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No. 1828

THE MOODY EJECTOR TURBINE

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ONE of the greatest difficulties met in the operation of low-head hydroelectric power plants is the reduction of capacity during flood periods. It is usually impossible, owing to the nature of the surrounding country and the enormous quantity of water required, to provide storage reservoirs to such an extent that all excess water flowing during these flood periods can be entrained and used when the stream flow is below normal. Some regulation, of course, is effected in a few instances, but for the most part there is sufficient only to carry the plant over the daily peak.

2 Where the storage is thus limited, owing to the character of the country surrounding the plant, the flow of water during the flood season is much greater than is required by the turbines. This excess is therefore wasted over the dam, and in addition causes the tail water to rise in level.

3 Since the turbine operates upon the difference in level of the head and tail waters, should these levels vary the same amount, then this difference or head would be constant. Usually this is not the case, for with a given rise in level of the water going over the dam, the rise in the level of the tail water is greater, thus decreasing the net head. This effect is illustrated in Fig. 1. The head at low water is evidently much greater than at high water as reference to this sketch will show. The line intermediate between the two extremes shows the effect when the level of the head water is increased by a small amount. It is evident even in this case that the net head is decreased.

4 Since the horsepower of the turbine varies as the three-halves power of the head acting upon it, the output of the station is

Extracts from a thesis submitted to the University of Pennsylvania for the degree of Bachelor of Science in Mechanical Engineering by the author in June 1921. Awarded A.S.M.E. Junior Prize for the best paper during the year 1921.

decreased in the same ratio, but as the operation is of necessity at constant speed, the efficiency may be less as well, depending upon the characteristics of the wheel, causing a still further decrease in capacity. Thus the resultant loss in power may be very great, depending upon the rate of off-flow of the tail water, which in turn is dependent upon the character of the stream bed below the dam.

5 As was stated above, in low-head plants where the quantity of water required to generate a given amount of power is large, the storage facilities are usually small. Therefore the turbine should be designed to give maximum efficiency at rated output when the head is a maximum, for at this time water is scarce and economical operation is imperative.

6 Due to their inherent characteristics, the high- or relatively

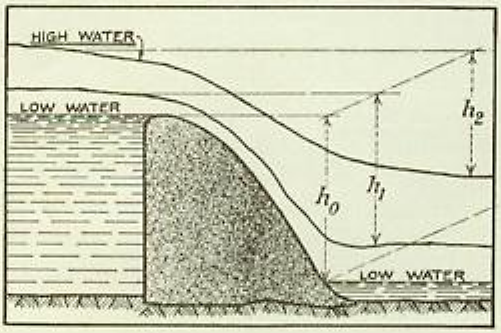

Fig. 1 Sketch Showing Decrease in Effective Head During Periods of High Water

high-speed wheels used in low-head work give as a rule a rather narrow operating range at maximum efficiency, and a small overload capacity beyond this point of maximum efficiency. To design a turbine, then, that would deliver the required output under a reduced head would be inadvisable, for this same turbine would not operate at best efficiency when delivering rated horsepower under the higher normal head resulting from low water.

7 This is illustrated in Fig. 2 where the turbine is designed to deliver 100 per cent of rating at 75 per cent of normal head. When this same turbine is operated under 100 per cent head, its capacity is increased to 140 per cent of rating; but as only 100 per cent is required, it is evident that the efficiency will be several per cent less than the maximum. This method of meeting the problem would be

wasteful from the standpoint of economy of operation at maximum-head conditions.

8 Another method would be to install a number of additional units that could be placed in operation when the capacity of the plant was cut down at high water. Such a plant would require a much greater expenditure of money in the first cost, which could render a return on the investment for a short period only each year. This, then, is a possible solution, but the resultant fixed charges upon the plant would be so great, in many cases, as to make the cost of power prohibitive.

9 There remains this problem: to design a turbine that will

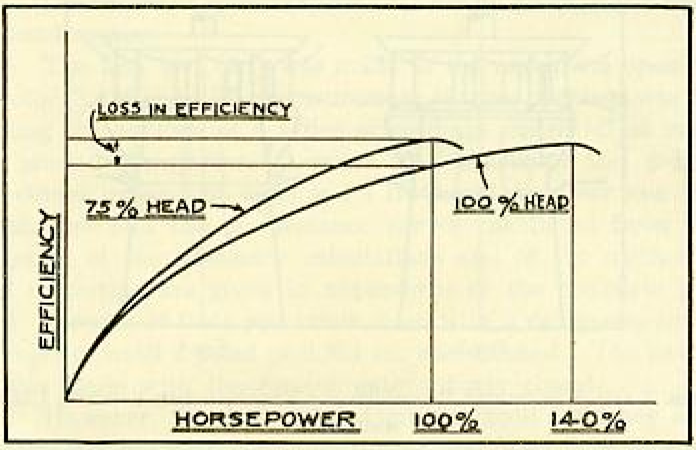

Fig. 2 Sketch Showing Disadvantage of Designing Turbine for Maximum Efficiency at the Minimum Head Conditions

operate as economically as possible at times when the available supply of water is small, and yet develop normal rated power when the overabundance of water decreases the effective head.

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10 A turbine designed to meet these requirements has been developed by Mr. Lewis F. Moody. The apparatus consists of a turbine of the normal type designed to operate at maximum efficiency under the average head prevailing at times of low water, but at the top of the draft tube, just below the runner, an annular opening is made in the wall of the tube, the amount of this opening being controlled by a cylinder gate.

11 When the turbine is operating normally, this gate is closed and the contour of the draft tube is unchanged, as shown in Fig. 3. When the head falls off due to the rise of the tail water, this gate is opened slightly, allowing a jet of water at high velocity to enter the draft tube without first passing through the runner. This case is shown in Fig. 4. The action of the jet, expressed simply, is to "pull" more water through the runner and hence to increase the horsepower output of the turbine.

12 The jet actually creates an additional suction head that compensates for the loss in the difference in level between the head and tail water so that the turbine is in reality operating under a

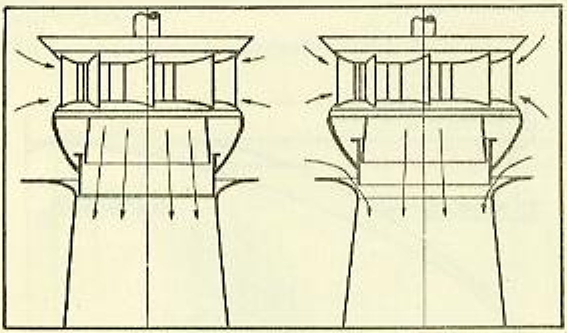

Fig.3 Fig.4

FIGS. 3 AND 4 SKETCHES RESPECTIVELY SHOWING THE EJECTOR CLOSED AND OPEN

head made up of two parts: first, the effective head measured as the difference in levels of the head and tail water (less, of course, the frictional head in the water passages); and second, the "suction head" produced by the action of the jet of water admitted below the runner. For this reason the device has been called an "ejector turbine."

TESTS MADE ON THE TURBINE

13 The apparatus described above was set up and tested in the I. P. Morris Experimental Laboratory of the William Cramp & Sons Ship & Engine Building Company in Philadelphia. The turbine was equipped with guide vanes placed in the ejector. The function of these vanes, which will be called ejector vanes hereafter in order to avoid confusing them with the guide vanes and

speed ring vanes of the turbine proper, is to give the water entering the ejector an initial whirl, so that there will be a minimum amount of interference with the water coming off the runner.

14 In the first series of tests (A1 to A6) model runner No. 55 was used. This runner (16 in. in diameter at the throat) is of the mixed-flow type and has a specific speed of 84.4 at best ϕ. The symbol ϕ represents the coefficient of peripheral velocity or the ratio of the linear velocity of a point on the runner to the theoretical spouting velocity of the water under the operating head. This ratio may be based on any one of several diameters, but in this article, the ϕ based on the diameter at the throat of the runner is meant. The values of specific speed (N_s) are all given in the English foot-pound system.

15 The first test (A1) was made at the maximum opening of the ejector (3.155 in.). The measurement of these openings was made by taking the average of a series of readings scaled off at various points around the ejector. A series of ϕ-Efficiency and ϕ-Horsepower curves under 1 ft. head and 1 ft. throat diameter was drawn for each test and the performance curves calculated from these. Description of the necessary calculations and of the method employed in testing are given in appendices to the complete paper.

16 A series of tests was made, each with a decreasing opening of the ejector until a point of 0.824 in. was reached. The next test (A6) was made with the ejector gate entirely closed.

17 However, this test did not give as good efficiency as was previously secured with the same runner and draft tube in another test. Upon investigation, it was found that the cylinder gate of the ejector did not seat properly upon the lower ring and that a small opening was left on one side of the turbine, thus admitting a small amount of water on about one-third of the periphery of the ejector. It is of interest to note that for any given gate opening of the turbine, although the efficiency was reduced about five per cent from that previously obtained, there resulted a slight increase in horsepower with even this small ejector opening.

18 In plotting performance curves of the values in test No. A6 the test efficiencies were stepped up to correspond with those obtained in previous tests. This is permissible, for in tests Nos. B6 and C2 made with runner No. 65, the ejector was calked tight to eliminate this difficulty and to correspond to the more perfect construction that would be found in large units built for commercial purposes. The results in these tests gave efficiencies that checked

very closely with those obtained when the same runner was tested without the ejector installed.

19 Another series of tests was made using runner No. 65 in place of runner No. 55. This runner (16.25 in. in diameter at the throat) is of the "diagonal-flow" type and has a specific speed of approximately 100. This series of tests (B1 to B6) was run with varying openings of the ejector, but the angle of the ejector vanes was changed to conform with the variation in the theoretical value of the whirling component of the water leaving the runner with the increase in specific speed.

20 When this series had been run, it was thought advisable to test runner No. 65 with the ejector vanes removed in order to study the effect of the ejector water without this initial whirl.

INVESTIGATION OF THE INFLUENCE OF EJECTOR UPON THE FLOW IN DRAFT TUBE

21 It was desired as well to see the effect that the water injected into the draft tube through the ejector would have upon the normal flow of water coming off the runner. In order to measure the velocity of water in the draft tube it is necessary to know the direction that it takes, since the flow is not the same as in a pipe line or canal, but moves with a whirling motion and follows a path similar to a helix in form. The steepness of this helix depends upon a great number of conditions, principal among which are the type of runner, the speed, and the load under which the turbine is operating.

22 An instrument was developed by the writer with the co-operation of the staff in charge of experimental work of the I. P. Morris Department especially for this test to measure the velocity and direction of flow in the draft tube. The apparatus, because of its form and the use for which it was designed, has been called a "pitot velometer."

23 The pitot velometer consists of a double pitot tube similar to the "Pitometer" with symmetrical openings 180 deg. apart. The head was arranged so the maximum dimension would pass through a hole the size of the outside diameter of standard $\frac{3}{8}$-in. pipe. The whole head, rod, and end connections are free to rotate and move in and out from center line of the draft tube. A pointer is fixed on the outer end of the instrument, and a dial hung on a ball bearing and weighted to keep the zero of the scale vertical showed the position of the center line of the openings in the head.

24 These two openings were attached to the legs of a differential gage filled with mercury and when the head was turned to a position so that the two openings were in a line with the flow of the water, the velocity head was indicated on the gage. When the rod was rotated through 90 deg., the center line through the openings was perpendicular to the flow, and the two legs of the gage stood at the same level. It was found that the apparatus was very much more sensitive in this position than with the openings in line with the flow, and that the angle of deviation from the vertical could be more accurately read.

25 For this reason two pointers were attached to the rod;

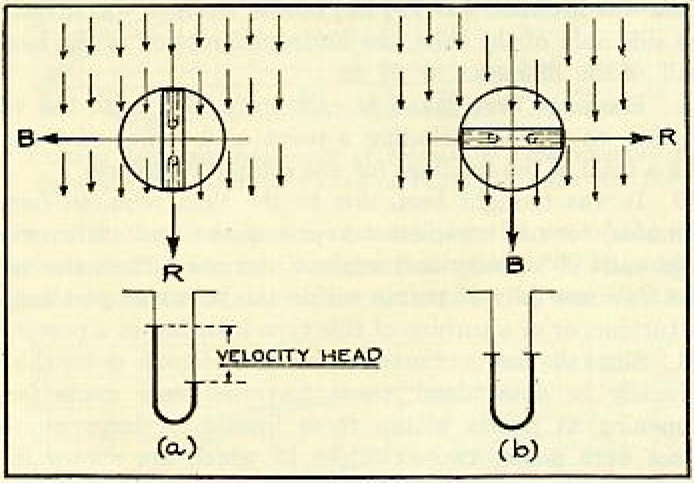

FIG. 5 POSITIONS OF PITOT VELOMETER (*a*) WHEN READING VELOCITY AND (*b*) WHEN READING DEVIATION FROM VERTICAL

(R indicates red pointer; B, black pointer.)

one painted black was set at right angles to the line through the center of the openings, and the other painted red set parallel with this line.

26 In taking a reading, the rod was set at the desired point on the diameter of the draft tube and rotated until the two legs of the differential gage stood equal; this position is shown in Fig. 5 (*b*). The reading of the deviation was recorded as shown by the black pointer. The red pointer was then brought to this point, automatically placing the openings in line with the flow and the

velocity read by the difference in levels of the mercury columns. This position is shown in Fig. 5 (*a*). The scale of the dial was graduated in degrees and half-degrees. It was possible to measure the deviation of this velocity from the vertical accurately to within a single degree.

27 The scale of the mercury gage was graduated to read zero when the two legs were equal; below this line the scale was laid off in inches of mercury. Above the zero line the scale was arranged to give the velocity directly in feet per second. The pitot velometer was located approximately 24 inches below the horizontal center line of the turbine.

28 The diameter of the draft tube at the point where the instrument was installed was $19\frac{3}{4}$ in., but as the flow was investigated on one side only of the tube, the horizontal motion of the head was one-half of the diameter or $9\frac{7}{8}$ in.

29 Readings were taken at each even inch from the vertical center line up to and including a point at a radius of 9 in., thus making a total of ten readings for one complete traverse.

30 It was thought best, due to the time required (seven to ten minutes) for one complete traverse of the draft tube, with ten readings each of velocity and angle of deviation from the vertical, to take traverses only at points within the possible operating range of the turbine, or of a turbine of this type installed in a power plant.

31 Since the limits of maximum and minimum ϕ for this range could easily be determined, three traverses were made for each gate opening at points within these limits. A large number of traverses were made, two examples of which are shown in Figs. 14 and 15.

32 Two tests were made, the first (C1) was run with the ejector open about one and one-half inches, the second (C2) was made with the ejector closed and calked, to have, as a basis of comparison, the flow of water coming from the runner uninfluenced by the ejector water.

RESULTS OF TESTS

33 In order to show the increase in horsepower obtained by use of the ejector over that normally delivered by the turbine, a series of performance curves was drawn with efficiencies as ordinates and horsepower under 1 ft. head reduced to 1 ft. throat diameter as abscissae. These curves were drawn for each of the tests in the first and second series. The envelope of these curves was then

drawn in and is taken as the performance of the turbine and ejector combined.

34 It may be seen from Figs. 6 and 9 that a considerable in-

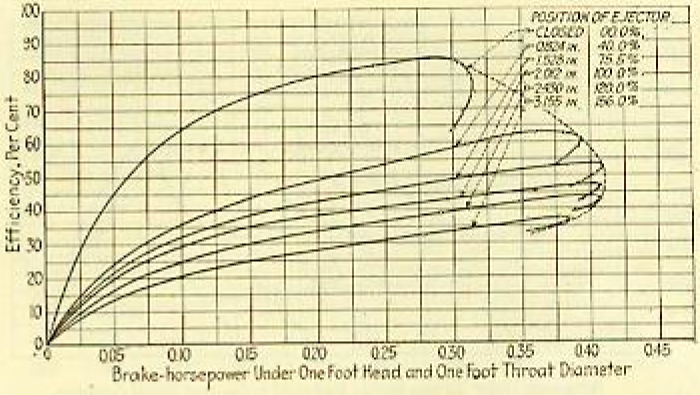

Fig. 6 Curves Showing Operation at Normal ϕ with Varying Ejector Openings and Ejector Vanes in Place; Runner No. 55

crease over the rated output is obtained by the use of the ejector. This increase is 36.7 per cent based on rating of 0.300 hp. under 1 ft. head and 1 ft. throat diameter for runner No. 55 and 30.0 per cent

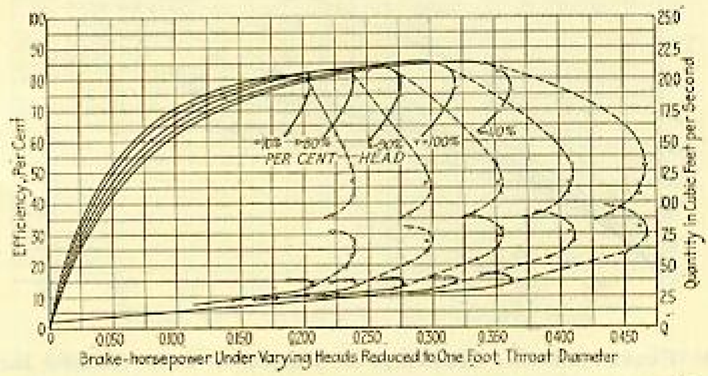

Fig. 7 Curves of Horsepower, Efficiency and Quantity for Runner No. 55 Under Varying Heads

Normal Operation Indicated by Solid Lines and Operation with Ejector by Broken Lines.

for runner No. 65 equipped with vanes in the ejector. Based on the maximum power obtainable these values are 34.5 per cent for runner No. 55 and 17.0 per cent for runner No. 65.

35 A second set of curves (Figs. 7 and 10) was plotted showing

the performance of the turbine and ejector under varying heads, each curve consisting of two parts, the normal performance curve and the envelope of the several ejector performance curves. Curves

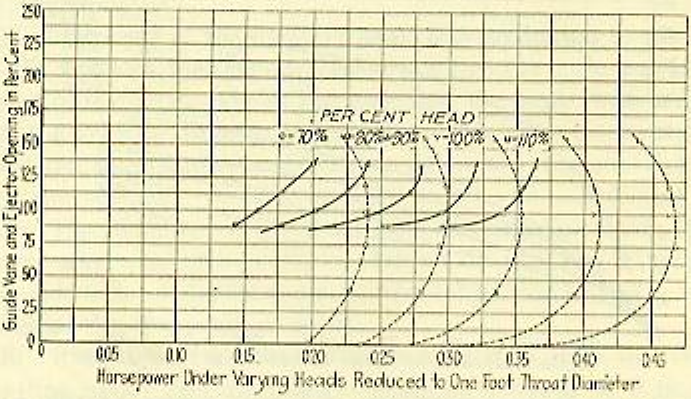

FIG. 8 CURVES OF HORSEPOWER FOR VARYING HEADS FOR DIFFERENT GATE (SOLID LINES) AND EJECTOR (BROKEN) OPENINGS, RUNNER NO. 55.

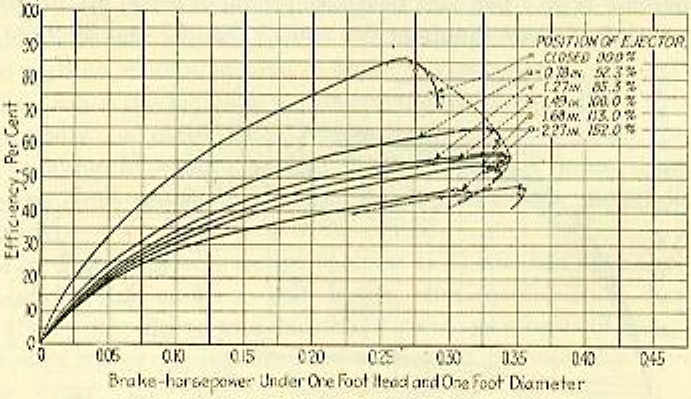

FIG. 9 CURVES SHOWING OPERATION AT NORMAL ϕ WITH VARYING EJECTOR OPENINGS AND EJECTOR VANES IN PLACE; RUNNER NO. 65; ONE CURVE WITH VANES REMOVED

are plotted for 70, 80, 90, 100 and 110 per cent of normal head, the latter being taken as 1 ft. to have a standard for a basis of comparison. Along with these curves have also been plotted curves of horsepower against quantity.

36 Guide vane and ejector openings have also been plotted

against horsepower and appear as Figs. 8 and 11. In all of these results there has been no allowance made for the increased frictional losses in the water passages leading up to the turbine, since the

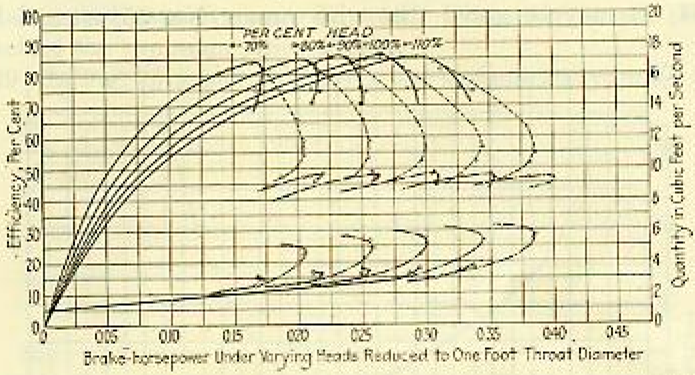

Fig. 10 Curves of Horsepower, Efficiency and Quantity for Runner No. 65 Under Varying Heads

Normal Operation Indicated by Solid Lines and Operation with Ejector by Broken Lines.

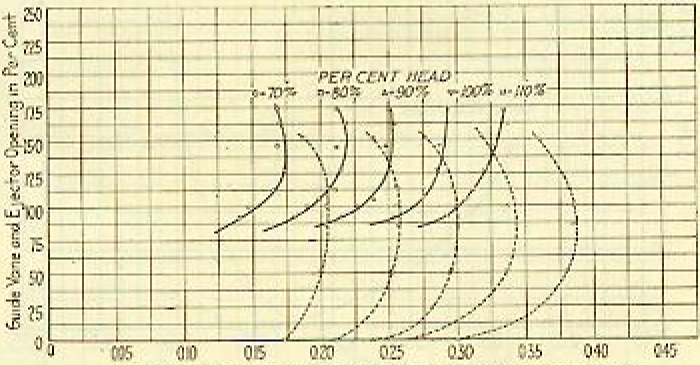

Fig. 11 Curves of Horsepower for Varying Heads for Different Gate (solid lines) and Ejector (broken lines) Openings, Runner No. 65

model tested was of the open-flume type. Therefore the losses resulting from the increase in quantity flowing must be calculated for each individual case where the ejector is to be applied. It should be noted that the results include some of the losses, namely, the losses within the turbine and the losses in the draft tube, but not the losses that would occur in the penstock or the casing. It is probable that these will not be excessively large, since low-head installations have relatively short penstocks which are usually integral with the dam.

37 It may be noted that with runner No. 55, normal rated

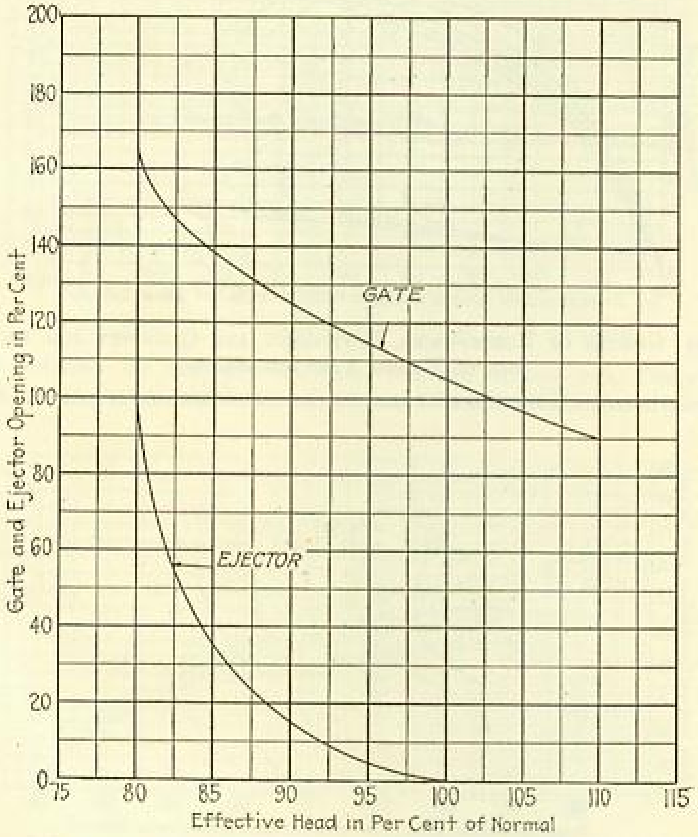

Fig. 12 Constant-horsepower Curves, Runner No. 55

horsepower can be maintained under 80 per cent head and under 83 per cent head with runner No. 65, when both are equipped with guide vanes in the ejector.

38 From the curves of horsepower, gate and ejector opening, another series of curves was obtained (Figs. 12 and 13). These might be called very appropriately "Constant-Horsepower Curves." The ordinates are per cent ejector or gate opening and the abscis-

sae are per cent normal head. If, therefore, the per cent of normal head under which the turbine is operating is known, by finding the points where the gate and ejector opening curves cross this line, the proper setting of the turbine and ejector gates that will maintain rated capacity may be read. These curves are plotted for each of the two main series of tests.

39 In Fig. 10 (curves of horsepower and efficiency under vary-

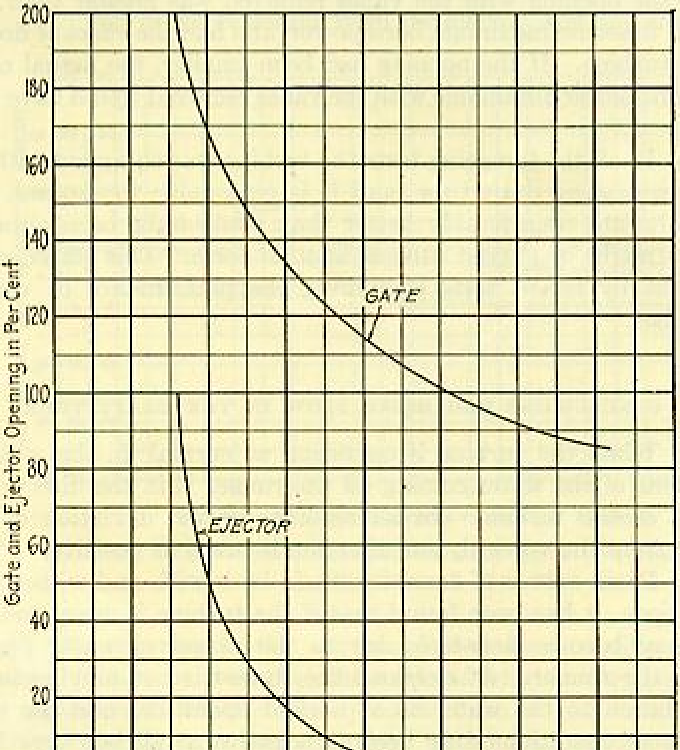

Fig. 13 Constant-horsepower Curves, Runner No. 65

ing heads for runner No. 65) a set of small curves, one for each 10 per cent of normal head from 70 to 110 per cent, are drawn for comparison with the other performance curves. These are the results of test No. C1 (runner No. 65 with ejector vanes removed), and it may be seen that there is a slight numerical increase in horsepower over that obtained with the vanes in place, but judging from the position of these curves with relation to the others and to the shape of the envelope of the curves of varying ejector openings, it is reasonable to suppose that the maximum increase obtainable with the ejector vanes removed is much greater than the actual numerical difference would indicate. It would seem that the turbine was operating on the drowned part of the envelope.

40 The explanation of this is probably that the increase in area of the opening with the vanes removed was greater than that required to secure maximum horsepower, and had the effect of drowning the turbine. If the opening had been smaller, the actual maximum horsepower obtainable with the vanes removed would have been secured.

41 In all the foregoing tests the turbine was equipped with the Moody spreading draft tube, and it is reasonable to suppose that the results are considerably better than could have been obtained with a straight or curved tube of another form. This statement is borne out by recent tests comparing the performance of various draft tubes.

CONCLUSIONS REGARDING FLOW IN THE DRAFT TUBE

42 When the turbine is operating at normal ϕ, the whirling component of the water coming off the runner is in the direction of rotation of the turbine. In all readings of the deviation of this velocity from the vertical, this direction is taken as positive.

43 From curves of traverses made both with and without the ejector open, it has been found that if the turbine is overgated this whirl may become negative, due to the excess of water passing through the runner. At overload the runner is not moving in the same relation to the water as at normal operation, and the whirl under these conditions may become negative at all but very large values of ϕ. When the turbine is operating at high values of ϕ, the whirl is positive, or nearly so, even though the turbine is overgated.

44 From comparisons of traverses made with and without the ejector open, the following general conclusions may be drawn (see Figs. 14 and 15):

1 The characteristic form of the curve of the angle of deviation from the vertical was not influenced by the water from the ejector
2 The velocity of the water at the outer portion of the draft

tube was increased by the action of the ejector water over values obtained without the ejector open

3 The ejector had little or no effect upon the flow of water coming from the runner up to and including a point about 6¼ in. from the center line of the draft tube, or in other words, the inner two-thirds of the draft tube was not affected by the ejector.

4 The action of the runner influenced the direction of the flow of water issuing from the ejector, and not vice versa.

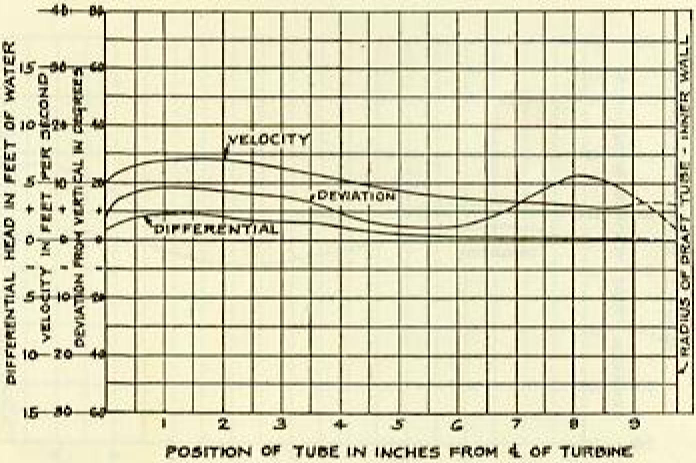

Fig. 14 Flow in Draft Tube, Ejector Closel; Gate Opening 40 Per Cent; ϕ = 1.262; Runner No. 65

45 By referring to the curves plotted from results of these traverses, it may be seen that the angle of whirl is very nearly the same regardless of whether the ejecor is open or not, but that the velocity of the water in the outer third of the draft tube is increased considerably over that with the ejector closed.

SUMMARY

46 As was stated in the introduction, the need for the ejector type of turbine is felt only in low-head installations where the reduction in head due to flood periods reduces the capacity of the plant to an appreciable extent.

47 It is obvious that instead of wasting the excess water that

would ordinarily pass over the dam during these periods, the ejector uses this same water to counteract the reduction of the effective head and to maintain the rated output of the plant.

48 While it is imperative that the turbine operate at maximum efficiency when the head is a maximum and water is scarce, it is also true that economical operation is secondary to the maintenance of rated output when there is so great an overabundance of water that the capacity is reduced.

49 The Moody ejector turbine meets these requirements, for

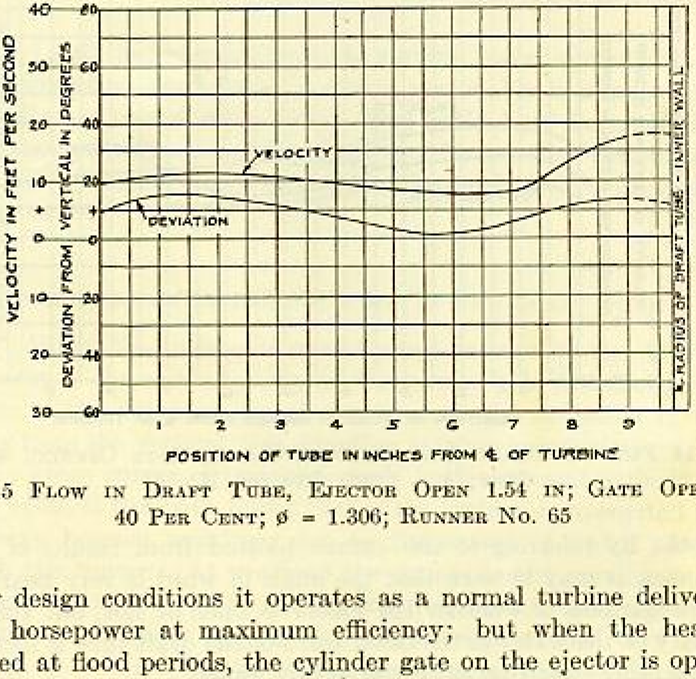

FIG. 15 FLOW IN DRAFT TUBE, EJECTOR OPEN 1.54 IN; GATE OPENING 40 PER CENT; ø = 1.306; RUNNER NO. 65

under design conditions it operates as a normal turbine delivering rated horsepower at maximum efficiency; but when the head is reduced at flood periods, the cylinder gate on the ejector is opened and the output is maintained.

50 This turbine also finds its application, where stream flow and storage conditions permit, in carrying peak loads. At this time the ejector would have the effect as illustrated in the curves of horsepower and efficiency under constant head (Figs. 6 and 9). The maximum possible horsepower that can be obtained is that indicated by the envelope of the several ejector curves. The turbines would therefore be designed to deliver rated power for the average load and rely upon the ejector to carry the peaks. The application

of the ejector to these conditions could only be effected where the peak loads could be accurately estimated in advance.

51 In conclusion, the Moody ejector turbine finds its principal application in low-head installations where the effective head is reduced by flood conditions and great fluctuations occur in the stream flow between high and low water.