71.28

NOTE.—For Experiments Nos. 1, 11, 21, 34, and 57, the jacket was loose.

During the above experiments, the weight of the dynamometer and of that portion of the shaft which was above the lowest coupling was 2 000 lb.

With the flume empty, a strain of 1.0 lb., applied at a distance of 3.2 ft. from the center of the shaft, sufficed to start the wheel.

they have been omitted from this paper in order to economize space.* When included, such a diagram might be termed in common parlance a "personality chart," because it embodies all the characteristics and peculiarities of the wheel. From it separate performance curves can be plotted for any speed whatsoever within the range of the test. The heavy speed line drawn through each of these charts indicates the "normal speed" of the wheel, that is, the speed at which it gives its maximum efficiency.

The second set of curves shows the performance of the wheels along these heavy lines, or at normal speed. The gate openings are also plotted to discharge, so that it is possible to read the efficiency at any fractional part of the full gate opening as well as at any fractional part of the full discharge of the wheel.

The first set of curves shows, in addition to the information already referred to, a scale of values of ϕ, in the equation,

$$W_1 = \phi \sqrt{2\,gh},$$

where W_1 is the peripheral speed of the wheel, based on its rated

* For full details of this method of plotting test curves, see Chapter XVI of "Water Power Engineering," by Daniel W. Mead, M. Am. Soc. C. E.

TABLE 5.—TESTS OF A 31-INCH R. H. WELLMAN-SEAVER-MORGAN COMPANY TURBINE WHEEL, No. 1800.

Date, March 4th and 5th, 1909.

Wheel supported on ball-bearing step. Swing-gate. Conical draft-tube.

Number of experiment.	Proportional Part of: Percentage of full opening of speed-gate.	Proportional Part of: Percentage of full discharge of wheel.	Head acting on wheel, in feet.	Duration of experiment, in minutes.	Revolutions of wheel per minute.	Quantity of water discharged by wheel, in cubic feet per second.	Horse-power developed by wheel.	Percentage of efficiency of wheel.
65	1.000	1.052	17.38	3	112.33	64.87	82.90	64.91
64	1.000	1.040	17.41	3	133.33	64.20	91.24	71.96
63	1.000	1.025	17.43	4	154.25	63.33	96.21	76.85
62	1.000	1.014	17.42	3	169.67	62.60	98.64	79.76
61	1.000	1.009	17.40	3	183.33	62.25	99.92	81.34
60	1.000	1.006	17.42	4	193.25	62.13	100.65	82.00
59	1.000	1.004	17.43	4	201.00	62.01	101.08	82.42
58	1.000	1.002	17.44	4	207.00	61.88	101.54	82.96
56	1.000	1.000	17.35	4	211.00	61.65	100.95	83.22
57	1.000	0.999	17.39	4	216.75	61.65	101.07	83.13
55	1.000	0.998	17.36	4	221.50	61.53	100.90	83.05
54	1.000	0.995	17.38	4	247.50	61.40	97.42	80.50
53	1.000	0.951	17.45	4	266.75	58.80	80.77	69.41
52	1.000	0.730	17.71	4	321.25	45.44	.....	.....
51	0.883	0.889	17.48	4	132.25	55.02	79.29	72.09
50	0.883	0.888	17.48	4	147.00	54.91	83.68	76.88
49	0.883	0.885	17.47	4	165.00	54.71	88.93	82.04
48	0.883	0.879	17.39	4	179.25	54.24	91.18	85.24
47	0.883	0.876	17.42	5	189.20	54.11	91.66	85.75
44	0.883	0.873	17.54	4	197.50	54.11	93.29	86.67
43	0.883	0.873	17.40	4	204.25	53.85	92.77	87.30
45	0.883	0.872	17.40	3	209.00	53.79	92.40	87.05
46	0.883	0.871	17.42	4	215.50	53.79	92.66	87.19
42	0.883	0.858	17.40	3	225.00	52.92	88.57	84.81
41	0.883	0.810	17.45	3	248.00	50.06	75.09	75.80
40	0.883	0.609	17.68	3	314.00	37.87	.....	.....
75	0.733	0.801	17.76	4	158.50	49.93	81.59	81.13
73	0.733	0.798	17.73	3	183.33	49.70	86.60	86.66
72	0.733	0.797	17.71	3	190.33	49.63	87.00	87.88
71	0.733	0.796	17.72	3	197.00	49.57	88.28	88.62
74	0.733	0.795	17.73	3	199.00	49.50	87.97	88.39
70	0.733	0.793	17.72	3	201.67	49.37	87.93	88.63
69	0.733	0.788	17.70	4	204.50	49.02	86.69	88.10
68	0.733	0.775	17.71	4	213.50	48.27	84.04	86.69
67	0.733	0.722	17.79	3	234.00	45.04	70.85	77.97
66	0.733	0.544	18.03	3	308.67	34.15	.....	.....
38	0.667	0.751	17.58	3	171.00	46.58	78.70	84.75
37	0.667	0.750	17.57	4	184.00	46.52	81.34	87.75
39	0.667	0.749	17.56	4	187.00	46.45	81.54	88.14
35	0.667	0.749	17.59	4	185.50	46.45	80.88	87.20
36	0.667	0.747	17.59	4	188.75	46.38	81.73	88.53
34	0.667	0.742	17.58	4	192.50	46.02	80.44	87.67
33	0.667	0.731	17.59	4	198.25	45.37	78.04	86.22
32	0.667	0.758	17.70	3	127.67	46.87	68.04	72.32
31	0.667	0.752	17.68	3	144.67	46.76	72.73	77.56
29	0.667	0.752	17.66	4	164.25	46.76	77.59	82.85
30	0.667	0.753	17.68	3	178.67	46.82	81.15	86.44
28	0.667	0.750	17.62	3	189.00	46.57	82.41	88.56
27	0.667	0.741	17.63	3	193.67	46.02	80.93	88.00
26	0.667	0.731	17.64	3	201.00	45.44	79.12	87.04
25	0.667	0.725	17.63	3	204.00	45.05	77.83	86.41
24	0.667	0.709	17.64	3	213.33	44.05	74.93	85.03
23	0.667	0.678	17.70	4	234.50	42.23	67.98	80.19
22	0.667	0.506	17.85	3	308.33	31.61	.....	.....

TABLE 5.—(*Continued.*)

Number of experiment.	Proportional Part of: Percentage of full opening of speed-gate.	Proportional Part of: Percentage of full discharge of wheel.	Head acting on wheel, in feet.	Duration of experiment, in minutes.	Revolutions of wheel per minute.	Quantity of water discharged by wheel, in cubic feet per second.	Horse-power developed by wheel.	Percentage of efficiency of wheel.
21	0.500	0.566	17.72	2	124.00	35.27	50.31	70.98
20	0.500	0.560	17.71	3	130.33	34.87	52.84	75.32
19	0.500	0.551	17.71	4	140.50	34.50	53.23	76.82
18	0.500	0.543	17.72	4	155.75	33.82	52.82	77.72
17	0.500	0.535	17.74	3	160.00	33.31	52.27	78.00
16	0.500	0.521	17.77	3	170.67	32.47	51.30	78.48
15	0.500	0.507	17.79	3	180.50	31.61	50.30	78.97
14	0.500	0.494	17.83	4	200.25	30.87	48.51	77.71
13	0.500	0.481	17.86	3	214.00	30.08	45.30	74.45
12	0.500	0.464	17.80	4	228.75	29.02	41.56	70.58
11	0.500	0.380	18.00	4	290.50	23.87	.....	.....
8	0.333	0.360	18.09	4	112.00	22.65	30.52	65.98
7	0.333	0.351	18.12	3	127.00	22.10	31.53	69.43
6	0.333	0.350	18.13	3	136.00	22.02	32.12	70.95
5	0.333	0.346	18.22	4	141.50	21.84	31.71	70.26
4	0.333	0.340	18.24	4	148.00	21.47	31.37	70.03
3	0.333	0.332	18.31	4	154.75	21.03	30.93	70.82
2	0.333	0.323	18.25	4	165.25	20.42	30.02	71.04
10	0.333	0.307	18.17	4	212.50	19.33	28.31	71.08
9	0.333	0.300	18.15	3	231.33	18.93	25.22	64.72
1	0.333	0.248	18.35	4	287.25	15.74	.....	.....

NOTE.—The jacket was loose for Experiments Nos. 1, 11, 22, 40, 52, and 66.

During the above experiments, the weight of the dynamometer and of that portion of the shaft which was above the lowest coupling was 2 600 lb.

With the flume empty, a strain of 0.5 lb., applied at a distance of 3.2 ft. from the center of the shaft, sufficed to start the wheel.

diameter. The third set of curves shows the performance of the wheels at speeds corresponding to several different values of ϕ. When a wheel operates at constant speed under a fluctuating head, the value of ϕ is continually changing. A comparison of these curves, therefore, is a very instructive study, because it shows how the efficiency varies as the head changes. The change in efficiency corresponding to a given change in ϕ is very different for different types of runner.

High-Speed Wheels.—In the use of the term "high-speed wheel," there seems to be much confusion. In the sense in which it is used here, the term does not apply necessarily to wheels of high peripheral speed, although a high value of ϕ is usually an attribute of a "high-speed wheel." As the term is used here, the wheel which, power for power, under a constant head, runs at the highest number of revolutions, is the highest-speed wheel. A high-speed wheel is one which will develop any given quantity of power at a high speed, relative to other wheels developing the same power under the same head. It is not

FIG. 1.

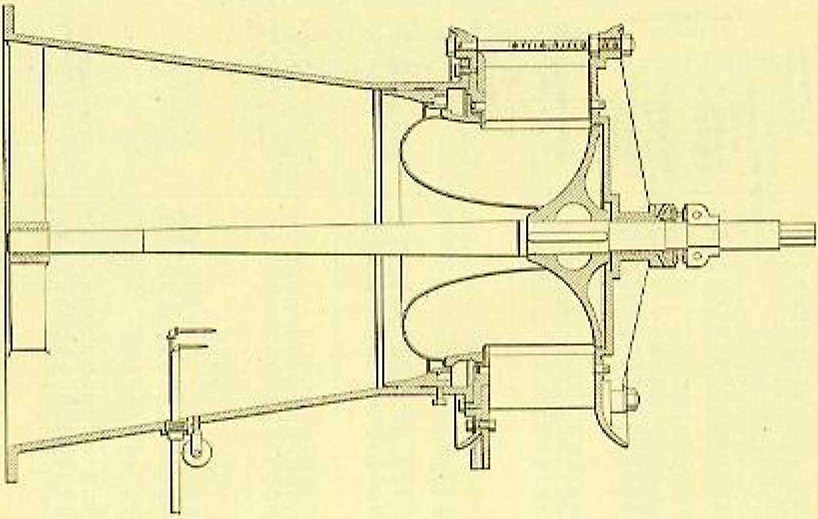
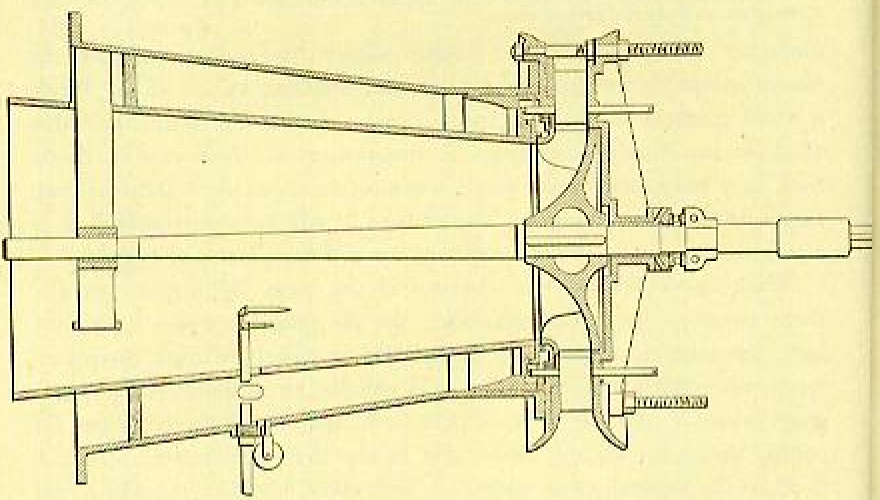


FIG. 2.



possible to compare the speeds of wheels upon the basis of diameter, because the power is not a consistent function of the diameter, except in wheels of homologous design. To determine which of two wheels is the higher-speed, it is usually necessary, unless they differ widely, to reduce them to a common basis of power and head.

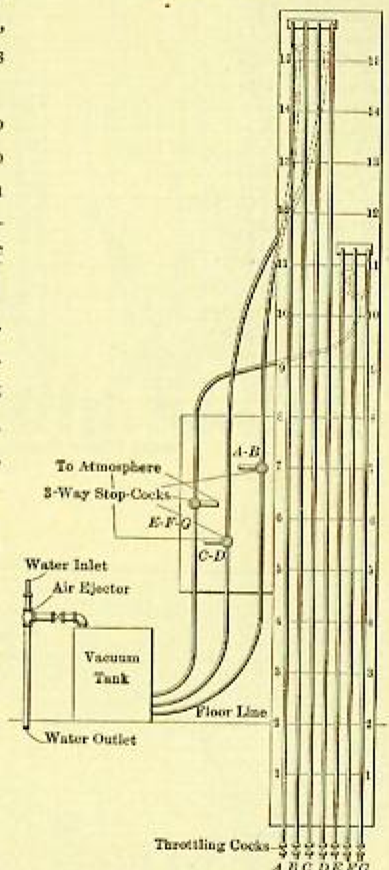
Suppose, for example, that two types of runner, A and B, are to be compared, and it is known from tests, or otherwise, that some certain size of A (the diameter of which need not be given) will develop a maximum of 2 080 h.p. at 500 rev. per min. under 100 ft. head, whereas an entirely different size of B will develop a maximum of 4 590 h.p. at 580 rev. per min. under 150 ft. head. It is impossible to determine from an inspection of these data which of these types is the higher-speed.

They should first be reduced to a common head, say 1 ft. Dividing the horse-power in each case by $\sqrt{h^3}$ and the speed by $\sqrt{h}$, the result is

A..2.07 h.p. at 50 rev. per min.
B..2.50 h.p. at 47.4 rev. per min.

A runs at a higher speed than B, but B develops more power than A. One cannot be sure but that if B were reduced in diameter until it developed the same power as A, it might run at as high a speed as A, or higher.

DIAGRAM OF GAUGE-BOARD AND AUXILIARY VACUUM APPARATUS FOR MEASURING VELOCITIES THROUGH GUIDE OPENINGS AND DRAFT-TUBE



A-Pitot Reading at Bottom of Guide Opening
B-Pressure " " " " " "
C-Pitot " " Top " " "
D-Pressure " " " " " "
E-Pitot " from Movable Point in Draft-Tube
F-Pressure " " " " " "
G-Pressure " " Wall of Draft-Tube at same Elevation as Movable Point.

FIG. 3.

Next reduce them to the same power, say 1 h.p. It is known that the power of wheels of homologous design varies directly as the

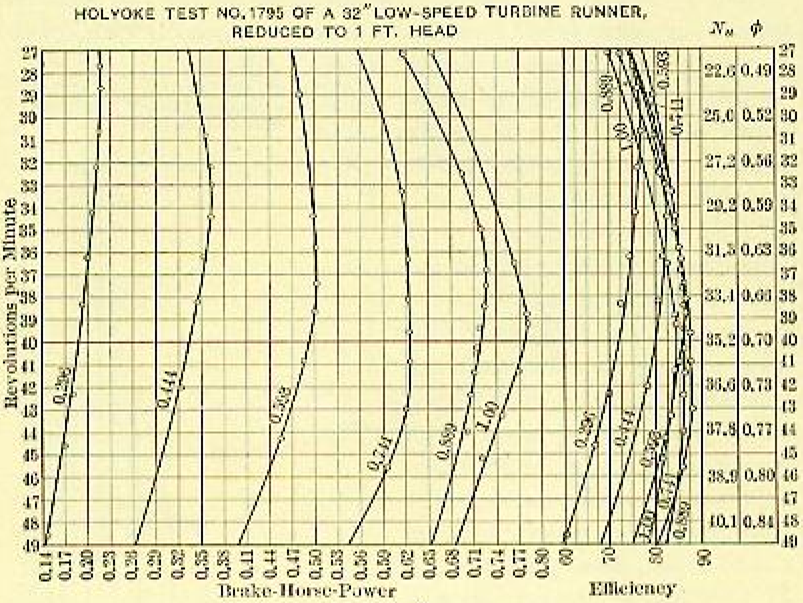

HOLYOKE TEST NO. 1795 OF A 32" LOW-SPEED TURBINE RUNNER, REDUCED TO 1 FT. HEAD
Revolutions per Minute
Brake-Horse-Power
Efficiency
N_u ϕ

FIG. 4.

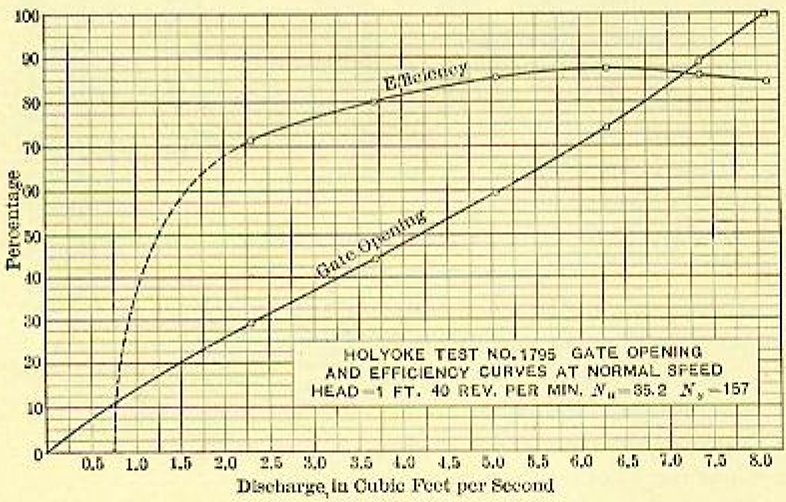

Percentage
Efficiency
Gate Opening
Discharge in Cubic Feet per Second
HOLYOKE TEST NO. 1795 GATE OPENING AND EFFICIENCY CURVES AT NORMAL SPEED HEAD = 1 FT. 40 REV. PER MIN. N_u = 35.2 N_s = 157

FIG. 5.

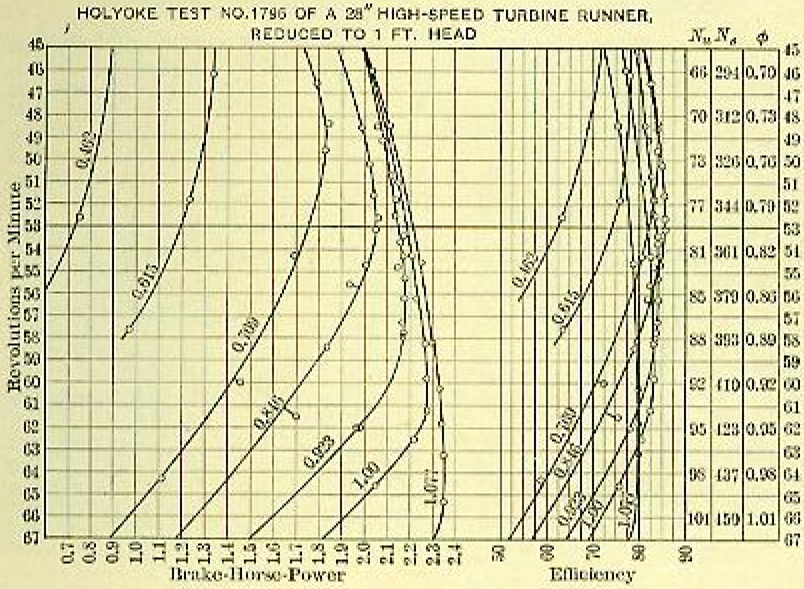

HOLYOKE TEST NO.1796 OF A 28" HIGH-SPEED TURBINE RUNNER,
REDUCED TO 1 FT. HEAD
Revolutions per Minute
Brake-Horse-Power
Efficiency
N_u N_s ϕ
66 294 0.70
70 312 0.73
73 326 0.76
77 344 0.79
81 361 0.82
85 379 0.86
88 393 0.89
92 410 0.92
95 423 0.95
98 437 0.98
101 450 1.01

Fig. 6.

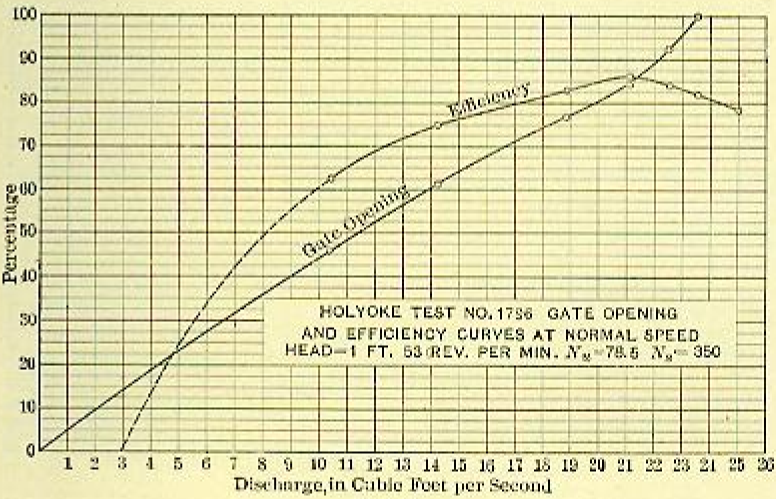

Efficiency
Gate Opening
Percentage
HOLYOKE TEST NO.1736 GATE OPENING
AND EFFICIENCY CURVES AT NORMAL SPEED
HEAD=1 FT. 53 REV. PER MIN. N_u=78.5 N_s=350
Discharge, in Cubic Feet per Second

Fig. 7.

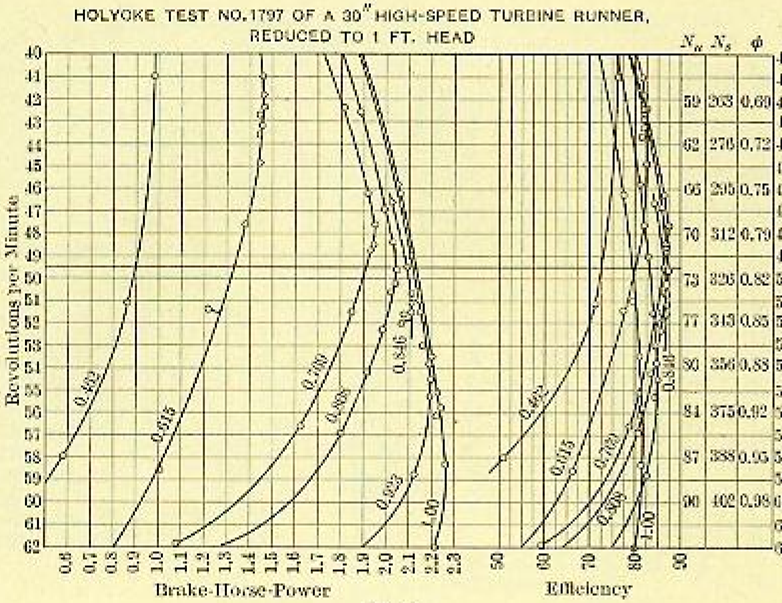

HOLYOKE TEST NO. 1797 OF A 30" HIGH-SPEED TURBINE RUNNER, REDUCED TO 1 FT. HEAD
Revolutions per Minute
Brake-Horse-Power
Efficiency
N_u N_s ϕ
59 263 0.69
62 276 0.72
66 295 0.75
70 312 0.79
73 326 0.82
77 343 0.85
80 356 0.88
84 375 0.92
87 388 0.95
90 402 0.98

Fig. 8.

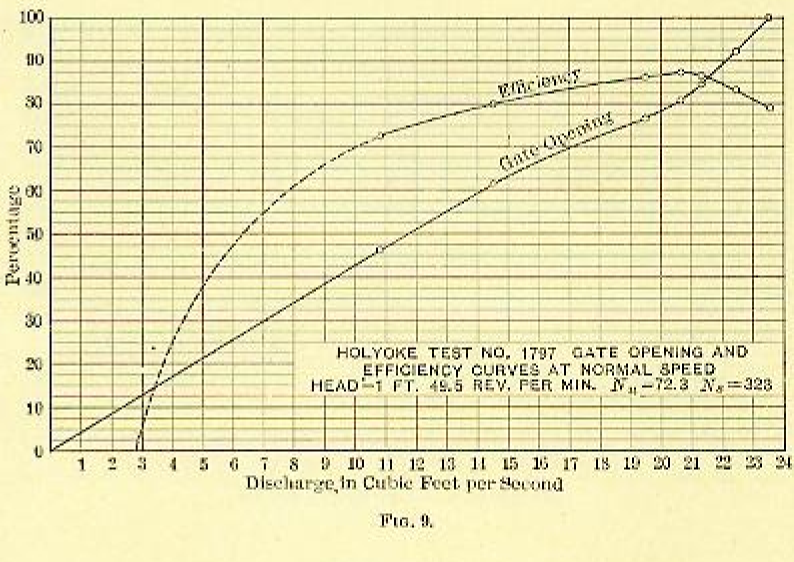

Percentage
Discharge, in Cubic Feet per Second
Efficiency
Gate Opening
HOLYOKE TEST NO. 1797 GATE OPENING AND EFFICIENCY CURVES AT NORMAL SPEED
HEAD=1 FT. 49.5 REV. PER MIN. N_u=72.3 N_s=323

Fig. 9.

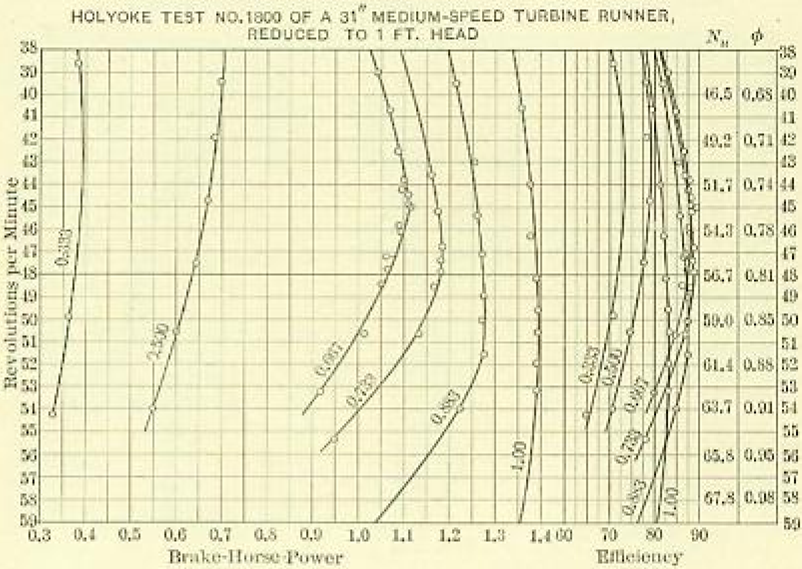

HOLYOKE TEST NO.1800 OF A 31" MEDIUM-SPEED TURBINE RUNNER, REDUCED TO 1 FT. HEAD
N_u ϕ
46.5 0.68
49.2 0.71
51.7 0.74
54.3 0.78
56.7 0.81
59.0 0.85
61.4 0.88
63.7 0.91
65.8 0.95
67.8 0.98
Revolutions per Minute
Brake-Horse-Power
Efficiency
0.333
0.500
0.667
0.733
0.883
1.00

FIG. 12.

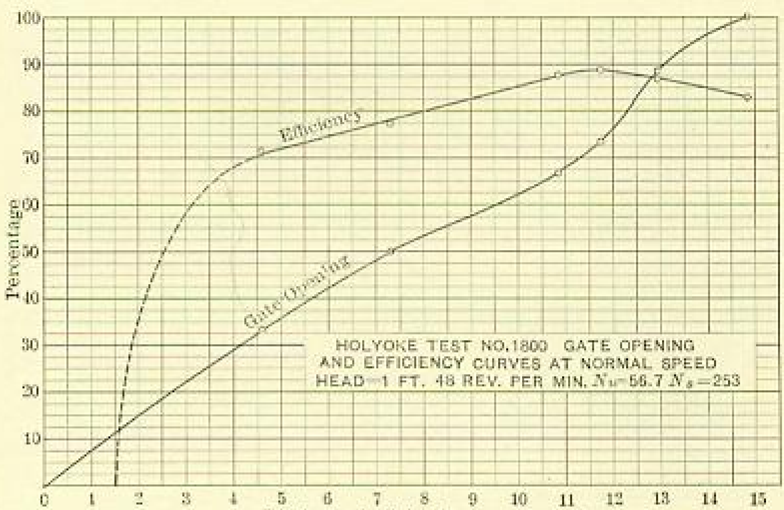

Efficiency
Gate Opening
Percentage
HOLYOKE TEST NO.1800 GATE OPENING
AND EFFICIENCY CURVES AT NORMAL SPEED
HEAD=1 FT. 48 REV. PER MIN. N_u=56.7 N_s=253
Discharge, in Cubic Feet per Second

FIG. 13.

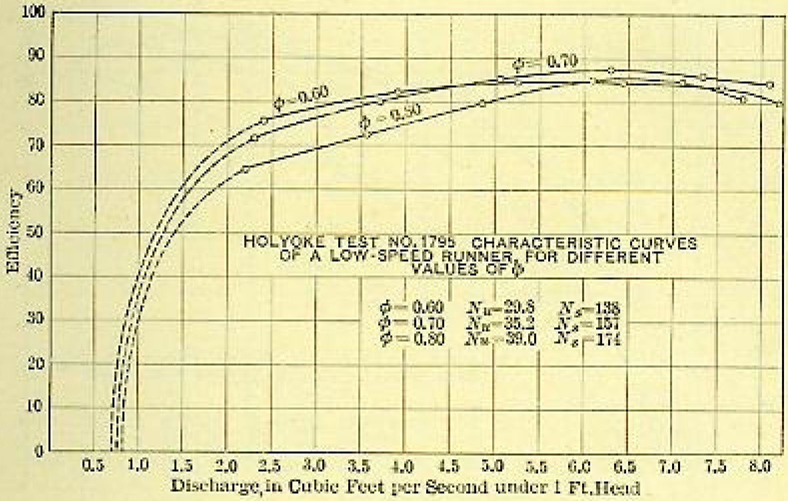

HOLYOKE TEST NO. 1795 CHARACTERISTIC CURVES OF A LOW-SPEED RUNNER, FOR DIFFERENT VALUES OF φ
φ = 0.60 N_u = 29.8 N_s = 138
φ = 0.70 N_u = 35.2 N_s = 157
φ = 0.80 N_u = 39.0 N_s = 174
φ = 0.70
φ = 0.60
φ = 0.80
Efficiency
Discharge, in Cubic Feet per Second under 1 Ft. Head

Fig. 14.

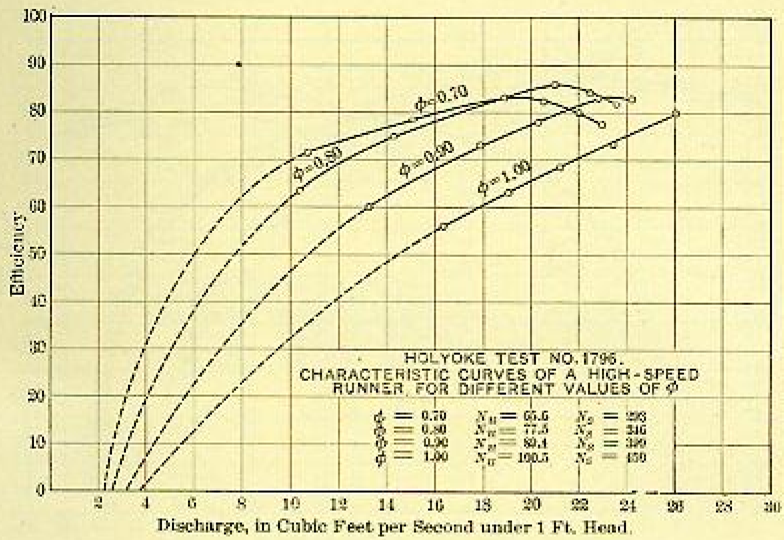

HOLYOKE TEST NO. 1796. CHARACTERISTIC CURVES OF A HIGH-SPEED RUNNER, FOR DIFFERENT VALUES OF φ
φ = 0.70
φ = 0.80
φ = 0.90
φ = 1.00
Efficiency
Discharge, in Cubic Feet per Second under 1 Ft. Head.

Fig. 15.

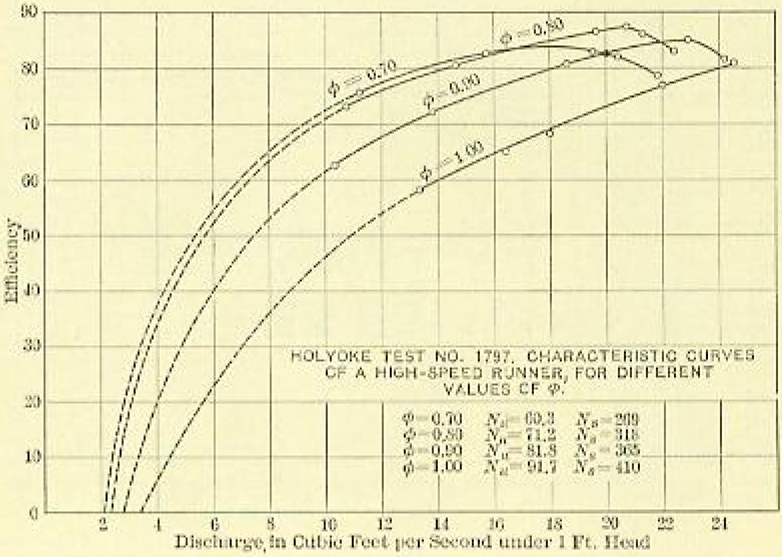

Efficiency
Discharge, in Cubic Feet per Second under 1 Ft. Head
φ = 0.70
φ = 0.80
φ = 0.90
φ = 1.00
HOLYOKE TEST NO. 1797. CHARACTERISTIC CURVES OF A HIGH-SPEED RUNNER, FOR DIFFERENT VALUES OF φ.
φ = 0.70 N_u = 69.3 N_s = 269
φ = 0.80 N_u = 71.2 N_s = 318
φ = 0.90 N_u = 81.8 N_s = 365
φ = 1.00 N_u = 91.7 N_s = 410

FIG. 16.

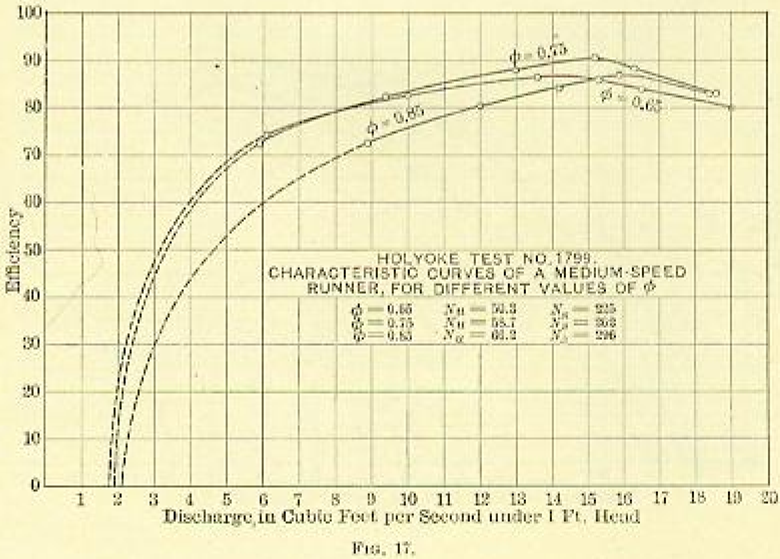

Efficiency
Discharge, in Cubic Feet per Second under 1 Ft. Head
φ = 0.55
φ = 0.65
φ = 0.85
HOLYOKE TEST NO. 1799.
CHARACTERISTIC CURVES OF A MEDIUM-SPEED RUNNER, FOR DIFFERENT VALUES OF φ

FIG. 17.

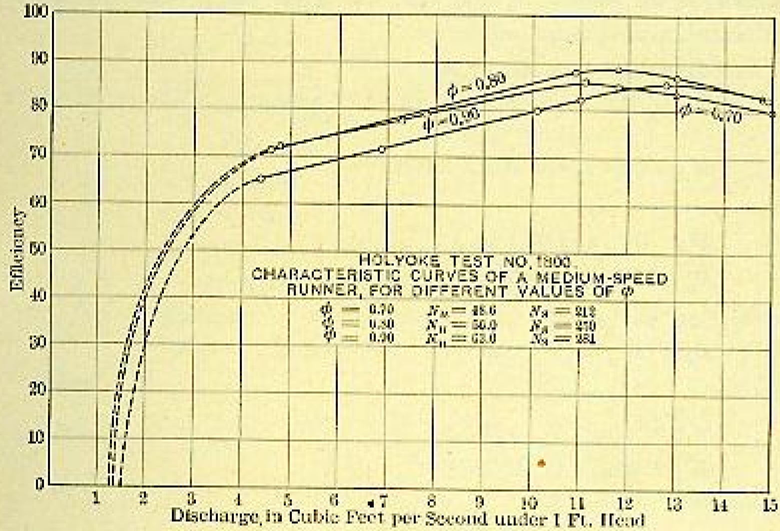

HOLYOKE TEST NO. 1800.
CHARACTERISTIC CURVES OF A MEDIUM-SPEED
RUNNER, FOR DIFFERENT VALUES OF ϕ
ϕ = 0.80
ϕ = 0.90
ϕ = 0.70
Efficiency
Discharge, in Cubic Feet per Second under 1 Ft. Head

FIG. 18.

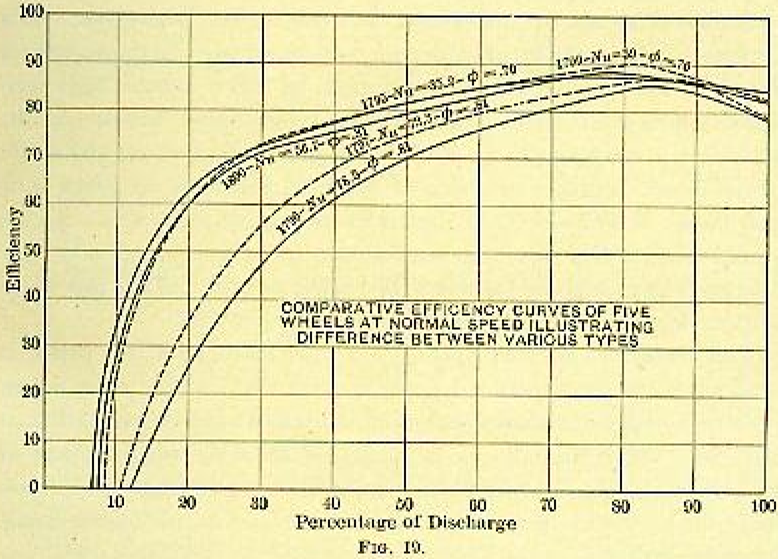

COMPARATIVE EFFICENCY CURVES OF FIVE
WHEELS AT NORMAL SPEED ILLUSTRATING
DIFFERENCE BETWEEN VARIOUS TYPES
Efficiency
Percentage of Discharge

FIG. 19.

squares of the diameters, and that the speed varies inversely as the diameters, with constant efficiency.

Let D_a = the diameter of A,

D'_a = the diameter of a wheel of homologous design, having a capacity of 1 h.p.,

N'_a = the speed of the latter.

Then

$$D^2_a : D'^2_a = 2.07 : 1$$

$$D_a : D'_a = \sqrt{2.07} : 1$$

also

$$D_a : D'_a = N'_a : 50$$

then

$$\sqrt{2.07} : 1 = N'_a : 50$$

$$N'_a = 50 \sqrt{2.07} = 72$$

Similarly $N'_b = 75$.

It is now plain that B is the higher-speed type, because, when both wheels are developing 1 h.p., B runs at 75 rev. per min., whereas A runs at only 72 rev. per min. As will be explained later, these are the "unit speeds" of these two wheels.

It should be clearly understood that this term, "high-speed," thus involves a consideration of power as well as speed. It is not related directly to the peripheral speed, although, as stated before, high peripheral speed is usually an attribute of a high-speed wheel. At the same time, it may also be an attribute of a low-speed wheel, and therefore it is not a proper criterion of speed in the sense in which it is here used. Wheel No. 1800 illustrates this point very well. At the normal speed of this wheel, the coefficient of peripheral speed, $\phi = 0.81$. This value is as high as for any of the other wheels, and yet this wheel is lower in speed than all the others, except No. 1795.

Unit Speed.—The "unit speed," N_u, used throughout this paper, is in all cases the speed, under 1 ft. head, of a wheel of the same design as the one under consideration, and of such size as to develop 1 h.p. at full gate. Other units might be used and serve the same purpose of comparison, but in America these are the most convenient. In Europe, under the metric system, the practice is to reduce to one metric horsepower, under 1 m. head. The result is termed the "specific speed," and is usually designated by the symbol, N_s.

The general equation for N_u is developed as follows:

N = speed of any runner,
P = power of same,
D = diameter of same,
h = head on same,
h_u = unit of head,
P_u = unit of power,
N' = speed of above runner under unit head,
P' = power of above runner under unit head,
D_u = diameter of homologous runner developing unit power under unit head.

Then

$$N' : N = \sqrt{h_u} : \sqrt{h} \qquad N' = N\sqrt{\frac{h_u}{h}} \quad \dots\dots (a)$$

$$P' : P = h^{\frac{3}{2}}_u : h^{\frac{3}{2}} \qquad P' = P\left(\frac{h_u}{h}\right)^{\frac{3}{2}} \quad \dots\dots (b)$$

$$P' : P_u = D^2 : D_u^2 \qquad D_u = D\sqrt{\frac{P_u}{P'}} \quad \dots\dots (c)$$

$$\frac{N'}{N_u} = \frac{D_u}{D} \quad \dots\dots (d)$$

Substituting Equations a, b, and c in Equation d.

$$\frac{N\sqrt{\frac{h_u}{h}}}{N_u} = \frac{D\sqrt{\frac{P_u}{P\left(\frac{h_u}{h}\right)^{\frac{3}{2}}}}}{D} = \sqrt{\frac{P_u}{P\left(\frac{h_u}{h}\right)^{\frac{3}{2}}}}$$

Simplifying,

$$N_u = \frac{N \times \sqrt{P} \times h_u^{\frac{5}{4}}}{\sqrt{P_u} \times h^{\frac{5}{4}}} \quad \dots\dots (1)$$

With $P_u = 1$ h.p. and $h_u = 1$ ft. this becomes

$$N_u = \frac{N \times \sqrt{P}}{h^{\frac{5}{4}}} \quad \dots\dots (2)$$

Given the power and speed of a wheel under any head, the unit speed may be determined by substituting in Equation 2. It is not necessary to go through the process of double reduction as was done in comparing wheels A and B. For example, take wheel A, just considered.

Here

$$h = 100 \qquad P = 2\,080 \qquad N = 500$$

Substituting in Equation 2,

$$N_u = \frac{500 \times \sqrt{2\,080}}{100^{\frac{5}{4}}} = 72$$

which checks the former computation. Under the metric system, $P_u = 0.98$ h.p. and $h_u = 3.28$ ft. Substituting in Equation 1.

$$N_s = \frac{N \times \sqrt{P} \times 3.28^{\frac{5}{4}}}{\sqrt{0.98} \times h^{\frac{5}{4}}} = 4.46 \times \frac{N \times \sqrt{P}}{h^{\frac{5}{4}}} \dots\dots\dots (3)$$

or

$$N_s = 4.46\ N_u.$$

Hence, to transform a unit speed from the English to the metric system, multiply N_u by the coefficient, 4.46.

This method of classifying wheels according to their unit speeds is one of great convenience. Suppose, for example, that it is desired to equip a plant with units using runners having a maximum capacity of 1 000 h.p. each, at 200 rev. per min., under 40 ft. head. From Equation 2, $N_u = 63.3$, and therefore only wheels approximating this unit speed need be considered.

Again, suppose that the sizes of the units and the speed for a prospective plant under 100 ft. head have not been determined. It is desired to know what results could be obtained with a certain runner, the characteristics of which are well suited to the proposed service. The unit speed of this runner is 70. Substituting in Equation 2,

$$70 = \frac{N \times \sqrt{P}}{100^{\frac{5}{4}}} \qquad N \times \sqrt{P} = 22\,136.$$

This type of wheel will give any combination of power and speed, whatsoever, that will satisfy this equation. With the inverted scale of a duplex slide-rule, innumerable combinations of power and speed may be read off as rapidly as one can move the runner. This is a great convenience in preliminary calculations. In the above case, for instance, any of the following combinations of power and speed can be produced, depending on the size of the runner used.

1 000	h.p.	at 700	rev. per min.
1 360	"	at 600	"
1 960	"	at 500	"
3 060	"	at 400	"
5 450	"	at 300	"
12 200	"	at 200	"
49 000	"	at 100	"

Classification According to Unit Speeds.—Some method of classifying turbine runners on the basis of the duty which they are able to perform is essential to an intelligent study of turbine characteristics. It is obviously futile to compare the relative merits of two wheels which cannot accomplish the same work. They are in different classes, and, as such, might naturally be expected to possess different characteristics.

Since the unit speed is a measure of complete performance, both as to speed and power, it supplies an admirable basis for the desired classification. In this it should be noted that such a classification is independent of the details of design. The angles of the vanes, general proportions, peripheral speed, etc., may all be different, and yet the wheels are classified as of the same type if they have the same unit speed, because they, and they alone, are capable of doing the same work.

In Europe it is the practice to classify runners according to their coefficients of peripheral speed. A "low-speed" runner has a value of ϕ equal to about 0.6, a "normal-speed" runner, about 0.7 and a "high-speed" runner, about 0.8. These distinctions, however, embody the element of capacity as well as speed, although in an indirect and rather indefinite way. This is because a value of $\phi = 0.6$ is not ordinarily used, except with wheels of low capacity, 0.7 with those of normal capacity, and 0.8 with those of high capacity. The distinguishing element of capacity is really there, although its influence is not definitely measured. Such a method, of course, cannot be as exact as that based on unit speeds. In the latter case, both capacity and speed are definite mathematical factors.

Influence of ϕ Upon the Characteristic Curves.—The extent to which changes in the value of ϕ at normal speed affect the efficiency curves of turbine runners is difficult to determine, because, with wheels of rational design, the value of ϕ, as just stated, keeps step with the variation in the capacity of the wheel. It is, of course, a principle of economical design that a runner shall be made no larger in diameter than necessary. For a given speed, the lower the value of ϕ the smaller the wheel, and hence a low value of ϕ is suitable to a low-capacity wheel, and a high value to a wheel of large capacity. It may be generally stated as a principle of rational design that ϕ should always be made as low as possible, and that it should be forced up to high values only in order to get additional capacity.

Wheel No. 1800 was an experimental runner designed contrary to this principle for purposes of investigation. It is of irrational design in that it is a wheel of comparatively low capacity with a high value of ϕ. The value of ϕ for a wheel of this capacity need not be over 0.7 at normal speed, whereas it was 0.81, according to test.

The results obtained with this wheel appear to indicate that, with wheels of similar capacity, the value of ϕ at normal speed has no very important effect on the characteristic curves, although a high value of ϕ does show some tendency to pull down the efficiency. Fig. 19 shows that the three wheels of medium and low capacity gave similar results at normal speed, although the values of ϕ were 0.70, 0.76, and 0.81, respectively. The three wheels, however, which had the same value of ϕ, namely, 0.81, but differed considerably in capacity, ranging from medium power to the extreme limit of high power, gave widely different curves. All of which seems to indicate that forced capacity is the chief factor which causes the variation.

A comparison of Figs. 14, 17, and 18 shows a further similarity between the three wheels having different values of ϕ. The variation in efficiency at part gates, due to a variation of 0.1 in ϕ both above and below its normal value, is quite similar in all three cases. In fact, what difference there is between the three wheels is in favor of the high value, $\phi = 0.81$. This wheel shows less drop in efficiency at part gates, due to an increase in speed, than either of the others.

In view of this evidence, and for lack of any experimental proof to the contrary, the writer believes it safe to say that the efficiency is not a function of ϕ, to any considerable extent.

Principal Difference Between Low-Speed and High-Speed Wheels.—Before proceeding further, it may be well to state that the deductions made from these tests should not be given too strict an application. Results necessarily vary with changes in design, and, while the results obtained with these wheels are of a nature broadly representative of other wheels of similar capacity, nevertheless, considerable variation in individual cases must be expected.

The most significant fact to be observed, in connection with the efficiency-speed curves of these wheels at various gates (Figs. 4, 6, 8, 10, and 12), is that all of them peak at different speeds. The most efficient speed of the wheel varies with the gate opening, increasing as the latter increases. This is true of all reaction turbines, whether of low or high speed, but not in the same degree. In the case of high-speed wheels, the difference between the best low-gate speed and the best full-gate speed is much greater than in the case of medium-speed and low-speed wheels. Table 6 shows the comparison between 0.4 gate and full gate for all the wheels under 1 ft. head, listed in the order of their capacity.

TABLE 6.

Wheel No.	1795	1800	1799	1797	1796
Most efficient full-gate speed	40	52	48	58	64
Most efficient 0.4-gate speed	31	41	38	35	35
Percentage of drop in speed	22.5	21.1	20.8	39.6	45.3
N_s	35.2	56.7	59.0	72.3	78.5

It may be observed from Table 6 that the drop for low-speed wheel No. 1795 and for the two medium-speed wheels, Nos. 1799 and 1800, is about the same, whereas it is nearly twice as great for the high-speed wheel No. 1797, and even greater for the extremely high-speed wheel No. 1796. This drop does not seem to vary much with the capacity of the wheel for values of N_s below 60, approximately. Above $N_s = 60$, or thereabouts, the drop increases rapidly as N_s increases.

When operating at constant speed, the inevitable result of this variation in the most efficient speed of the wheel is a sacrifice of efficiency at all gates except one. For example, if a speed is selected which will give the highest efficiency which it is possible to obtain at 0.75 gate, that speed will be too high to give the best obtainable efficiency at the lower gates and too low to give the best results at the higher gates. The best speed at which to run a wheel, therefore, depends on which gate opening is the most important as regards efficiency. The best speed is the one which strikes the peak of the efficiency curve for that gate.

Naturally, the less variation there is in the best speed from low gate to full gate the less sacrifice of efficiency there will be when the wheel is run at constant speed. It has just been shown that this variation is much less with wheels of low and moderate unit speeds than with those of high unit speed, and therein lies the most important difference between these several types.

As an illustration of this point, compare low-speed wheel No. 1795 with high-speed wheel No. 1796 (Figs. 4 and 6). The maximum efficiency of the former at 0.444 gate is 82%, occurring at a speed of 34.5 rev. per min. The efficiency at the normal speed of 40 rev. per min., however, is 80%, and hence 2% has been sacrificed at this gate to the requirement of constant speed.

In the case of No. 1796, the maximum efficiency at 0.462 gate is 74.8%, occurring at 37.8 rev. per min. At normal speed of 53 rev. per min. the efficiency is only 62.5% and about 12% has been sacrificed to the requirement of constant speed. In addition to this, the peak of the curve of No. 1795 is about 7% higher than that of No. 1796, and hence the total difference in favor of the low-speed wheel at this gate is about 17 per cent. Fig. 19 shows graphically what a wide difference exists between the efficiency curves of medium-speed and high-speed wheels. The relatively poor showing of the two high-speed wheels, Nos. 1796 and 1797, is due to the two characteristic deficiencies just alluded to, namely, rapid drop of the efficiency-speed curves at part gate, and relatively lower maximum efficiencies at all gates. The latter condition is theoretically inevitable. It is impossible to discharge a large quantity of water through a wheel of given size as efficiently as a moderate quantity.

Another peculiarity which is distinctly characteristic of high-speed wheels is that the power at the high gates increases as the speed increases beyond normal. By over-gating the wheel it is possible to hold up a substantial increase of power with good efficiency at extremely high values of ϕ. For example, wheel No. 1796 (Fig. 6), at 1.077 gate, increases constantly in power and efficiency until ϕ has reached a value of 1.00. No. 1797 (Fig. 8), at 1.00 gate, increases until $\phi = 0.97$.

This feature of high-speed wheels is a valuable one, because it fits them especially for the only work for which they are at all adapted, namely, low heads. Low-head plants are almost invariably subject to relatively large reductions of head, due to back-water in the tail-race. If the head drops very much, the value of ϕ necessary to maintain constant speed will run up very high. A wheel which gives more power at a high value of ϕ than it does at the normal value, is a great advantage in such a case, because it increases the capacity of the turbine by just that much at a time when the power is perilously deficient on account of the reduced head. At such a time, it is necessary to open the turbine gates farther than would be advisable under the normal head. Under the latter, such over-gating would be wasteful, because the increase in power is very small and not at all porportionate to the increase in discharge. With some wheels, the power is even less than it was at the previous gate, on account of excessive drop in efficiency. Under the reduced head, however, with high values of ϕ, the increase in power is quite considerable. With No. 1796 (Fig. 6), for instance, the increase in power from 1.00 gate to 1.077 gate, at normal speed, is practically nothing, due to a 4% drop in efficiency. At $\phi = 1.00$, however, the increase in power is about 20%, and the efficiency is about 6% better. The over-gate, therefore, should be used only under a reduced head.

Reference to Figs. 4, 10, and 12 will show that the wheels of lower unit speed do not possess this characteristic. In every case, the high-gate curves fall off at the high values of ϕ.

Considerations Affecting the Choice of Wheels for Low Heads.—It is futile, of course, to attempt to define within fixed limits the heads which should be included in this classification. Some plants under moderately high heads operate under conditions very similar to those which are distinctly characteristic of the typical low-head plant. Others under low heads possess some of the characteristics common to high heads. If any useful deductions are to be drawn from such a study, they must be based on a consideration of typical conditions, with a liberal allowance for the inevitable exceptions, which require special treatment.

A typical low-head plant may be illustrated by an installation where the head is created by building a dam across a large river flowing through a comparatively flat country. In such cases, it is impracticable to build a high dam, because the water cannot be confined behind it.

Having, probably, little storage area, other than the bed of the stream, such a power cannot usually be developed much beyond the minimum flow of the stream. If it were possible to impound the water which is not used during light loads for use during peak loads, then the part-gate efficiency of the turbines for such a plant would be of importance; but, usually, the water which is not used during light loads goes over the dam, and therefore the efficiency of the turbines at such times is relatively unimportant. Under such conditions, the poor part-gate efficiency of high-speed wheels is of no importance.

The important considerations are high efficiency at normal gate, as affecting the normal quantity of power which can be developed from the available flow; high speed, as affecting the cost of the electrical machinery and, to a limited extent, the cost of the turbines; high power at constant speed under reduced heads due to back-water, which is the inevitable coincident of flood conditions.

In respect to the first consideration, the high-speed wheel is somewhat deficient, depending on the extent to which the wheel is pushed for speed. Taking the efficiency of No. 1799, 90.43%, as the standard for a wheel of medium speed, the high-speed wheel No. 1797, yielding 87.39%, suffers about 3% by comparison. Wheel No. 1796, which is a more extreme type than No. 1797, gave 86.15%, or about 4½% less.

Another point to be noted is that the maximum efficiency of high-speed wheels occurs at a higher gate opening than that of lower-speed wheels. With the latter it usually occurs at from 0.70 to 0.75 gate, whereas with the former it is usually from 0.80 to 0.85 gate. In order to provide for the customary overload of 25% with high-speed wheels, it is necessary that they shall carry the normal load at a lower gate than the most efficient one. This causes a further loss of efficiency.

In respect to the second and third considerations, the high-speed wheel has an overwhelming advantage, and is undoubtedly the best wheel for typical low-head service. Table 7 gives a comparison of the results which could be accomplished by the three types of runner, Nos. 1796, 1797, and 1799, respectively, if applied to a plant having a maximum capacity of 500 h.p. per runner under 20 ft. head. Low head is assumed to be 13 ft.

TABLE 7.

Wheel. No.	20 FEET.			13 FEET.			N_u
	Horse-power.	Revolutions per minute.	Efficiency.	Horse-power.	Revolutions per minute.	Efficiency.	
1796	500	149	82.0	279	149	80.0	78.5
1797	500	186	83.0	277	136	80.5	72.3
1799	500	112	82.7	246	112	79.5	59.0

Runners for Intermediate Heads.—It is as the head increases and back-water conditions become less important, and as the speed of the ordinary runner, increasing with the head, more nearly approaches an economical value, that the use of the high-speed runner becomes more questionable. As these conditions change, the considerations which favor high-speed wheels gradually disappear, and others of more importance become dominant.

As the head increases the storage capacity usually increases, and, with the capacity to impound water for peak loads, the part-load economy of the turbines becomes highly important, and the medium-speed wheel with its high efficiency is clearly indicated. The higher the head the more valuable a cubic foot of water becomes, and the more important it is to save it. High speed becomes a subordinate consideration. If the additional power which can be developed by the lower-speed wheels with their high efficiency is worth more than the interest on the saving in electrical apparatus which would result from the use of high-speed wheels, then the former are certainly the more economical proposition. This principle, of course, is fundamental and commonly recognized, but is not always intelligently applied, for lack of specific knowledge regarding the disadvantages incidental to high speed. If purchasers were as well informed as they ought to be, as to the deficiencies of high-speed runners, there would be fewer of them in use at the present time. It is hoped that this paper may supply some welcome information.

Table 8 shows comparative efficiencies at various gates for medium-speed wheel No. 1799 and high-speed wheel No. 1796. In view of these figures, a very weighty reason would be required to justify the use of No. 1796 instead of No. 1799, if efficiency is of any importance.

High-Head Wheels.—Runners for high heads pass through all the intermediate stages of design between the mixed-flow wheel and the purely inward-flow or so-called Francis type, according to the head available and the power developed. As the head increases the unit speed tends to decrease, and the design approaches that of the low-capacity Francis type (No. 1795). Although the designs of these wheels vary considerably, the characteristic curves vary but slightly from the best results of medium-speed wheels.

TABLE 8.

Wheel No.	Gate Opening.									
	0.30	0.40	0.50	0.60	0.70	0.75	0.80	0.85	0.90	1.00
1799	68.5	76.5	82.0	86.0	88.0	90.4	90.0	89.2	88.0	83.0
1796	41.0	55.5	66.0	73.5	79.5	82.0	84.5	86.2	85.0	82.0

The efficiency curves of wheel No. 1795 (Fig. 5) at normal speed, with $N_u = 35.2$, and wheel No. 1799 (Fig. 11), with $N_u = 59$, are very similar. On account of the discrepancy between the peaks of these two curves, the part-gate efficiencies are about the same (Fig. 19). Otherwise, the latter would be a trifle higher for the low-speed wheel. The general tendency is for the efficiency, particularly at low gates, to improve as N_u decreases.

The reason wheel No. 1795 did not show as high maximum efficiency as No. 1799 is that, being a low-speed wheel, it developed much less power under test, and hence the loss, due to mechanical friction, was a much greater percentage of the total power developed than was the case with No. 1799. Wheel No. 1795, operating under a head of several hundred feet, particularly in large diameters, would undoubtedly develop 90% efficiency or better. The effect of mechanical friction, which has been discussed at length, is especially detrimental to the efficiency of a low-power wheel such as this, when tested at Holyoke. No high-head wheel can by any possibility show as high efficiency in the Holyoke flume as it would in regular operation, assuming, of course, that in the latter case it is properly installed.

DISCUSSION

LEWIS F. MOODY, ESQ.* (by letter).—This may be considered the most noteworthy contribution on the subject since the late R. H. Thurston, M. Am. Soc. C. E., presented his paper on "The Systematic Testing of Turbine Water-Wheels in the United States." This paper was presented before the American Society of Mechanical Engineers in 1887.† The turbine tests reported by Professor Thurston, like those reported by Mr. Larner, were made at the Holyoke flume, and were notable for the efficiencies developed, the highest which had been obtained in reliable tests up to that time. The tests were made in 1883.‡ Mr. Moody.

The wheel tested by Professor Thurston was a 36-in. R. H. "Hercules" turbine, built by the Holyoke Machine Company, and the maximum efficiency, on each of the three tests made by him, was 86.94%, 86.18%, and 87.08%, respectively (Tables IV, V, and VI). The specific speed corresponding to the point of maximum efficiency is 196, in the metric system.

Since the date of Professor Thurston's paper, great progress has been made in water-wheel design, both in America and in Europe. Mr. Larner's paper, and the remarkable results reported by him, indicate the recent advances which have been made in raising both the efficiency and speed of turbines. Mr. Larner's wheels have shown specific speeds as high as 450 (in the metric) or 100, in the foot-pound system, with an efficiency of practically 80 per cent.

As efficiency and specific speed are the most important issues raised in Mr. Larner's paper, the writer has endeavored to show how they are related, and also how various turbines compare in regard to these two factors, by constructing a curve, Plate XXXII, with specific speed for the scale of abscissas, and efficiencies for the ordinates, using the results of all available tests showing the highest known efficiencies. This curve furnishes a basis for comparing the performance of any turbine with the best previous results shown by turbines of the same specific speed.

In plotting it, the speed, head, and delivered horse-power at the point of highest efficiency of a given wheel have been used in calculating the specific speed. That is, a wheel, when running at a gate opening in excess of that giving maximum efficiency, and at a speed in excess of the normal speed, may develop a much higher specific speed than that plotted; but, for the purpose of comparison, the specific speed corresponding to the run on which the efficiency was developed, has been used in plotting the point. On wheels developing unusually high

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† *Transactions*, Am. Soc. Mech. Engrs., Vol. VIII, p. 359.
‡ Bodmer, "Hydraulic Motors," pp. 401, 481–490.

Mr. Moody. speed, additional points have, in some cases, been plotted, giving the efficiencies corresponding to higher specific speeds for the same wheel, and showing by a short curve how the efficiency drops on a given wheel as the specific speed is increased beyond the normal.

Thus, points 24 and 27 on Plate XXXII both correspond to the same wheel, a 56-in. Samson turbine, point 24 corresponding with the maximum efficiency shown by the test, and point 27 showing that with a much greater specific speed the efficiency dropped somewhat but not greatly in this case (this curve is indicated by a dash and two dots). Other tests of various Samson turbines also give points lying on this same curve, indicating similarity of design.

In the same way, a Holyoke test (No. 1778), reported by the Allis-Chalmers Company, has been plotted at several points, giving the curve shown by a fine line; and the test of a Swedish turbine (built by the Karlstad mek. Verkstad, Filiale in Kristinehamn, Sweden),* is given by a dotted line, *H-I*.

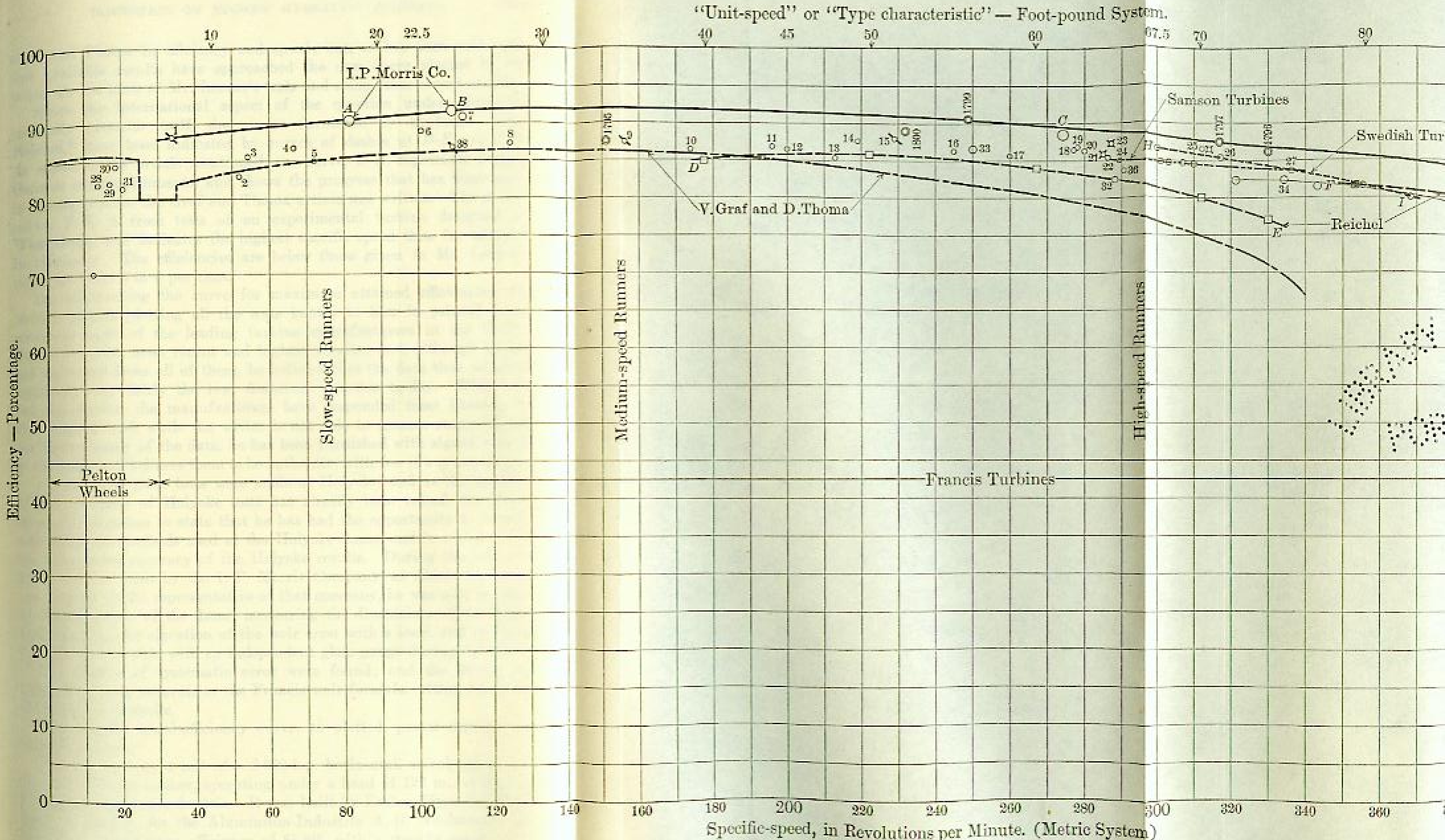
It will also be noticed that the highest curve given (a solid, heavy line) passes through two points of Mr. Larner's wheel No. 1796. This indicates that, while no turbine is known (to the writer, at least) which exceeds in efficiency the results shown by this test, No. 1796, at the speeds corresponding to it, still, the high-speed point ($N_s = 446$) was not obtained at the point of maximum efficiency of this wheel; and it would seem probable that a wheel could be designed for this particular specific speed which would exceed this value of the efficiency. (The specific speeds, N_s, are in the metric system throughout, in order to adhere to an international standard.) The equivalent "unit speeds," to use Mr. Larner's term, or "type characteristics," Mr. Zowski's term—both meaning specific speed converted to the foot-pound system of measures—are given at the top of the diagram.†

In order to show the recent progress of water-wheel design, and to compare American and foreign achievements in this branch of engineering activity, a curve constructed by Messrs. V. Graf and D. Thoma,‡ has been reproduced by dots and dashes. This curve was at that time of the same nature, and was intended to show the same things as the writer's full-line curve on Plate XXXII. That is, Graf and Thoma intended their curve to represent the best known results, and, up to a specific speed of 200, it represented very fairly the state of the art at that time. Above that speed, however, their curve fell considerably below many results which had been obtained. The reason for this was that many German engineers did not then, and do not now, accept Holyoke tests as correct, and are inclined to ignore all results obtained in America. There were many American tests considerably exceeding the curve of

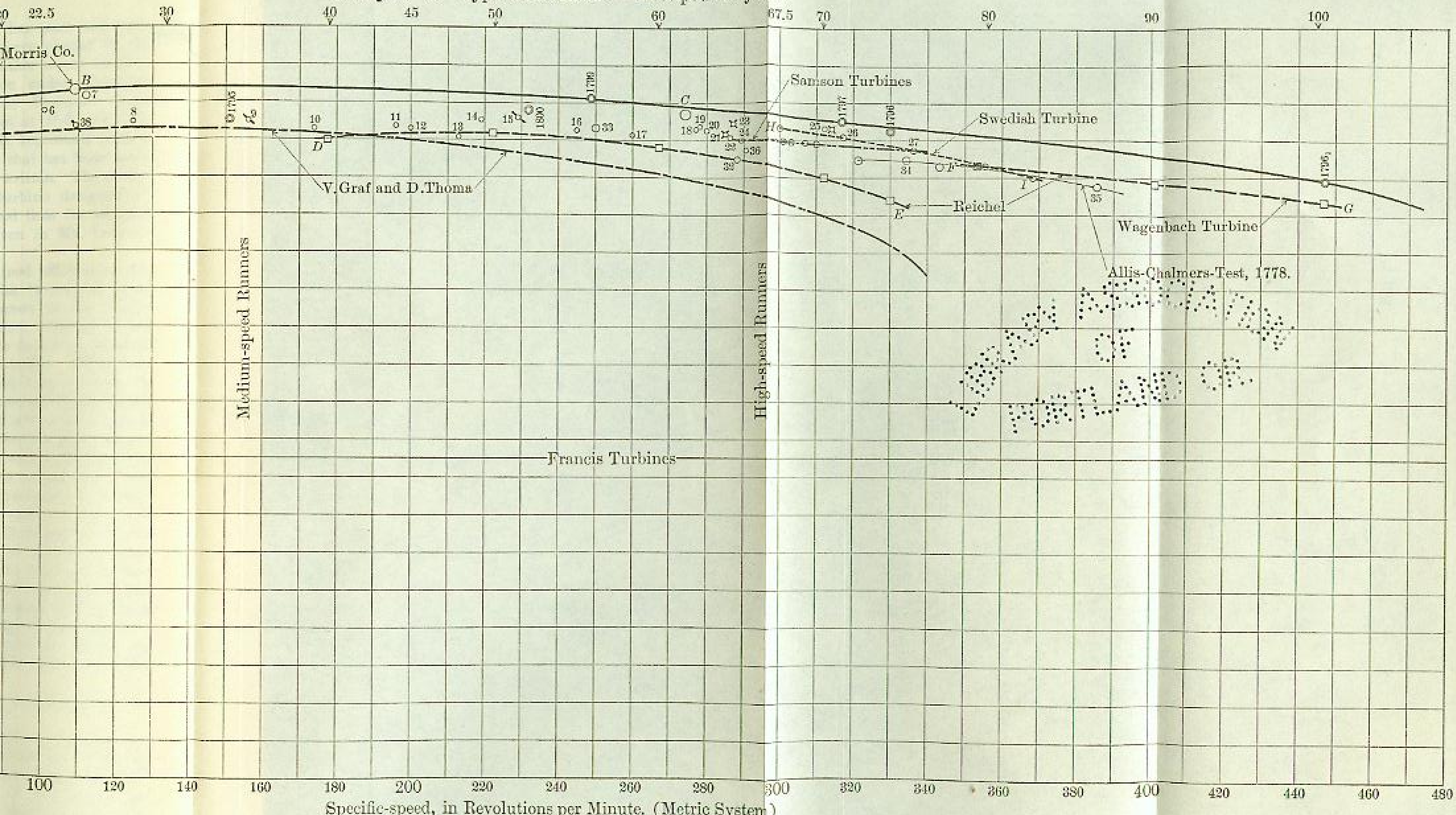
* *Zeitschrift für das gesamte Turbinenwesen*, Heft 3, V Jahrgang, Jan. 30th, 1908.

† See article by S. J. Zowski, *Engineering News*, Jan. 28th, 1909.

‡ Published in the *Zeitschrift des Vereines deutscher Ingenieure*, June 29th, 1907.



"Unit-speed" or "Type characteristic" — Foot-pound System.
Efficiency — Percentage.
Specific-speed, in Revolutions per Minute. (Metric System)
Slow-speed Runners
Medium-speed Runners
High-speed Runners
Pelton Wheels
Francis Turbines
I.P. Morris Co.
V. Graf and D. Thoma
Samson Turbines
Swedish Tur
Reichel


"Unit-speed" or "Type characteristic" — Foot-pound System.
22.5
30
40
45
50
60
67.5
70
80
90
100
Morris Co.
Samson Turbines
Swedish Turbine
V. Graf and D. Thoma
Reichel
Wagenbach Turbine
Allis-Chalmers-Test, 1778.
Medium-speed Runners
High-speed Runners
Francis Turbines
100
120
140
160
180
200
220
240
260
280
300
320
340
360
380
400
420
440
460
480
Specific-speed, in Revolutions per Minute. (Metric System)

Graf and Thoma in efficiency and speed; but, as has been said, very few available results have approached the new curve plotted by the writer on the basis of Mr. Larner's tests and some other recent results. Mr. Moody.

Since the international aspect of the question under discussion naturally presents itself, the curves furnished by Professor Ernst Reichel,* have been indicated by curves of dashes at *D-E* and *F-G*. As stated in the article mentioned, the curve, *D-E*, represents present German accomplishments, and shows the progress that has been made in Germany since the Graf and Thoma article was written. The second curve, *F-G*, is from tests of an experimental turbine designed by Wagenbach, and indicates the highest specific speed thus far attained in Germany. The efficiencies are below those given in Mr. Larner's paper by from 3 to 4 per cent.

In constructing the curve for maximum attained efficiencies, the writer, besides plotting all the tests known to him in printed form, wrote to eight of the leading turbine manufacturers in the United States for their most recent and highest results; and, although he has not yet heard from all of them, he believes that the data thus collected represent very fairly the best American practice to-day. With only two exceptions, the manufacturers have responded most liberally to this request, and, while the writer is not able to assume responsibility for the accuracy of the data, he has been furnished with signed reports of the tests, and believes them to be authentic, with few if any exceptions.

As many of these tests were made at Holyoke, and as the question of the accuracy of Holyoke tests has already been raised, the writer takes this occasion to state that he has had the opportunity to investigate fully the methods used at the Holyoke flume, and is convinced of the substantial accuracy of the Holyoke results. During the test of a 33-in. turbine built by the I. P. Morris Company, at which the writer was present as the representative of that company, he was able to check up the constants of the flume, measuring the dimensions of the brake, etc., checking the elevation of the weir crest with a level, and checking the head on the weir with an independent glass gauge during the test.

No sources of systematic error were found; and the results are believed to be as accurate as the Francis weir formula, within the limits of its use at Holyoke.

The specific speed-efficiency curve, as plotted, passes through the following points:

Point 1 is from a test of a 2 000-h.p. horizontal, spiral-casing turbine with double runner, operating under a head of 122 m., at Rauris-Kitzloch (Salzburg), Austria. It was built by Escher Wyss and Company, of Zürich, for the Aluminium-Industrie A.-G. Neuhausen, and developed a maximum efficiency of 88.2% with a specific speed of 34.

* Recently published in *Engineering News*, September 9th, 1909, p. 288.

Mr. Moody. The data for this and other tests of Escher Wyss wheels are furnished by the bulletin published by the Escher Wyss Company (1908).

Points *A* and *B* are from tests of two wheels built by the I. P. Morris Company, of Philadelphia, the guides and runners of which were designed by the writer, under the direction of Mr. W. M. White, Hydraulic Engineer of the company. Point *A* represents the 750-h.p. turbines, built for J. G. White and Company, and installed at the Comerio Falls Plant, Porto Rico, of the Porto Rico Light and Power Company. These turbines are of the horizontal, volute-casing type, with single runner and single draft-tube. They have movable guide-vanes, or "wicket-gates," with outside operating mechanism; and the runners are of bronze. When tested in place, the output being measured electrically and the water by a Francis weir, they showed an efficiency of 90.3% when developing 755 h.p. under a head of 182.75 ft. The speed was 450 rev. per min.

Point *B* is from the acceptance test of unit No. 12 at power-house No. 3 of the Niagara Falls Hydraulic Power and Manufacturing Company. The test was made in place, under operating conditions, by the engineers of that company and of the I. P. Morris Company, and was very carefully conducted. The water measurement was by weir, using the Francis formula. The turbines are of 10 000 h.p., of the horizontal, volute-casing type with double runners and two draft-tubes per unit. They have movable guide-vanes and bronze runners. The test showed that when delivering 8 934 h.p. under a head of 214.3 ft., at a speed of 299.9 rev. per min., the efficiency was 91.7 per cent. Mr. White states in a letter:

"I have no doubt but that the 10 000-h.p. units are the most efficient units in operation to-day."

Point 7, just below the curve, is from a test furnished by the Pelton Water Wheel Company on a turbine built by that company for the Black Hills Traction Company at Spearfish, S. Dak. The test showed a maximum efficiency of 90.8%, as read from the curve. The wheel is of the "Pelton-Francis" type, with volute casing and single runner, and was tested in place by the engineers of the companies.

Point *C* is from the Holyoke test of the 33-in., I. P. Morris Company wheel previously mentioned. This wheel showed an efficiency of 88.49%, with 17.51-ft. head, 184.57 rev. per min., and 142.18 h.p., the corresponding specific speed being 274.

Other points on the curve are from Mr. Larner's paper, and are indicated by the numbers used by him.

The only other turbine known to the writer which, in efficiency, approaches very closely the values given on the curve, is a French wheel built by the *Société des Établissements Singrun*, at Epinal, for the French Government, the efficiency of which is reported to be 90.4 per cent. The data for calculating the specific speed have not been ascertained. Mr. Moody.

Many excellent results not falling on the curve, but showing very creditable efficiencies and speeds, are shown in Table 9.

TABLE 9.

Number of point.	Turbine.	Builder.	Place of test.	Efficiency, Percentage.	Specific speed, N_s.
20	Hercules 36-in	Holyoke Machine Co	Holyoke Flume	86.17	280
21	" 36-in	" " "	" "	86.51	285
23	" 30-in	" " "	" "	87.07	287
25	" 30-in	" " "	" "	86.50	314
31	Samson 56-in	James Leffel & Co	" "	85.01	289
27				83.58	336
36	" 50-in	" " " "	" "	83.59	291
..	" 45-in	" " " "	" "	84.55	310
..	" 40-in	" " " "	" "	84.72	307
..	" 35-in	" " " "	" "	84.78	303
5	Pelton-Francis	Pelton Water Wheel Co	In place, Sultepec	85.0	72.5
10	" "	" " " "	In place, Schaghticoke	86.5	174
33	Type E-30-in	Allis-Chalmers Co	Holyoke	86.51	250
34	Type F-30-in	" " "	"	82.31	321
..				82.5	334
35				81.5	343
				78.89	385
8	10 500-h.p. Shawinigan	I. P. Morris Co.	In place	87.2	125
3	Samson Niagara Type	James Leffel & Co	" "	85.6	54.5
4	6 100-h.p. at Hamilton Cat. Co.	J. M. Voith, Heidenheim	" "	86.8	66.5*
9	At Rheinfelden	Escher Wyss & Co	" "	87.4	156.5
15	At Manegg-Zürcher Papierfabrik	" " " "	" "	88.0	229
28	For A. Giraud & Cie., Lyon	" " " "	" "	86.8	109.4
32	Experimental at Sagan	Designed by Dr. Camerer.	In Germany, Experimental Flume	82.8	288
13	Swain	Swain Turbine & Mfg. Co.	Holyoke	85.89	213†
17	McCormick 30-in	S. Morgan Smith Co	"	85.78	260†
19	McCormick 51-in	J. & W. Jolly	"	86.84	278†
18	McCormick 33-in	S. Morgan Smith Co	"	86.3	277†

NOTE.—Point 11, at 196 specific speed, is Professor Thurston's test previously referred to.
* From "Hydro-electric Developments and Engineering," by F. Koester.
† From "Water Power Engineering," By D. W. Mead, M. Am. Soc. C. E.

In conclusion, the writer wishes again to express his appreciation of Mr. Larner's valuable paper; and to extend thanks, for their courtesy in furnishing information, to the following: Mr. H. Birchard Taylor, Assistant Hydraulic Engineer, I. P. Morris Company; Mr. J. E. Strothman, Assistant Manager, Pumping Engine and Hydraulic Turbine Department, Allis-Chalmers Company; The Holyoke Machine Company, Mr. W. W. White, Manager; The Pelton Water Wheel Company, Mr. J. V. Kunze, Manager; James Leffel and Company; and The Platt Iron Works Company, Mr. John Sturgess, Western Representative. It is regretted that lack of time has prevented the writer

Mr. Moody. from awaiting the data which are being sent by the Platt Iron Works Company.

Mr. Parker. John C. Parker, Assoc. M. Am. Soc. C. E.—In this paper Mr. Larner has contributed so much that is of signal technical interest that one may lose sight of some of his general remarks. The speaker would like to call attention to a very significant paragraph on page 344, in which the author deals with the advisability of sacrificing, in some cases, the fractional-load efficiency of wheels for the sake of the economical advantage in initial cost of installation. This is of some interest, but the speaker thinks that the general principle laid down should be viewed with caution. As the statement stands, it is undeniably correct, as applied to developments such as those at Niagara Falls, where the maximum quantity of water that may be diverted is limited by law; and also where the quantity of water used at fractional loads is of no importance, being well below that which may be diverted.

It also applies well to those cases where the development is not greater than the minimum flow of a stream. In most installations, however, it will be found economical to over-develop the minimum flow, and to supply a steam reserve to take care of the periods of deficiency in the natural flow. In such cases the steam reserve will stand idle for the major part of the year. It becomes desirable, therefore, to minimize the fixed cost by installing a cheap steam plant (necessarily an inefficient steam plant). The wheels would operate at times under reduced flow, and, unless there is a multiplicity of units, enabling all the units in service at any time to be operated nearly at their full load, they will be worked uneconomically, because the plant is an uneconomical low-load plant. A considerable quantity of coal will be burned, under these conditions. Therefore, in a plant where the minimum flow is less than that required to carry the minimum load which may be thrown on the plant, it becomes desirable to look to the fractional-load efficiencies of the wheels.

Another comment by Mr. Larner, to which the speaker would like to call attention, refers to tests. The data contributed by the author are of so much interest and value that it would be a pity to overlook the facility with which tests on units connected to electrical generators may be made at various speeds, especially of wheels which are not amenable to such tests as may be made at the Holyoke flume.

There is not much mystery connected with the losses in electrical generators. Losses in the power going to waste in the conductors of the armatures of the machines are readily calculated by the application of Ohm's law to readings taken during the operation. The friction and windage in the machines are a purely mechanical proposition, and can be determined by tests in the field. The only other losses are those in the iron in the generators. These losses are a very simple speed function, and can also be readily determined. As the total losses will not be more than from 2 to 4%, even in the larger generators, a slight inaccuracy in determining their variations with the speed is not very serious. Mr. Parker.

The readiest method known to the speaker for determining these losses as a speed function is by using what is known as a deceleration curve. With the water shut off from the turbine, the machine is brought up to speed by some outside source—possibly electrical—by interconnecting the generators on two wheels, one of which can take water; bringing them up to more than their running speed, and then cutting loose that wheel in which the electrical generator losses are to be determined.

It is then possible to determine losses by the electrical meters and by observing the speed of the wheel as a time function. Knowing the moment of inertia of the rotating parts—which can be readily calculated—observing the negative time differential of the speed, and multiplying by the moment of inertia, one can readily plot the power losses as a time function.

The moment of inertia can be determined experimentally in one of two ways. In a small machine, the addition of a known and readily determinable weight gives a ready solution. Its influence upon the deceleration curve will give the moment of inertia of the masses of the rotating parts under normal conditions.

Another method which the speaker has intended to try (but has not yet succeeded in doing because of lack of time) consists in using an electrical braking device. The simple connection of the electrical generator through a known resistance, and the measurement of the current carried by it, will show what the power usefully generated does in the way of braking the machine. Determining this moment of inertia and taking the whole group of curves should not take more than half a day, and therefore it would be easy to obtain the electrical losses as a speed function, and thereby extend to large units the very valuable study that Mr. Larner has made with such units as are capable of being tested in the Holyoke flume. The speaker believes that engineers who have facilities should look into the matter and see if Mr. Larner's data cannot be extended to such wheels.

Another of Mr. Larner's facts is of great interest, and the speaker is very glad that he has put himself on record as being opposed to the radical application of "standardization," a shibboleth which has been one of the most baneful things in practice, namely, that of taking "goods off the shelf"—the selling of stock goods for any and every purpose.

In building up American enterprises, standardization has been very helpful; but the speaker thinks that those who have had anything to do with the purchase of apparatus and who have attempted its intelligent

Mr. Parker. application, have met, in all lines of manufacturing, with the unfortunate use of "standards" to prevent progress.

The attitude taken by Mr. Larner—who is connected with the manufacturing industries—is certainly most hopeful, and gives promise of some advance in American practice toward standards which heretofore have prevailed almost exclusively in Europe.

Mr. Allen. C. M. Allen, Assoc. M. Am. Soc. C. E. (by letter).—This paper is of great interest to the writer, because he is especially concerned in the behavior of water-wheels, not only at the Holyoke testing flume, but particularly after installation. The hydraulic turbine is by far the most efficient of any of the prime movers, and when one considers that to-day it is possible to obtain an efficiency of 90%, as shown by these tests, it certainly seems fitting that the turbine, including the setting, should receive as much intelligent and scientific attention as the steam engine, the steam turbine, or the gas engine.

During the past ten years, the writer has tested water-wheels after installation, for both horse-power and efficiency, using an Alden absorption dynamometer to weigh the load accurately, measuring the water sometimes with a standard weir and sometimes with a current meter. Tests have been made on wheels varying in horse-power from 50 to more than 4 000, acting under heads ranging from 10 to 200 ft., and running at speeds of from 75 to 800 rev. per min., and the writer is fully convinced that turbines can show as good an efficiency after installation as it was possible to show at the Holyoke testing flume, and that, in some cases, a better efficiency may be obtained. This is especially true in the behavior of high-head turbines; one runner, tested at the Holyoke flume, showed an efficiency of 83 or 84% at its best gate-opening and speed, and when it was tested after installation, under a 200-ft. head, it showed an efficiency of more than 86% in several instances. The load was measured very carefully with an Alden dynamometer, and the water was measured over a standard weir. A great many turbines composed of a pair of wheels have also been tested under heads varying from 20 to 50 ft., where the actual horse-power developed after installation was practically what the Holyoke tests would show when corrected for new head and speed conditions, according to the rule that, if the velocity ratio remains constant, the horse-power varies with the third power of the square root of the head.

There are also many wheels which do not come up to the rated power, as calculated from the Holyoke tests. In order to have wheels give their proper rating "in the field," it is necessary to give them good settings and to run them at the proper speeds. A good setting is a most important item, and means: That the water should be brought to the wheel with a low velocity; the draft-tube should be air-tight, and should be designed to take the water away from the individual runners with as little conflict as possible; in order to avoid obstruction around the center of the buckets, the shaft should not be larger than necessary; the wheels—if a pair, and center-discharging—should not be set so closely together that the discharge of one interferes with that of the other; the bearings should be kept in line, etc. Mr. Allen.

Brake testing of turbines after installation is as essential to the designer of turbine settings, as the testing of runners at the Holyoke flume is to the designer of turbine runners. Many a good wheel has been cramped and mistreated in its setting, and, therefore, will never give its legitimate power and efficiency.

A great deal of information is still needed concerning the behavior of turbines installed in various settings, and under medium and high heads. If the efficiency of the plant as a whole is to be put on as high a plane as possible, more engineering effort should be exerted in securing efficient settings, and at the present time there is more chance in this way to increase the efficiency of the whole plant several per cent. than in any other.

E. Kuichling, M. Am. Soc. C. E.—The author has presented in excellent manner a number of interesting features on the construction and performance of hydraulic turbines, most of which are inadequately discussed in textbooks and rarely found in manufacturers' publications. Occasionally, the report of some favorable test of a particular style of wheel is encountered, but the data are usually insufficient to enable one to determine satisfactorily the efficiency of the motor under varying conditions of speed and discharge. In this paper, however, Mr. Larner submits all data pertaining to a wide range of tests, made at Holyoke, of five turbines designed by him for the Wellman-Seaver-Morgan Company; he then shows how to compare properly the results attained for the several wheels, and, in conclusion, gives numerous valuable suggestions relating to the choice of turbines for different heads. Mr. Kuichling.

This contribution to the literature of American water-wheels is very welcome, even though it omits a consideration of the details of design on which satisfactory performance must depend. The author takes the ground that a manufacturer is justified in withholding from the public a thorough knowledge of the points of superiority in design, which he may be able to devise; but, in opposition thereto, it may be urged with much force that such a course tends to foster doubt as to the real merit of the invention, and to retard its general use. To the speaker it seems entirely reasonable that a purchaser should have a correct knowledge of the physical and mathematical principles on which a successful invention is based, in order that he may be able to select the best design or type for his purpose; and it usually happens that if such information is deliberately concealed, a more or less adequate empirical criterion will be developed, in course of time, by which a choice can readily be made. In general, the manufacturers

Mr. Kuichling.

who display the most thorough knowledge of the principles of mechanical design obtain the largest share of business.

Many features must be taken into consideration, in finally selecting a turbine for a particular locality and service, but only a few are involved at the outset. The quantity of water and the head available at different seasons of the year are known, approximately, as well as the desirable magnitude of the power units, but when it comes to determining the most efficient size, discharge, and speed of the turbines, much variation in practice is found. This, without doubt, is due to the lack of exact knowledge of the relations between the diameter, depth, discharge, speed, and power of American wheels, in which the buckets are usually of such complex form as to render a mathematical analysis of the motion of the water through them impracticable. It is necessary, therefore, to depend in large degree on the unpublished experimental results acquired by the wheel maker, and to exact of him a guaranty that his turbine will perform the required work at the stipulated efficiency. In the speaker's opinion, this condition is unsound, and should be replaced by a more extensive distribution of experimental data and the formulation of reliable principles of design.

The author has referred to his experiments for determining the velocity of the water at the top and bottom of the guides and in the draft-tube. Much difference of opinion as to the motion of the water in these places has been expressed by writers, and Mr. Larner, therefore, could increase the value of his paper greatly by adding some of his observations in these respects, such as the manner in which the velocity is distributed in a draft-tube of large diameter when the wheel is running at partial gate and full gate. It has been claimed by some engineers that a draft-tube cannot be made too large, while others advise the restriction of its diameter to a velocity of 10 ft. per sec. at full gate. Similarly, with the guides to the turbine buckets, some claim that these adjuncts are wholly unnecessary, while others attach great importance thereto. Experimental data in these directions are rarely encountered, and their presentation will doubtless be highly appreciated by all who are interested in hydraulics.

There are other obscure matters which Mr. Larner can doubtless elucidate, for example: Should there be any difference between the efficiencies of a right-hand and a left-hand turbine of the same type and size, under equal head and discharge? Is the power and efficiency of a turbine the same when it is set with its axis horizontal as it is when set with its axis vertical, it being assumed in each case that the surface of the head-water is far enough above the wheel to prevent any entrainment of air, that the head and discharges are equal, and that the wheel is similarly mounted, either in a casing or in a capacious open flume, and with like bearings and draft-tube?

DANIEL W. MEAD, M. AM. SOC. C. E. (by letter).—This paper is a valuable contribution to the literature of the hydraulic turbine, and deserves the careful consideration of every engineer interested in the development of water-powers. Mr. Mead.

As the author states, there is still room for vast improvement in the design and manufacture of the turbine. There may be little chance for securing greater efficiencies than are shown by some of the tests recorded in the paper, but such results are seldom obtained in practice under the actual operating conditions of speed and load. In other words, a well-designed and well-constructed turbine under the best test conditions will frequently give high efficiencies; but it is much more difficult to design a wheel so that such efficiencies will be secured under the fixed conditions of speed and load in the actual installation.

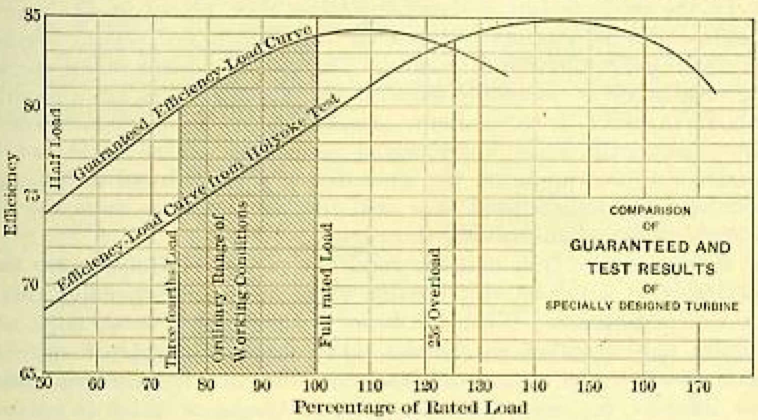


FIG. 20.

From the writer's observations, it is safe to say that the design and manufacture of water-wheels has not yet reached a stage where, without considerable experimental work, a wheel can be designed and constructed which will unquestionably secure the desired results under actual operating conditions. The only apparent exception to this is secured by the adaptation of wheels already experimentally developed, or "standardized," to conditions to the necessities of which their characteristics closely approximate.

This difficulty is well illustrated in Fig. 20, in which the guaranties for a recent installation are compared with the actual test results secured by the manufacturer with the first wheel especially designed and constructed for the fixed conditions. The maximum efficiencies secured on the test were above the efficiencies guaranteed, but the power of the wheel was too great, and therefore the efficiencies under the conditions of operation were deficient.

Mr. Mead. In the writer's opinion, the development of numerous experimentally perfected designs, or what are termed by many engineers "standard wheels," or "stock wheels," is of decided advantage, for it is found that by care in manufacture a successful wheel may be duplicated, enlarged, or reduced in size, without any great changes in its characteristics. With such wheels already developed, it remains for the water-power engineer only to see that the wheel selected is of proper size and character for the conditions under which it is to operate, and the results that can be attained are reasonably dependable.

The results attained in water-power installations are seldom better than the demand, and the carelessness or ignorance of the engineers in selecting turbines is as much a fault in this matter as the poor design or construction of the manufacturer.

In the past, too much dependence has been placed on the manufacturer and in guaranties made by him which are seldom verified by actual determination of results. The test of a plant under operating conditions is commonly exceedingly expensive, and often so difficult as to make it inexpedient. Knowing this, some manufacturers have frequently made unwarranted guaranties which could not be carried out—depending on the ignorance of the purchaser or his engineer, and the unlikelihood of a final test, to free them from the disastrous results of an unfulfilled contract.

It is perfectly true that the results of a test at Holyoke are only applicable to the same or a similar wheel under corresponding conditions of installation. When the wheel is equally well constructed, and equally well installed, the results that can be attained will approximate closely to the test results; and when the conditions of installation are different, the test results afford the best basis on which to estimate the results which may reasonably be anticipated.

Good mechanical work in water-wheel construction can be secured by close inspection, as in the case of any other mechanical work. The engineer, then, must select the proper wheel to fulfill the conditions of his installation and see that the mechanical details are properly executed.

In another place (see the author's reference at the foot of page 323), the writer has pointed out certain characteristic coefficients of water-wheels, and has discussed their use in the preliminary consideration of the selection of a water-wheel for a given installation. The writer's K_5 in that discussion corresponds to the square of the author's N_s, and is used in that way in order to facilitate calculation. In this discussion, however, the author's nomenclature is adopted so as to avoid confusion.

While, in a general way, the value of the "unit speed" of various wheels will give a ready means for their approximate comparison, it must be remembered that all wheels having the same value of N_u are not necessarily of similar value in a given installation. Mr. Mead.

N_u is made up of two factors, namely, the "unit speed" under the "unit head" (1 ft.) $= \dfrac{N}{\sqrt{h}} = N'$ and the square root of the "unit power" under the "unit head," $\sqrt{P'} = \sqrt{\dfrac{P}{h^{\frac{3}{2}}}}$,

$$\text{that is, } N_u = N' \sqrt{P'} = \frac{N}{h^{\frac{1}{2}}} \sqrt{\frac{P}{h^{\frac{3}{2}}}}.$$

The value of N' enters directly into this value, while P' enters only as its square root. The value of N', therefore, has much the greater influence on the character of the wheel.

In a recent installation it was desired to develop about 650 h.p. with a unit of two turbines under a 16-ft. head at 128 rev. per min. Therefore, it was required that the following relations should obtain approximately for each turbine:

$$
\begin{aligned}
h &= 16, \\
N &= 128, \\
N' &= 32.1, \\
P &= \frac{650}{2} = 325, \\
P' &= 5.07, \\
N_u &= N' \sqrt{P'} = 32.5 \sqrt{5.07} = 72.4.
\end{aligned}
$$

The turbines on which bids were submitted had the following characteristics:

$$
\begin{aligned}
&\text{Required } N_u = 32.5 \sqrt{5.07} = 72.4, \\
&\text{No. 1} \ldots \; N_u = 32 \sqrt{4.85} = 70, \\
&\text{No. 2} \ldots \; N_u = 30.2 \sqrt{5.2} = 70, \\
&\text{No. 3} \ldots \; N_u = 37.5 \sqrt{5.02} = 85, \\
&\text{No. 4} \ldots \; N_u = 35.7 \sqrt{4} = 71.4, \\
&\text{No. 5} \ldots \; N_u = 32.1 \sqrt{5.2} = 74.7.
\end{aligned}
$$

All these turbines, with the exception of No. 3, closely approximated the required value of N_u. No. 4, which most closely approximated the true value, had too high a value of N' and too low a value of P', and, therefore, was not suited to the conditions. No. 5 had individual characteristics which rendered it unsuitable. Nos. 1 and 2 most closely approximated the desired results, but No. 1, while slightly deficient in power, was highly efficient, and was selected as the most desirable wheel offered. (See Figs. 26, 27, and 28.)

The comments made above, however, could be based only in the most general way on the values of N_u, for it must be understood that the

Mr. Mead. values of N_s, or other turbine coefficients (as discussed in the previous reference), represent only the best results under one condition, that is, the point of highest efficiency at full or seven-eighths gate, as the case may be.

The engineer, while interested in the point of maximum efficiency, usually has to consider various other conditions of power (and hence of gate), and often variations in head must be taken into account as well. The engineer, therefore, must make a much broader analysis of operating conditions. For this purpose, a graphical analysis of an actual test has been found by the writer to be apparently the only method at once accurate and rapid by which the required information can be obtained. Two methods for such analysis have been pointed out by the writer in the discussion already referred to. One of these methods, and perhaps the simplest form, is that used by the author in this paper. Using either of these methods, the engineer can readily determine the results which he can expect to secure from a wheel in which the design is homologous to one for which test results are available. The results of actual analysis will perhaps best serve to show the desirability of such analysis and its necessity, if the best results are to be attained.

The tendency of manufacturers is always to furnish wheels of more than ample power, usually at a sacrifice of efficiency (see Fig. 20). This tendency is encouraged by the fact that capacity is readily determined, while the determination of efficiency is more difficult and frequently impossible. To guard against this tendency, analysis is necessary, and the specifications under which bids are invited should be clear and specific. In a contract let some time ago, the writer prepared, among others, the following specifications:

"Each unit shall consist of a pair of turbines in tandem, and shall be capable of developing a maximum of 1 900 actual horse-power under a working head of 70 ft. when running at 375 rev. per min. These turbines are to be designed to operate satisfactorily under a maximum head of 80 ft., and a minimum head of 65 ft. if so required.

"The contractor shall furnish a Holyoke test sheet of a turbine of homogeneous design to that he proposes to furnish. Said test should be of a turbine of similar size to the turbines on which proposals are made, but, if such is not available, it may be on the nearest size available."

On these specifications seven bids were submitted, and the guaranties shown in Fig. 21 were made. Two of the bidders failed to submit test data, and their bids were not considered. An analysis of the test data (Fig. 22) showed that the results which could be actually attained by the wheels offered were, according to the tests, in every case less than the guaranty. The wheels were in all cases too large for the best results, as is clearly apparent from the diagram (except in case of wheel No. 1), and the points of maximum efficiency were, in most cases, far beyond the ordinary range of operation. Therefore, all bids were rejected, and the matter was taken up directly with the manufacturers of wheel No. 1 which came nearest to suiting the conditions of operation. A graphical analysis of test results of wheel No. 1 as originally submitted is shown in Fig. 23, from which it will be seen that this wheel was too large to give satisfactory results under the operating condition and was not of a satisfactory type. It was apparent that a wheel of less diameter and of a slightly different type would fill, Mr. Mead.

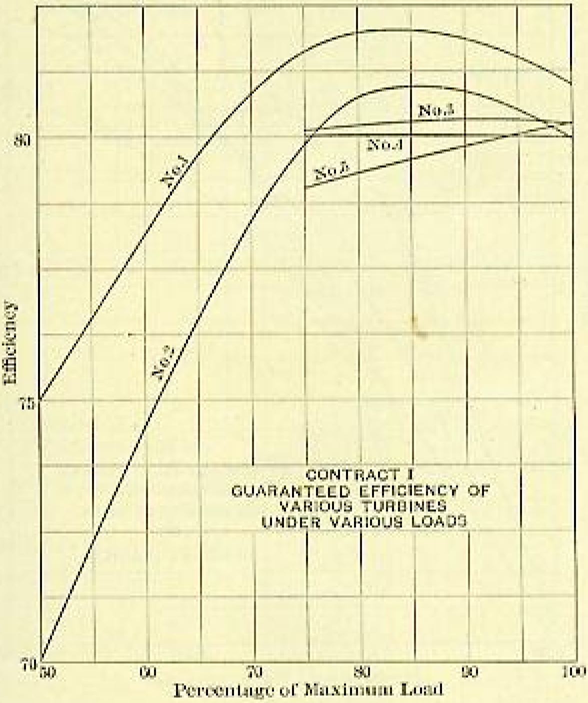


FIG. 24.

much more nearly, the conditions of operation, and a contract was entered into whereby the efficiencies shown in Fig. 24 were guaranteed. The results were to be demonstrated by a Holyoke test and by a test in place, if such was desired. The Holyoke test was to be made at the expense of the manufacturer, in the presence of the engineer of the purchaser; and the test in place, if made, was to be made at the expense of the purchaser.

It may be remarked that final surveys, completed before the contract was let, developed the fact that a maximum head of 85 ft. could

Mr. Mead. be obtained. It also became apparent that, to secure the best results at all heads, less power than specified at the 70-ft. head was desirable.

With the wheels selected, four units at full load and maximum head will give the designed capacity of the plant, and five units (which includes a reserve unit) will give the full capacity of the plant under the minimum head, which, however, seldom obtains.

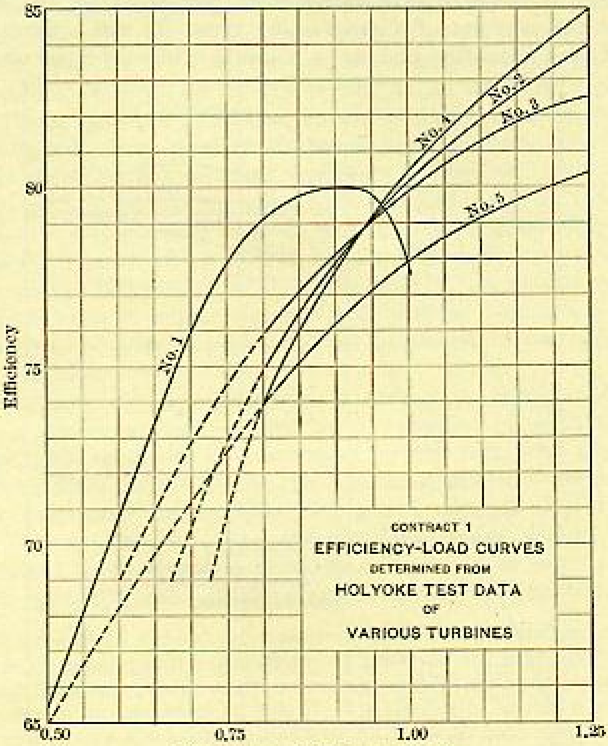


FIG. 22.

It was evident from the bids received that the specifications were not sufficiently explicit, and, while the desired results were secured before the final award, the specifications were clearly faulty.

In a more recent installation, an attempt was made to remedy the defect in the foregoing specifications, and the clauses referring to conditions of operation were prepared as follows:

"Bids are desired on three units of two turbines each. Each unit shall consist of one pair of turbines mounted on a horizontal shaft. These turbines will be installed in open penstocks, and shall be capable of developing a maximum of about 650 actual horse-power under a

Mr. Mead.

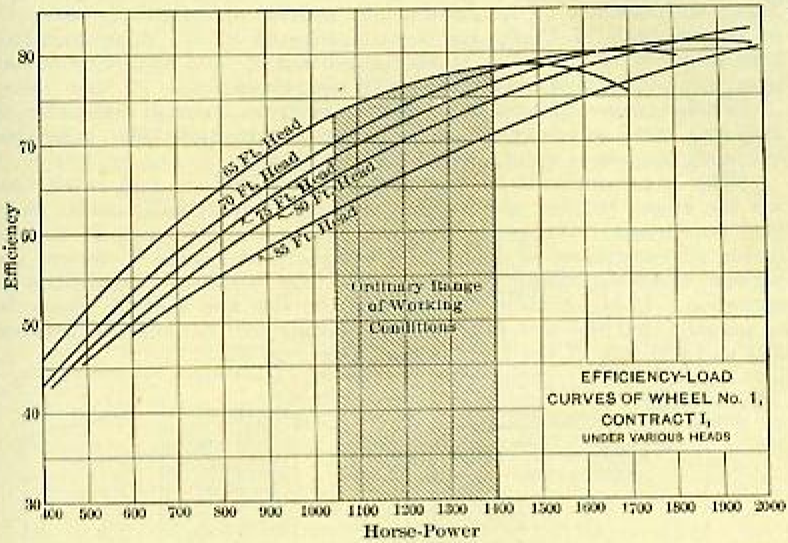


FIG. 23.

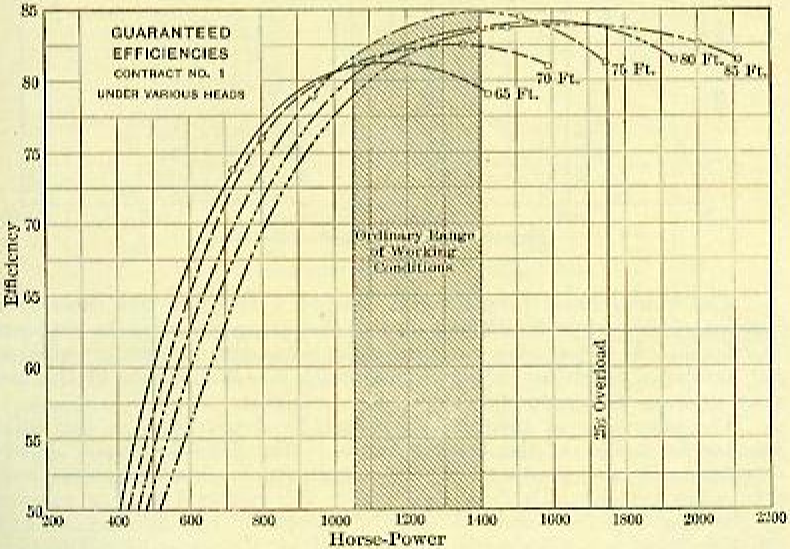


FIG. 24.

Mr. Mead. working head of 16 ft., and running at approximately 128 rev. per min. These turbines are to be designed to operate satisfactorily under a maximum head of 17 ft. and a minimum of 12 ft., if so required. The efficiency of the units at each condition of load shall be not less than that guaranteed by the bidder in his proposal.

"Each turbine unit is to operate a 60-cycle, 3-phase alternator of 360 kw. rated capacity at 90% power factor and 94% generator efficiency, and to a maximum of 25% over-load.

"The generator will be operated at from 75% to full rated load for the larger portion of the time, and only occasionally under overload conditions. While the highest practicable efficiency is desired under all conditions of load, it is especially desirable to secure the highest efficiency under the most usual and continued conditions of operation. It is therefore desired that the size and type of wheels be so selected that the best practicable efficiency will be obtained between 390 and 520 h.p. of the turbine load (see Fig. 25).

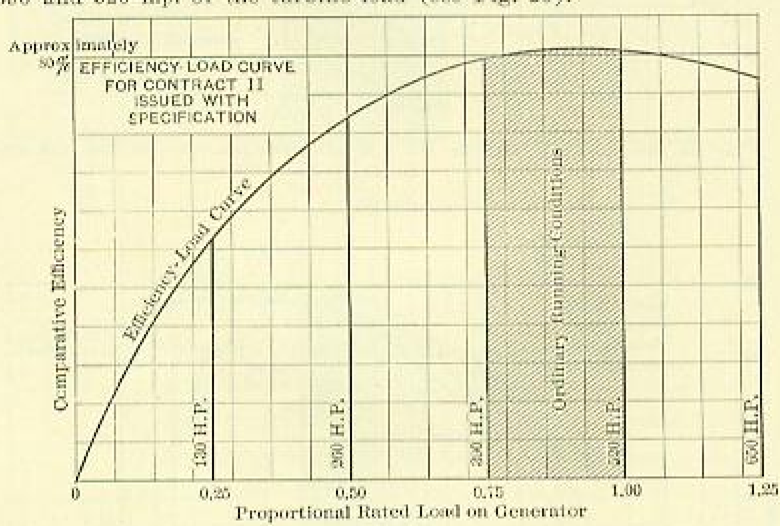


Fig. 25.

"The bidder must furnish with his bid a Holyoke test sheet of a turbine of homogeneous design and of the same size that he proposes to furnish. If a test of a turbine of the same size as that proposed is not available, then the bidder may furnish two test sheets of the two sizes nearest that proposed, which are available.

"The efficiency of turbine units shall be determined in the usual manner by a test at the testing flume of the Holyoke Water Power Company, in the presence of the Engineer, or his representatives. One main turbine unit furnished under this contract will be selected by the Engineer, and shall be carefully tested to his satisfaction, and the efficiency shown by these wheels will be the basis of calculating the efficiency of the unit.

"These tests shall be made at the expense of the contractor.

"If, after the Holyoke test is made and found satisfactory, and, after the turbines are erected, the turbine units fail to develop the power specified, or are found to use too much water for the amount of power generated, then, and in that event, any or all of the turbine units may be tested in place. The test shall be made by the Chief Engineer of the —— Power Company, or by his authorized representative, and the turbine manufacturers shall be notified and shall have the opportunity of being present and taking part in said test. Mr. Mead.

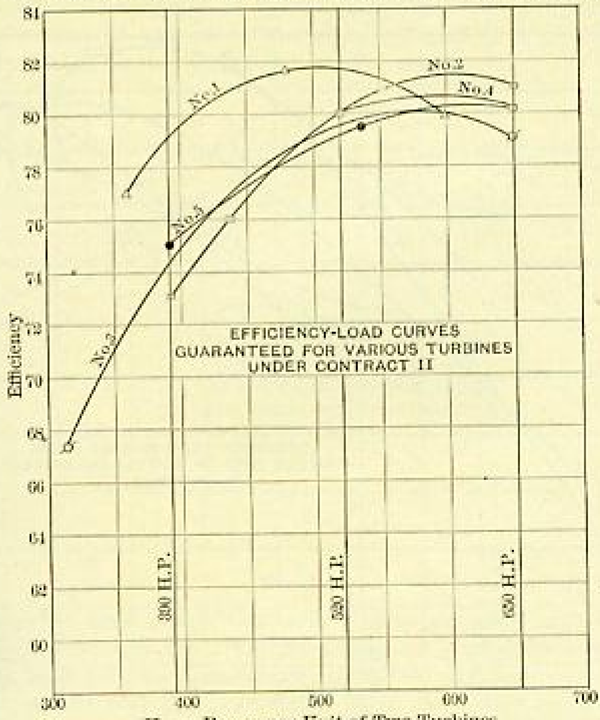


FIG. 26.

"The methods of making such tests shall be by means of recognized standard methods of measurement of water, power, and speed, and will be agreed upon, as far as practicable, by the parties hereto. If any disagreement arises, the decision of the Chief Engineer shall be final. The test in place will be made at the cost and expense of the —— Power Company. Any turbines found defective by the test at Holyoke, or by the test in place, shall warrant the rejection of such turbines."

The diagram accompanying these specifications is shown in Fig. 25. The guaranties submitted with the bids are shown in Fig. 26, and the analytical results of the test sheets are shown in Fig. 27. It will be noted from Figs. 26 and 27 that in two cases the guaranties were

Mr. Mead. higher than shown as possible by the test results. The contract was awarded to the manufacturer of wheel No. 1, as the lowest and best bid. A comparison of the guaranty and test results of this manufacturer is given in Fig. 28, which shows that the guaranty is conservative and will probably be exceeded if the mechanical workmanship of the wheel is carefully executed. It may be noted that the above wheels are the same as those used in the discussion of the value of N_s.

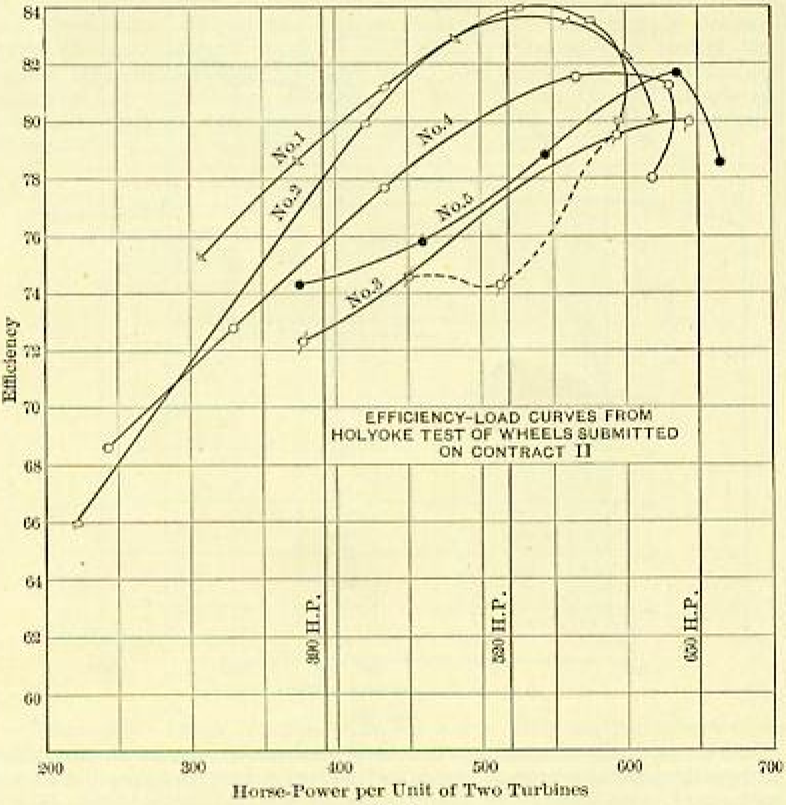


Fig. 27.

The latter form of specification is believed to be in general satisfactory. The Holyoke test specified is somewhat expensive, and its cost will usually be added by the manufacturer to the price bid. The same results may often be attained, without the actual expense of a test, by requiring a test of the completed wheel at a fixed price, provided the engineer so elects. This will assure care in manufacture, and the expense may be eliminated by the engineer if on inspection of the completed wheel he believes its mechanical construction is equal to that of similar wheels that have already been tested. Mr. Mead.

Close analysis and rigid and intelligent inspection will assure a higher grade of design and workmanship than is commonly offered for water-power installations. The engineer who neglects these precautions and blindly follows the recommendations and unverified guaranties of manufacturers must expect to secure such unsatisfactory results as the contingencies of close competition and the temptation of the possession of standards already developed (but probably not strictly applicable to the condition) always create.

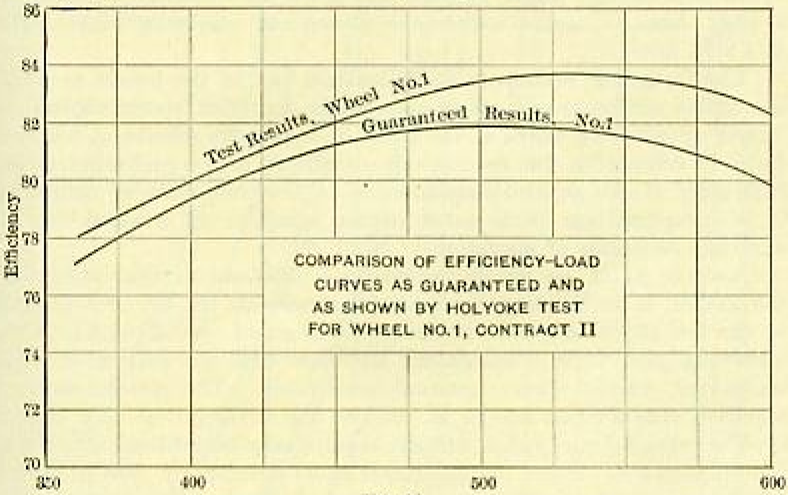


Fig. 28.

In many low-head plants the variations in the head under various conditions of river flow are often considerable. During low-water conditions the head may be a maximum, but with high water, on account of flowage conditions, the head-water must commonly be kept down by the use of a wide spillway section or flood-gates, while the tail-water will rise and hence the head will be reduced. When (without auxiliary power) it is necessary to run the plant under all these conditions, the detailed analysis based on test results of the wheels offered is the only basis from which their probable power and efficiency under the extreme range of conditions can be determined. Take, for example, the author's diagram, Fig. 12, of the test of a 31-in. wheel, and consider its use under heads varying from 16 to 9 ft. The relative unit speeds under the maximum and minimum head will vary as $\sqrt{h}$, and will be as 4 is to 3. If the wheel is arranged to run at 162 rev. per min., the values of N' will be 40.5 for a 16-ft. head and 54 for a 9-ft. head, which, at both maximum and minimum heads, will result in less than

Mr. Mead. the best efficiencies. The actual efficiencies and power under the full range of conditions and for all intervening heads, can readily be calculated from the graphical diagram.

It is to be noted that as high efficiency (and consequently economy of water) is more essential at low water (and high-head conditions), a better speed for the above wheel might be 176 rev. per min., under which conditions the value of N' would be:

$$N' = 44 \quad \text{for a 16-ft. head,}$$
$$N' = 58\tfrac{2}{3} \quad \text{for a 9-ft. head.}$$

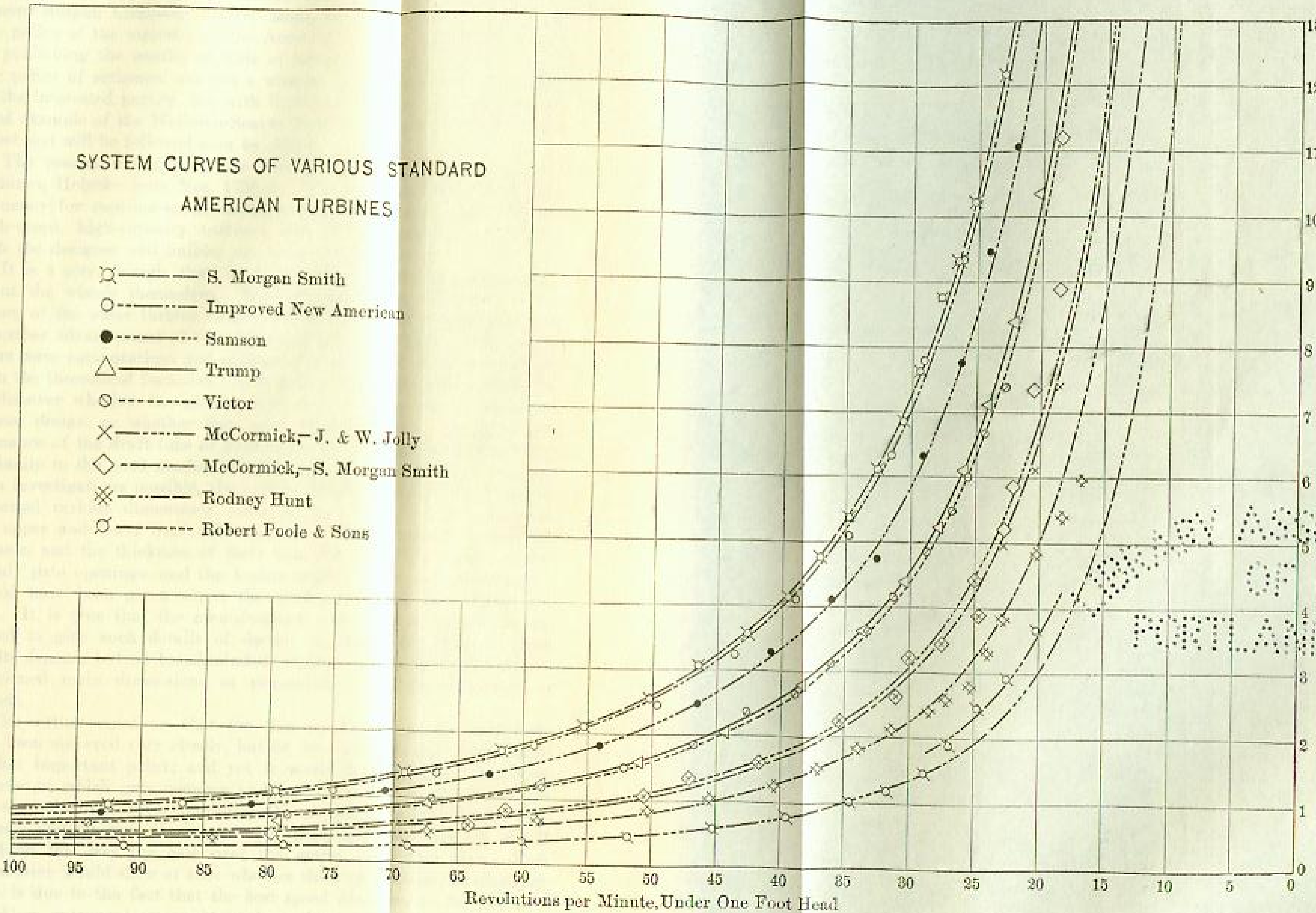
While this change would increase the power and efficiency slightly at a 16-ft. head, it would reduce the power and efficiency considerably at a 9-ft. head.

The foregoing comments will illustrate one of the points to which the writer wishes to call attention, that is, that the "specific speed" or "unity speed" of a wheel at the point of maximum efficiency, while of value in comparing test results and sometimes in the preliminary consideration of the general requirements of the conditions of operation, is of comparatively little value in the selection of a wheel for the varying conditions of operation.

Another point, on which the writer is inclined to take issue with the author, is on the "standardization" of wheels. In the present state of the art, the writer believes that the design of special wheels is too uncertain and their development too slow and too expensive to be warranted, except under unusual conditions. The development of standard construction means at its best the development of a line or type of wheels from which definite results may be realized quickly at limited expense. Every manufacturer must develop, by careful design and expensive experiments, lines or types of wheels which become valuable for broad application only as they are sufficiently standardized to assure the prompt fulfillment of definite results. Standard wheels already on the market differ widely in their characteristics. No one type of wheel is suited to every condition, but the wide variation offered by the several makers affords a range from which the engineer may readily select one or more wheels that will often fulfill in a satisfactory manner the conditions of his installation. The range of the standard wheels of various manufacturers is shown on Plate XXXIII. This diagram is computed from the catalogue recommendations of the manufacturers, which are approximately, but not always exactly, correct.

The writer believes that standardization is objectionable only when attempts are made to force the conditions to conform to the standards, instead of selecting or altering the standards to conform to the conditions.

Satisfactory results can be secured only by the ability of the purchaser's engineer to analyze the situation and the basic tests on which any intelligent bid of manufacturers must be based.


SYSTEM CURVES OF VARIOUS STANDARD
AMERICAN TURBINES
S. Morgan Smith
Improved New American
Samson
Trump
Victor
McCormick,—J. & W. Jolly
McCormick,—S. Morgan Smith
Rodney Hunt
Robert Poole & Sons
Revolutions per Minute, Under One Foot Head
Horse-Power, Under One Foot Head

January 1939

S. J. Zowski,* Esq. (by letter).—Mr. Larner and the Wellman-Seaver-Morgan Company deserve much credit for having abandoned the policy of the majority of the American turbine builders in regard to publishing the results of tests of newly developed turbines. That the policy of reticence was not a wise one was pointed out frequently to the interested parties, but with little success. Let us hope that the good example of the Wellman-Seaver-Morgan Company will have more effect and will be followed soon by others. Mr. Zowski.

The results obtained by the Wellman-Seaver-Morgan Company's turbines, Holyoke tests Nos. 1795 to 1800, are remarkable: 90% best efficiency for medium-speed, medium-capacity turbines, and 86% for high-speed, high-capacity turbines, are record-breaking values, and both the designer and builder are to be congratulated.

It is a pity, though, that the author did not give any information about the wheels themselves. Many hydraulicians interested in the theory of the water turbine would like to utilize these experiments for a further advancement of the science of this art. They would like to make some computations and compare the results of these experiments with the theoretical formulas. It is quite natural that they would like to discover whether the good results were obtained by an improved runner design, or whether they must be attributed to the good performance of the draft-tube or guide-case, or to the good workmanship, or finally to the good mechanical efficiency of the machine. To make such investigations possible, the author should have given at least the principal turbine dimensions, such as the height of the guide-case, the upper and lower draft-tube diameters, the number of vanes and buckets, and the thickness of their tips, the actual (not only proportional) gate openings, and the bucket angles. Also, full information should have been given about the workmanship of the runner and case. It is true that the manufacturer could not reasonably be expected to give such details of design on which the obtained good results depend, but no broad-minded engineer will consider the above-mentioned main dimensions as responsible for a given failure or success.

The author mentions that the flow conditions in the draft-tube have been observed very closely, but he does not give any information on that important point; and yet it would be especially interesting to know at which gate opening and speed the flow in the draft-tube was not whirling, thus obtaining the so-called "perpendicular discharge," and whether there was any speed and gate opening at all at which this was the case over the entire draft-tube area. This information would show at once whether the high peak in the efficiency curve is due to the fact that the best speed diagrams at the entrance (shockless entrance) were obtained at the same time at which the

* Professor, University of Michigan.

Mr. Zowski. best speed diagrams at the discharge (perpendicular discharge) were prevailing. It would also show whether any changes in these speed diagrams would raise the peak still more, or whether they would flatten the curve for the benefit of other gate openings, and it would indicate in which direction changes in the bucket angles and curvature should be made for a desired effect.

It is thought that in all publications of turbine tests which are made not solely for commercial reasons, the following rule should be adopted: That, in addition to the test sheet there should be given a sketch of the turbine with all main dimensions, and that the nature of the flow at the different gate openings and speeds should be indicated clearly.

All European turbine tests, for instance, those of Professor E. Reichel, of Charlottenburg, and in general all other experiments of scientific value, are reported in such a way. Why should an exception be made with American water-turbine tests? Undoubtedly, the observation of this rule would help to eliminate the controversies which exist between American and European turbine designers, and it would also help to eliminate all mutual accusations as to the reliability of the reports on the one side, and lack of knowledge and success on the other.

In Fig. 1, representing the high-speed turbine, it will be noticed that the water flowing along the lower gate ring must make quite a sudden turn in entering the runner. This, in connection with the high efficiency obtained in spite of the turn, would almost suggest that the important rule of hydraulics of avoiding all sharp turns does not apply to water turbines, which, of course, is not true—or else the turbine as a whole is so perfect that the loss undoubtedly caused by the sharp turn is balanced by other gains.

In speaking of the classification of water turbines as to speed, Mr. Larner makes the statement that in Europe the classification is done on the basis of peripheral speed. This statement is not quite correct, as it was in Europe that the method of classification on the basis of N_s (the author's N_u) was introduced first by Professor R. Camerer, of Munich, and Mr. Baashuus, and where, since 1905, it has been in general use. In the same year it was brought to the United States by European turbine designers who have transplanted their activities to America. Before that time, another characteristic, equivalent to N_u, but based on the discharge instead of the power, was used in Europe, also the "specific diameter" for which the author uses the symbol D_u, has been used by European engineers for the last five years. The first point considered by European engineers, in connection with turbine runners, is always the value of N_s (N_u). In the second place, comes the question of the peripheral speed, the coefficient of which, ϕ, is a very convenient runner characteristic, indicating at once whether a certain high or low value of N_u is due more to a high or low speed (large or small bucket angles), or to a high or low capacity. Mr. Zowski.

Although the boundary line between the different types cannot be drawn too sharply, and is subject to the fancy of the designer to a certain extent, the writer can scarcely agree with Mr. Larner in calling the runner of test No. 1795, a low-speed runner. Both the value of N_u and of ϕ make this runner belong to the medium-speed, medium-capacity type. Low-speed runners would have the following values, $N_u = 10$ to 28 and $\phi =$ about 0.58 to 0.65 (0.7). Therefore, Mr. Larner's statement that the most efficient speed varies with the gate opening, increasing as the latter increases, and that this is true of all reaction turbines, whether of low or high speed, is not justified by the tests, Nos. 1795 to 1800, as these tests covered only medium- and high-speed wheels. If the writer's memory is correct, experiments on low-speed turbines have not yet been made in this direction, or, at least, such experiments have not been published. That the best speed of medium- and high-speed runners increases with increase in gate openings is a well-known fact, established by theoretical analyses and many experimenters long ago.

In general, it might be said that there is a wide field for modifications and different solutions of the problem of variation of speed with varying gate openings. If, for instance, the main object is to obtain a flat efficiency curve, or, in other words, a so-called flexible runner, the runner should be designed in such a way as to obtain the best speed diagrams at the entrance (shockless entrance) at one, say, normal gate opening, and the best speed diagrams at the discharge (perpendicular discharge) at another gate opening. It is not necessary to state that this will reduce somewhat the absolute best efficiency.

In close relation to this is the question of the best efficiency gate opening. It is true, as stated by the author, that with low- and medium-speed turbines the best efficiency gate opening is usually at 0.7 to 0.75, whereas with high-speed turbines this is at 0.8 to 0.85. The reason for this difference is the characteristic of the gate-efficiency curves for the different types on the one hand, and the desire of getting the best possible performance at full gate, on the other hand. With the flatter curves of the first two types this last is easily obtained, even if the peak of the curve is at 0.7 gate, whereas with the more peaked curve of the latter type the drop in efficiency would be too large if the peak were at the same distance from the full gate as before. It should be understood, however, that it is in the hands of the designer to have also with high-speed turbines the best efficiency at 0.75 gate, if this is preferable.

Mr. Larner.

Chester W. Larner, Esq. (by letter).—Professor Moody's chart, Plate XXXII, comprising the best results obtained by the representative builders of America and Europe, is a very valuable contribution to this subject. The writer is particularly glad that it was submitted in connection with this paper, because he desired to prepare such a chart himself at the time of writing the paper, but realized the futility of attempting to secure the necessary information from his competitors.

This chart fully substantiates the statement, made in the paper, that the results published there are the best thus far obtained in the practice of the art. The efficiencies shown by the five tests presented are higher in each case than the efficiency of any other wheels of corresponding capacity and speed. In fact, through the range of "medium" and "high-speed" wheels, the curve of maximum efficiency is determined by the writer's points. Furthermore, it should be noted that these points are from a consecutive series of tests covering a wide variation of design, and do not represent, as do all the other points plotted, the best results culled from a collection of relatively inferior ones.

The only points from the paper which lie below Professor Moody's curve are 1800 and 1795. The former (efficiency 88.6%), as previously explained, was an experimental wheel of incongruous design, and was not expected to show maximum results. The other point, 1795 (efficiency 87.7%), does not do full justice to this wheel, as compared with wheels of the same type in operation under high heads, for the reasons explained in the paper; and if it had been tested under a head as high as that under which the two tests, *A* and *B*, of the I. P. Morris Company were made, there is little doubt that this point also would lie on or near the curve. Professor Allen gives interesting evidence corroborative of this statement when he cites the case of a high-head wheel which he tested in the Holyoke flume before installation and afterward in place, and found that the efficiency in operation was several per cent. better than that shown in the testing flume. This evidence is interesting because it is only in exceptional cases that such a direct comparison can be made, but the conclusion arrived at is so clearly obvious from a consideration of the elementary principles of mechanics, that experimental proof is almost superfluous.

Points *A* and *B* of the I. P. Morris Company, while speaking very highly for Professor Moody's ability as a wheel designer, testify also to the truth of the writer's contention regarding the advantage of testing wheels under high heads, because the best result obtained by this company, out of all the tests it has made in the Holyoke flume, is point *C*, which is about 3% lower than its best test in place, namely, point *B*. The writer is gratified to note that Professor Allen, who has had such a wide experience in testing water-wheels, agrees with him on this point. Mr. Larner.

Another interesting point brought out by this chart is the exceptionally high specific speed of the writer's wheel, 1796, as compared with the known records of other high-speed wheels. At a value of $N_u = 100$ it shows an efficiency of practically 80 per cent. The only other wheel which has ever attained as high a value of N_u is one designed by Professor Wagenbach of Charlottenburg.* The efficiency shown by this test was much lower than test 1796, as can be seen by reference to Professor Moody's chart, or to Fig. 29, where the efficiencies of both wheels are plotted for three different values of N_u.

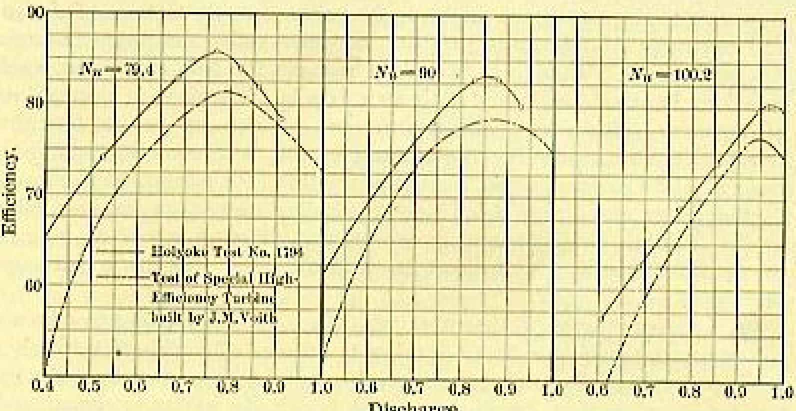


FIG. 29.

In order to show the real significance of the results obtained thus far in high-speed wheels, Table 10 has been prepared showing the maximum amount of power which could be produced at an efficiency of 80% by each of the four wheels on Professor Moody's chart which show the highest specific speeds. The power is computed for one runner at 150 rev. per min. under 50 ft. head. The values of N_u are taken from the chart.

TABLE 10.

Name of wheel.	N_u at 80% efficiency.	Brake horse-power.
Wellman-Seaver-Morgan Co., 1796	99	7 700
Wagenbach experimental	88	6 100
Allis-Chalmers Co., 1778	83.5	5 480
Swedish wheel (Curve *H-I*)	83	5 400

* *Engineering News*, September 9th, 1909.

Mr. Larner.

These values are obtained by substitution in Equation 2, which becomes:

$$\text{Horse-power} = \left(\frac{N_u \times 50^{\frac{5}{4}}}{150}\right)^2 = \left(\frac{N_u \times 133}{150}\right)^2$$

These figures will hold the same relation for any head or any speed, and, therefore, it may be stated in general that at any given speed and under any given head, wheel 1796 will develop about 40% more power at an efficiency of 80% than any other wheel of American manufacture, and about 25% more than the highest speed wheel thus far produced in Europe.

The writer is unable to appreciate the logic of Mr. Kuichling's objection to the statement made in the paper that the details of design and methods of obtaining results concern only the manufacturer who uses them. There are two chief reasons for this statement, each sufficient in itself to justify such an attitude. The first is that points of superior design in turbine runners require as much thought, originality, and outlay for experiments as the most complicated inventions in other lines of work. If the protection of a patent is sanctioned in the latter case, why should not a similar protection be justifiable in the former. Such improvements are not patentable, but as long as the builder keeps his designs to himself they are in a measure protected.

The second reason is the utter uselessness of attempting to educate the operating engineer to an intelligent choice of turbines through a study of variations in runner designs. It is the most complicated subject in mechanical engineering with which the writer is acquainted, and it would be a hopeless task for anyone but a specialist to acquire a sufficient knowledge of it to be of any practical value. Furthermore, the writer is fully convinced that most engineers do not care anything about such matters. They have neither the time nor the inclination to study them. They want to know as much as possible about the practical results which can be obtained, but they are satisfied to leave the methods of arriving at those results to the designer.

This, of course, is not true of all kinds of machinery, and the writer is inclined to think that Mr. Kuichling is partly right in what he says, but makes his mistake in that he does not distinguish between what is practicable and what is not. There is a great deal of machinery which embodies in its design only the simpler principles of mechanics, on which any competent engineer can readily pass judgment. Most of the mechanism of a turbine falls in this category, and no builder makes any secret of this part of his design; but the guides and runner are entirely different, and they are of more importance than any other parts. There is no living man who can tell with any degree of certainty, by merely looking at a runner, what the characteristics of its

Mr. Larner.

performance will be. There is no definite law, as Mr. Kuichling suggests, governing the relations of the diameter, depth, discharge, etc. These relations are entirely different for wheels of different types, and even for wheels of the same type, that is, the same unit speed. All these proportions vary with the design of the wheel vane, and the latter is an intangible factor to the average engineer. In fact, if a drawing of a wheel vane were put in evidence, there is not one engineer in a thousand who would know what he was looking at.

As to other features of turbine design which can be more readily analyzed, the writer agrees with Mr. Kuichling that more data are desirable. One point raised by him is the proper size of the draft-tube. He states that some engineers are of the opinion that it cannot be made too large, while others believe that it should be of such a size as to give a velocity of 10 ft. per sec. at full gate. When two such contradictory and equally erroneous theories are abroad, it would certainly seem that more light is needed.

The design of a draft-tube is no very complicated matter. The controlling factor in the design should be a uniform reduction of velocity from the runner to tail-water, and, in speaking of the "draft-tube," the writer includes the entire channel from the runner to tail-water. Bearing in mind the principle of uniformly reduced velocity, the problem becomes one of determining the diameter of the upper end of the tube, the diameter of the lower end, and the length.

The diameter of the upper end should be the same as the diameter of the discharge end of the runner, as shown in Figs. 1 and 2, and if, as in Fig. 1, the wheel is of large capacity, with some outward discharge, it is well to give the tube a slight flare at this point, although this can be easily overdone. There should be no sudden enlargement from the runner to the draft-tube, because such an increase in cross-section tends to check the velocity of discharge suddenly and create a disturbance of the stream. This principle of design is frequently violated, both in America and Europe. The writer has seen on turbines discharge cases which looked more like dry-goods boxes than anything else, which did not even approach the circumference of the runner, and made no pretense whatever of providing for uniform reduction of velocity. The runner discharged into the draft-chest like a jet from a hose discharging into a pond, and with about the same efficiency. One installation of this description which the writer has in mind was finally consigned to the scrap pile because the efficiency was found to be only about 65 per cent. The runners showed an efficiency in the Holyoke testing flume of more than 80%, and this astonishing discrepancy was probably due almost entirely to this particular feature of the installation.

The diameter of the draft-tube at the lower end is dependent to a considerable extent on the diameter at the upper end, and the length

Mr. Larner. of the tube. The latter is frequently fixed by conditions independent of the turbine, and as there is a limit to the amount of flare desirable in a draft-tube, the diameter of the lower end sometimes has to be made smaller than one would wish. The writer considers a flare of 1 ft. in diameter to 3 ft. in length about the limit, and prefers less than that if possible. The criterion for judging whether or not a draft-tube is too small at the discharge end is the percentage of energy lost in velocity at that point. Under low heads this loss can seldom be kept below 1%, whereas, under high heads, it is usually a fraction of 1%, but only because high-head wheels are usually of the reduced discharge type, and hence, on account of the relatively smaller quantities of water to be handled, the velocity of outflow from the draft-tube can be reduced to a relatively lower figure without making the tube of prohibitive proportions. For a given flare, the longer the tube the lower the velocity of discharge, and hence the more efficient the tube; remembering, however, that a safe limit for the elevation of the top of the runner above tail-water, when set in a horizontal position, is about 25 or 26 ft. In the case of low-head plants, where the wheels, in order to be sufficiently submerged, must be set close to tail-water, it is often necessary to form the draft-tubes in the foundations of the power-house, and to carry these passages sometimes below the level of the tail-race and come up to that level after the discharge from the wheels has been released to the atmosphere. It should be remembered that the effective draft-tube extends to the point where the water is released, and that the energy lost in residual velocity is the velocity head at that point. The percentage of energy lost is:

$$\frac{V^2}{2g} \div h \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (4)$$

where V is the residual velocity and h is the working head on the plant.

From these considerations, it must appear that no fixed figure can be named for the desirable velocity in a draft-tube. The velocity depends entirely on the head. Suppose, for example, a wheel is operating under 9 ft. head. It discharges 100 cu. ft. per sec. Assuming that the maximum size possible for the discharge end of the draft-tube, without excessive flare, is 5.7 ft. in diameter, then the residual velocity will be 2.4 ft. per sec., which, according to Equation 4, is:

$$\frac{2.4^2}{2g} \div 9 = 0.01, \text{ or } 1\% \text{ loss.}$$

Now suppose the head on the wheel to be increased to 900 ft. If the speed is allowed to change correspondingly, the discharge will vary as the square root of h. The discharge will become:

$$\frac{100 \times \sqrt{900}}{\sqrt{9}} = 1\,000 \text{ cu. ft. per sec.}$$

The residual velocity will then be $2.4 \times 10 = 24$ ft. per sec., and the loss, according to Equation 4: Mr. Larner.

$$\frac{24^2}{2g} \div 900 = 0.01, \text{ or } 1\% \text{ loss,}$$

as in the previous instance. Hence it is plain that a certain size of draft-tube, in connection with a certain runner, will give the same draft-tube loss, no matter what head it is used for, and that the velocity of outflow can have no arbitrary or empirical value, but must vary with the head.

As to the relative efficiencies of right-hand and left-hand wheels, there certainly should not be, nor is there, so far as the writer knows, any difference. The records seem to show that there are about as many efficient wheels of one type as the other. There is no theoretical reason to believe otherwise, and the only alleged reason that the writer has ever heard is a fanciful notion about the effect of the earth's rotation.

There is also no reason to believe that there is any difference between the efficiencies of wheels set in horizontal or vertical positions, or even in an inverted position. If the head is the same in all positions, the discharge will be the same under like conditions of gate and speed, and if the discharge is the same, then all the hydraulic conditions are duplicated and the efficiency must be the same. This, of course, refers to the efficiency of the runner alone, because the efficiency of the turbine may be very different on account of the mechanical losses incidental to different settings. In this respect the advantage is usually in favor of the vertical setting, because the weight is ordinarily carried on practically frictionless bearings, which is not the case in horizontal settings.

The only test to determine this point, of which the writer knows, was made some years ago at the Holyoke flume on wheels for the Michigan-Lake Superior Power Company at Sault Ste. Marie, Mich., one of the oldest installations of the Wellman-Seaver-Morgan Company. These wheels were tested both ways, and as there were forty-four units of four runners each, a great many were tested. Some were tested singly on a vertical shaft and then in pairs on a horizontal shaft, in the casings in which they were to be permanently installed, and it was found that the efficiency was the same in each position, provided the wheels were placed far enough apart on the horizontal shaft. If placed too close together, the loss of efficiency was very marked. It is to be regretted that more tests of this sort have not been made, for the purpose of checking the tendency of many builders to crowd their wheels together in order to shorten the shaft and effect a corresponding "economy" in design.

Referring to Mr. Kuichling's request for data in regard to the

Mr. Larner. distribution of velocities in the draft-tube and guide passages, as determined by Pitot tubes, the writer believes there is sufficient material in this subject for a separate paper, and, when the opportunity affords, he intends to prepare some of this information for publication. For the present it may suffice to say that the phenomena observed in the draft-tube are in general about as follows: Starting with the wheel held stationary and the gate open, the velocity of the water at the center of the tube is very high, and at the circumference it is low. The water is whirling at all points on the diameter in a direction opposite to the direction of rotation of the runner. Both these phenomena would naturally be expected. The incoming stream from the guides, having an initially radial direction, naturally crowds to the center and causes a high velocity there. When the wheel starts to revolve, and as it increases in speed, the centrifugal force of the revolving mass of water in the wheel tends to counteract the first condition. The water presses outward toward the band of the runner and the velocity across the diameter of the draft-tube becomes more evenly distributed. The distribution, however, is never uniform. The velocity in the draft-tube, as in any other pipe, is always highest at the center except at very high speeds, when the effect of the centrifugal force is so great that there is almost a void around the shaft.

The angle of whirl when the wheel is stationary is backward, because the flow of the water relative to the wheel is backward at all times, but is offset, when the wheel is running, by the velocity of the wheel in the opposite direction. When the velocity of the wheel at any point on its diameter is equal to the horizontal component of the velocity of discharge relative to the wheel, then the velocity of discharge relative to the draft-tube, or the absolute velocity, will be straight down, and there will be no whirl. As the speed of the wheel increases beyond this point the water begins to whirl with the wheel. This is just what is observed in testing with the Pitot tube.

One very interesting fact shown by these experiments is entirely at variance with the accepted beliefs regarding the action of the water in the runner. It is generally supposed that "radial outflow" or discharge without whirl occurs at the most efficient speed of the wheel. The idea is that because radial outflow is the theoretical condition for maximum efficiency, therefore the most efficient speed will be the one where radial outflow* occurs. The tests, however, show that this is not the case. The speed at which whirl is eliminated from all parts of the discharge area, except that close to the center of the shaft, is much lower than the most efficient speed. The water starts to whirl with the wheel at a comparatively low speed, and increases until it

* The term "radial outflow" is commonly used in reference to all types of wheels, but is strictly correct only in reference to purely inward- or outward-flow wheels. For mixed-flow wheels, the correct term would be "axial outflow" when applied to the stream after it has emerged from the band of the runner.

Mr. Larner.

shows an angle of from 15 to 30° from the vertical. This value varies with different wheels, but remains fairly constant up to speeds much higher than the best speed. The explanation apparently is that centrifugal force, increasing with the speed, accelerates the backward velocity of discharge about as fast as the forward velocity of the wheel increases.

At the point of maximum efficiency, the conditions observed were about as follows: The velocity was highest at the center of the draft-tube and lowest at the circumference. The water had no whirl whatever at the center, as far as could be measured. At the circumference it was whirling with the wheel at an angle of from 15 to 30 degrees. These conditions shaded into each other at intermediate points. The water seemed to flow in a quiet, solid mass, without the conflicting angles of whirl and velocities which were observed at other speeds.

The writer is convinced that one of the chief losses of energy in a turbine occurs at discharge from the runner, and that most poor wheels owe their inefficiency more to an improper distribution of the discharge area between the vanes than to any other cause. If the discharge opening is not properly proportioned, the adjacent angles of whirl and velocities of discharge will conflict, causing great disturbance.

The effect of gate opening on the flow in the draft-tube is to increase the velocity at the center. This, to a certain extent, accounts for the fact that the most efficient speed of the wheel increases with the gate opening. Each time the gate is opened farther the water tends to crowd more to the center, and hence more centrifugal force, requiring a higher speed, is needed to secure the proper distribution of flow across the diameter of the draft-tube.

The writer notes that Mr. Mead, in his interesting discussion, takes exception, in some respects, to the use of unit speeds as a criterion of the fitness of a wheel to perform certain work. Mr. Mead says that "all wheels having the same value of N_u are not necessarily of similar value in a given installation."

The writer did not claim that all wheels having the same unit speed are equally desirable, but that all such wheels are able to perform the same work. In other words, if an engineer intends to buy some turbines to operate under conditions of head, speed, and power which require a unit speed of 60 at full gate, and several builders submit tests of runners which show a unit speed of 60 at full gate, then he knows with absolute certainty that all these tests represent types which will produce the power he wants at the specified speed and head. The relative desirability of the wheels depends on their respective efficiencies.

The required wheels may not be of the same size as the wheels that

Mr. Larner. were tested, but there can be no shadow of doubt that some size of each of these types will do the desired work, and it is a simple matter to determine the sizes. Mr. Mead admits that changes of size can be effected with greater certainty than changes of design, and therefore it seems to the writer of far greater importance to determine whether the builder has developed a wheel of the right type, even though it is of considerably different diameter, than it is to determine whether he has a wheel of the desired size but of only approximately correct type. It was for this reason that the writer tried to emphasize the point that unit speed is an index of type and should be so regarded to be of the greatest practical use. In the first case, just referred to, the builder needs only to alter the size of his wheel, whereas in the second case the other builder would have to alter his design, and the outcome would be far more uncertain.

Mr. Mead illustrates his use of N_u in the selection of turbines by reference to an installation under the following conditions:

Head = 16 ft.
Horse-power = 650 (325 h.p. per runner).
Revolutions per minute = 128.

These conditions, when reduced to a basis of 1 ft. head, give:

$$N_u = 32.5\sqrt{5.07} = 72.4.$$

He refers to five tests submitted, but the writer will discuss only two—Nos. 1 and 4. These wheels showed the following values:

$$\text{No. 1}: N_u = 32\sqrt{4.85} = 70.$$
$$\text{No. 4}: N_u = 35.7\sqrt{4.00} = 71.4.$$

Wheel No. 1 was accepted and No. 4 was rejected, although the unit speed of the latter was nearer the desired value than that of the former. Mr. Mead justifies his choice on the ground that No. 1, while slightly deficient in power and speed, had high efficiency, whereas No. 4 had too high a value of N' (35.7) and too low a value of P' (4.00) and "was therefore unsuited to the conditions."

The writer takes exception to the last part of this statement, for the reason that wheel No. 4, although not itself of the proper size for this installation, nevertheless represents a type of wheel a little better suited to the conditions than No. 1. He does not criticize Mr. Mead's selection of wheel No. 1, because there is very little difference between the unit speeds of these two wheels, and, since it was possible to reduce the power of the units, wheel No. 1 fitted the conditions almost exactly both as to type and size. If, however, it had been essential to get the full 325 h.p., then (neglecting consideration of efficiency), the writer maintains that No. 4 was at least as suitable, if not more so, than No. 1, because, whereas both wheels would require alteration, No. 4 approximated more closely to the required type than No. 1, and was better evidence of the builder's ability to fulfill the specifications. The difference in this case is so slight that the distinction is purely academic, but, nevertheless, as illustrating a principle, No. 4 should have the preference, and the writer does not think that Mr. Mead is justified in stating that this wheel is not suited to the conditions. Assuming that No. 4 was a 41-in. wheel, the only change necessary to make it run at the right speed would be to increase its size. The new diameter would be: Mr. Larner.

$$D'_4 = \frac{41 \times 35.7}{32.5} = 45 \text{ in.}$$

As to the matter of standardizing runners, the writer thought that (on page 314) he had made his meaning clear, but Mr. Mead seems to have drawn mistaken conclusions. By "standardization" was meant the well-known practice of building several distinct types in various standard sizes and adhering to these standards even to the extent of using the wheels under conditions to which they are unsuited. By "special" wheels was meant the precise adaptation of the wheel to the conditions of operation, whether by the development of a new type or the modification of an old one, and it is specifically stated that the latter method embodies an element of greater reliability than the former. To quote from the paper:

"New designs will consist rather of modifications of and improvements on wheels which have been tested and found to be efficient, and, as such, will be more certain of success than if based wholly on theoretical considerations."

Mr. Mead, in the beginning of his discussion, speaks of the difficulty of designing wheels to develop their best efficiency under the fixed conditions of an actual installation. He says that he does not believe it is possible, without considerable experimental work, to design a wheel "that will unquestionably secure the desired results under actual operating conditions." The writer admits that the problem of making a wheel to fit fixed conditions, without any leeway, is the most difficult task the designer has to meet, but he believes that, with sufficient experience and a proper understanding of the work, it can be done without much uncertainty. Fig. 30 shows the efficiency curves of four runners of a new type designed by the writer for a set of fixed conditions. There was no experimental work of any kind preliminary to these four designs, which, although prepared to meet the same conditions, were radically different in their details.

Wheel 1801 was designed under the terms of a contract which required the wheels to show their maximum efficiency under conditions which, when reduced to a basis of 1 ft. head, were as follows:

Revolutions per minute = 28.3.
Discharge = 48.8 cu. ft. per sec.

Mr. Larner. There is no more absolute way of specifying the performance of a wheel than that. Fig. 30 shows that the efficiency curve of this wheel peaked at exactly the specified discharge.

Inasmuch as there was a large bonus for efficiency at that discharge, the writer, being familiar with the loss of efficiency incidental to tests of large wheels at Holyoke, wished to find out whether the efficiency obtained with the first wheel could be improved upon. For this purpose, three more designs were prepared simultaneously, and as much variety as was feasible was embodied in these three additional wheels. They were purposely made different from the wheel that had been tested in order to determine if it could be improved upon. Fig. 30 also shows the tests of these three additional wheels, 1828, 1838, and 1839. As may be observed, the efficiency curves of all these wheels show practically their maximum values at the specified discharge.

The writer believes that if results as precise as these can be obtained four times in succession under conditions as stringent as stated, it must appear that no very considerable element of uncertainty exists.

With reference to Fig. 20, which Mr. Mead presents as evidence in support of his contention, the writer has no hesitation in saying that this is an example of inexcusably bad design. There was considerable leeway in the guaranty, and the design of the wheel was a simple matter. The test, however, shows that the wheel was grossly over capacity.

A great many such instances could probably be cited from the general experience of water-wheel users, but they do not represent by any means the best that is being accomplished in turbine construction in America at the present time.

In reply to Mr. Zowski's criticism relating to the omission of detailed descriptions of the wheels and other parts of the turbine, the writer would point out that the paper was necessarily limited in its scope. It was addressed particularly to turbine users, rather than turbine designers, and the former are not generally interested in details such as asked for by Mr. Zowski.

The writer attributes the good results obtained to the correct design of the runners and guides and the proper co-ordination of these parts. The mechanical work was no better than usual, and, in fact, the writer has very little faith in the efficacy of putting a high finish on the runner and guides. It is true, of course, that the smoother the walls of the water passages, the lower the friction loss will be, but the passages through the runner and guides are so short that the loss due to skin friction alone is very small in any case. The principal losses are due to internal friction or disturbance. In the case of these runners, the only finishing done on the vanes consisted in chipping off the fins caused by the clearance between the cores. The guide-vanes were finished only on their points of mutual contact.

Mr. Larner.

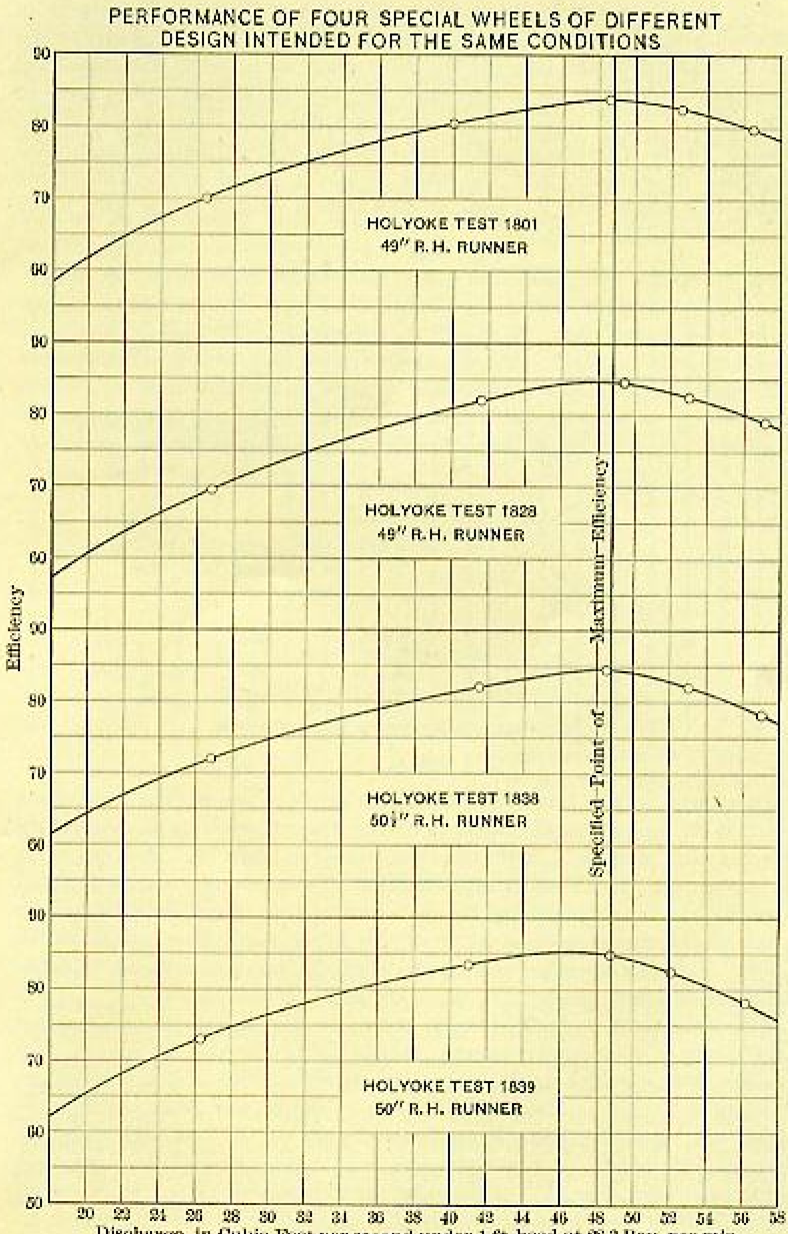


FIG. 30.

Mr. Larner.

Mr. Zowski refers to the sudden turn of the water in flowing from the guides into the runner, as shown by Fig. 1. If one imagines the water to flow over the edge of the curb plate in a radial direction, the condition would seem to be very bad, because the stream would have to turn a sharp right angle, but, if one considers the true course of the stream, which approaches this point in an approximately tangential direction, it is plain that the turn is not a sharp one. The water crosses this edge with a spiral motion which cannot cause any great disturbance of the stream.

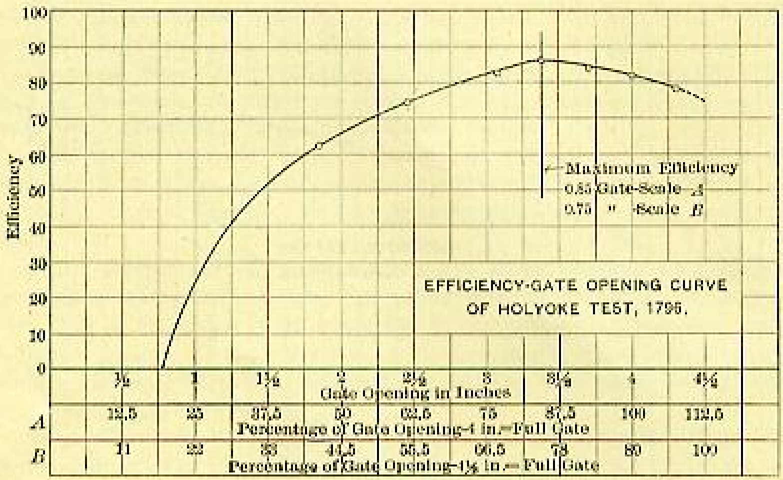


Fig. 31.

In regard to the European classification of runners, the writer has observed in many foreign textbooks that the value of ϕ is frequently used for this purpose. Attention was called to this fact in support of the contention that such a classification is not always consistent and never should be replaced by the use of the unit-speed method, which is exact. It was not claimed that the latter method is unknown in Europe. The writer was well aware that the method originated there and the European symbol, N_s, is referred to in the paper.

The writer takes decided issue with Mr. Zowski on the last statement in his discussion, namely, that "it is in the hands of the designer to have also with high-speed turbines the best efficiency at 0.75 gate, if this is preferable." Experiments and theoretical considerations show that this is an impossibility if the opening which is termed "full gate" is selected with reasonable regard for ordinary operating conditions. It can be done in the case of typical high-speed wheels only by over-gating the wheel to a point which would never be used under operating conditions.

Fig. 31, showing Test 1796 plotted to efficiency and gate opening, will serve as an illustration. In reporting this test a 4-in. gate opening was selected as full gate for the reason that, as shown by Fig. 32, the wheel gave practically its maximum power at this gate. Any further opening would be over-gating, because considerably more water would be used to obtain a very trifling increase of power. In fact, if the gates had been opened to 4½ in., there would have been a loss of power rather than a gain. Under such circumstances, 4 in. should certainly be regarded as "full gate." Mr. Larner

On the basis of 4 in. = full gate, the maximum efficiency occurs at 0.85 gate, as shown by Scale *A*. On the basis of 4½ in. = full gate, it occurs at 0.75 gate, as shown by Scale *B*. The latter assumption accords with Mr. Zowski's contention that the designer may make the maximum efficiency occur at 0.75 gate, but since it can be done only by over-gating to an extent which would never be used in practice, the assumption is untenable, from the standpoint of the operating engineer.

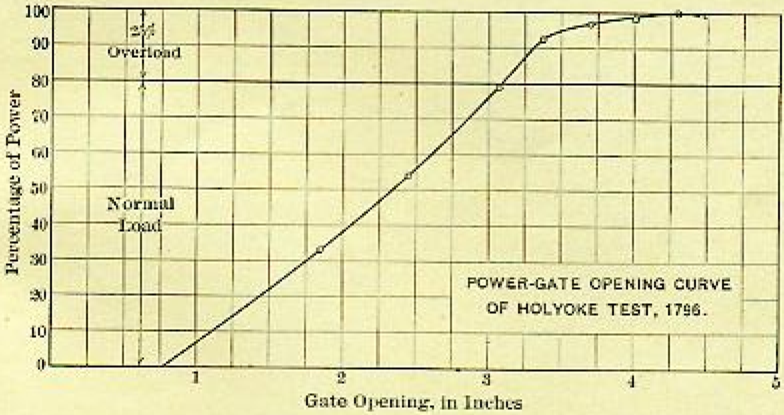


FIG. 32.

The writer regrets that he introduced the question of gate opening in his paper in this connection, because it is too indefinite a term to be of much value. One man will call a certain opening "full gate," and another will call some other opening "full gate." There is no way to determine the point definitely except to define "full gate" as the opening at which the wheel shows its maximum power. This method is open to the serious objection that in practice the gate opening is never, or very rarely, carried to this extreme point, because, by the time the peak of the power curve is reached, the efficiency will be relatively poor.

Percentage of load is a much more rational basis for performance guaranties than percentage of gate opening. Full power is readily

Mr. Larner. determined, whereas, full gate is not. Fig. 32, for example, shows plainly that the logical position of maximum gate opening is about 4 in. Also that "normal gate" is about 3 in. or 0.75 gate, and a comparison with Fig. 31 shows that maximum efficiency occurs at 92.5% load, or considerably above normal load. This is the point brought out in the paper, that with high-speed wheels the point of maximum efficiency is too near the peak of the power curve to give the best results, if the plant is designed for 25% overload.

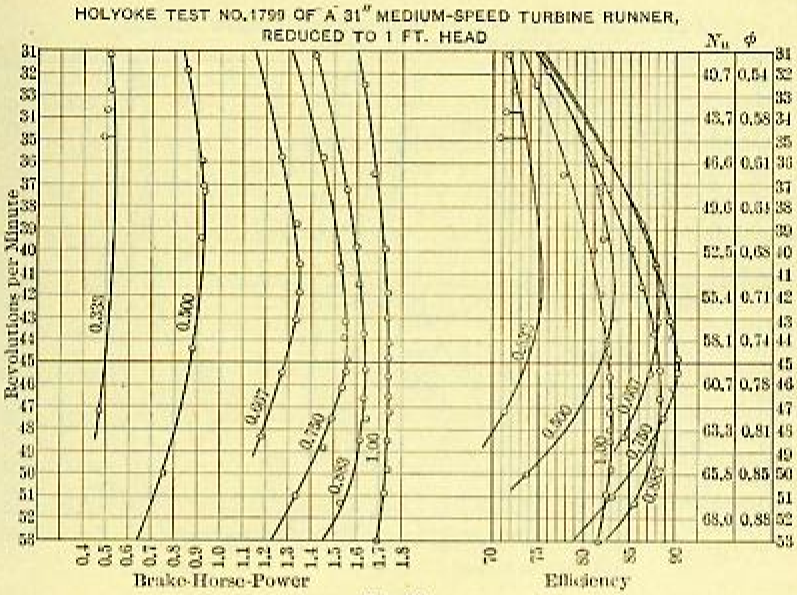

HOLYOKE TEST NO.1799 OF A 31" MEDIUM-SPEED TURBINE RUNNER, REDUCED TO 1 FT. HEAD
Revolutions per Minute
Brake-Horse-Power
Efficiency
N_u ϕ

Fig. 10.

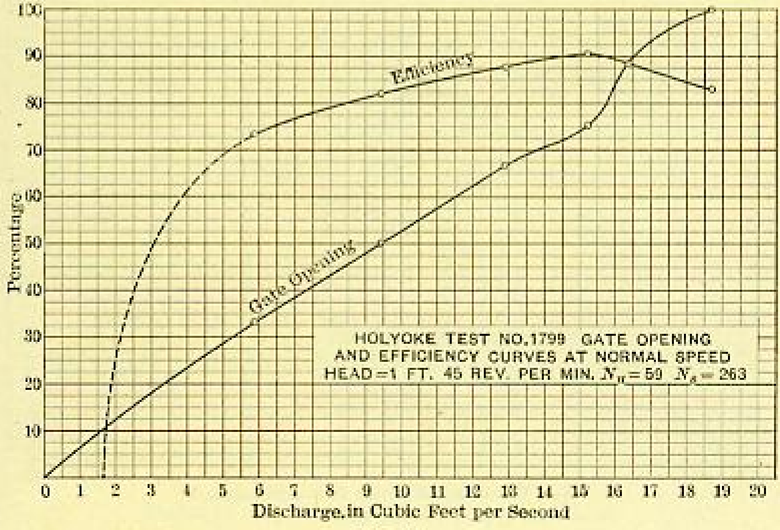

Efficiency
Gate Opening
Percentage
Discharge in Cubic Feet per Second
HOLYOKE TEST NO.1799 GATE OPENING
AND EFFICIENCY CURVES AT NORMAL SPEED
HEAD=1 FT. 45 REV. PER MIN. $N_u = 59$ $N_s = 263$

Fig. 11.