out. There is no indication of cavitation on these discharge edges as ordinarily characterized by pitting. The area affected and depth of the chipping are indicated on the Unit 1 runner inspection report.

"Unit 2. The Unit 2 discharge edges were modified in the field by chipping and grinding a groove in them. There is some cavitation in this groove in small patches. These patches were on the side of the groove next to the back of the vane as indicated on the runner inspection report. The amount of cavitated area in the groove was a small part of the total area of the groove and appeared to be caused by irregular surfaces resulting from the chipping operations.

"Unit 3. The Unit 3 discharge edge was modified in the field by chipping and grinding to a bullet shape. At the time of this inspection the Unit 3 discharge edges were in very good condition. The edges had been painted with red priming paint after they were modified. Some of the paint has come off but the edges are still very smooth, showing no indication of cavitation pitting.

"As a result of the observations made during these runner inspections we would summarize this report as follows:

"The three units are all fairly smooth running with Unit 1 being a little rougher than the others.

"The discharge edges of the Unit 3 runner indicate at this time that it is giving the least trouble, with Unit 2 showing a little cavitation in the grooves in the discharge edges and Unit 1 definitely showing some raveling of the discharge edge."

Edge configuration No. 10, Fig. 1, was also applied to two runners at Whitney Dam, Texas. These units when they were first started had a vibration at the point of best efficiency. All traces of this vibration were removed by the alterations to the edges.

The investigations described in this paper did not extend to a determination of how the various edge configurations act on the von Karman trail to affect the exciting forces associated with it. Such a study would be very interesting and might lead to other equally effective edge shapes.

Discussion

D. C. Hazen[2] and C. P. Kittredge.[3] Vibrations of turbine runners which are induced by von Karman vortex trails behind the discharge edges of the buckets can be eliminated if the vortex trail can be reduced sufficiently or removed altogether. The vortex trail will be eliminated if a vortex can be stabilized along the discharge edge of the bucket.

The Department of Aeronautical Engineering, Princeton University, has undertaken preliminary investigation of the flow phenomena around a cusp-shaped profile tentatively designed as the "suction vortex airfoil." Fig. 5 shows a typical profile. A

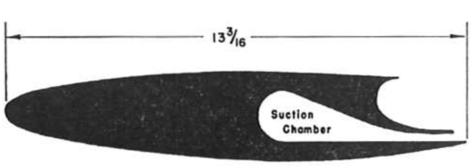

Fig. 5 Suction Vortex Wing

vortex is stabilized in the cusp by means of suction applied through a slot along the lower trailing edge. No vortex trail appears behind the profile as long as the vortex is stabilized within the cusp. Fig. 6 shows the profile in a wind tunnel with streamlines made visible by smoke. The suction is slightly less than that required

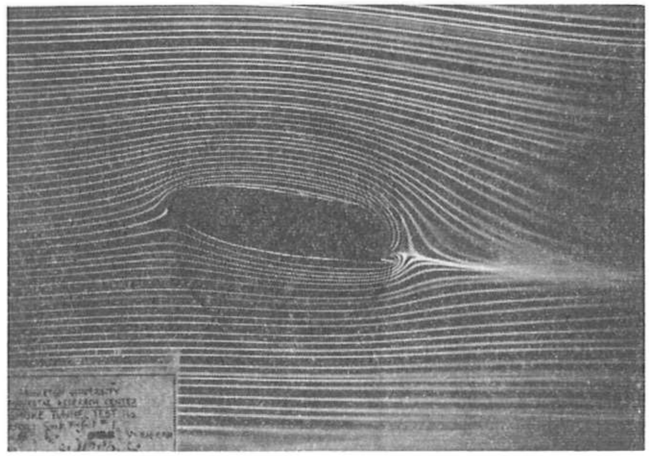

Fig. 6 Flow Around Suction Vortex Wing

to stabilize a strong vortex in the cusp. Experiments have indicated that the vortex can be stabilized under some circumstances by a component of the fluid velocity parallel to the trailing edge in which case suction is not required. It is emphasized that the research has not progressed far enough to permit statements regarding possible size effects or the magnitude of the velocity parallel to the trailing edge which would be required to stabilize a vortex.

The relative flow through a mixed-flow runner is of a type that might provide adequate velocity components parallel to the discharge edges of the buckets to stabilize a vortex along all or most of each discharge edge. A suitable cusp along the back of each bucket at the discharge edge would be required. Stabilization of a vortex along the discharge edge of each bucket should improve the exit flow conditions and eliminate one source of vibration.

F. E. Jaski.[4] This paper is an interesting description of a new method of correcting bucket vibration in Francis-turbine runners. The type of sharpening of the discharge edge on the back side of the bucket as shown by item 10, Fig. 1 of the paper, which was the most effective in stopping the vibration, appears to have an influence on the exciting forces causing the buckets to vibrate. As shown in the figure, the sharpening and rounding is carried back from the discharge edge along the bucket a distance of 1.8 times the thickness of the edge. If the vortexes in the von Karman trail coming off the face and back of the vane are in staggered formation it is possible that this starts them off on the back side a little ahead of those on the face of the bucket. By starting the sharpening on the back side at the most effective distance it may be possible that the two sets of vortexes quench each other with a result that there is no side force left to excite the bucket into vibration. The bucket vibration is influenced by its natural frequency and its relation to the frequency of the exciting forces. If the two are in resonance critical vibration may occur. Thus a possible remedy would be to separate them sufficiently so they will not be in resonance, or destroy the exciting forces such as perhaps occurs in the remedy used by the author.

This type of vibration is not limited to Francis-type runners, but also may occur in propeller runners. The writer had experience in correcting blade vibration on both Francis and propeller-type runners while he was employed by the author's company and believes it may be of interest to relate the remedy used to correct vibration in those applications. The first experience was with the 66,000-hp 165-ft-head units at Norris Dam of the Tennessee Valley Authority. In the original design the discharge

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[3] Associate Professor of Mechanical Engineering, Princeton University, Princeton, N. J. Mem. ASME.

[4] Engineer-in-Charge, Pump Turbines, Allis-Chalmers Manufacturing Company, Milwaukee, Wis. Mem. ASME.

edge of these buckets was rounded with about a $^3/_8$-in. radius and the back side was tapered off at the edge for about 5–6 in. These runners had a singing note that could be heard in the turbine pit. On these runners permanent steel struts about 3 in. wide and tapered from $1^1/_2$ to $^3/_4$ in. in thickness were welded between all the buckets to correct the vibration. They were located on the mean flow line and about midway between entrance and discharge of the bucket. These struts did not affect the output of the units. A duplicate of this runner was later used at the Hiwassee Power Plant for a rating of 80,000 hp under 190 ft head. In order to correct the vibration, the buckets for Hiwassee were made about $^1/_2$ in. thicker across the entire back side at the top and tapered off to the original thickness about at the center line of the distributor. The discharge edge was made square about $^3/_4$ in. thick and gradually tapered on the back side for a long faired surface. No vibration occurred in this runner. The specific speed is about 48.

At Claytor Dam of the Appalachian Electric Power Company vibration occurred in two Francis turbines rated about 26,000 hp under 110 ft head. The bucket vibration occurred at about 62 per cent gate opening. The specific speed is about 68. These units have 16 buckets and 16 wicket gates. However, the vibration was not caused by this combination of wicket gates to runner buckets but was caused by the von Karman trail at the discharge edge. The vibration could be felt in the generator floor. In this case the original design had a square edge about $1^1/_8$–$1^1/_4$ in. thick. This thickness of the edge was reduced to about $^{11}/_{16}$ in. by chipping off metal on the face or pressure side and the surface was then faired to a long taper. This eliminated the vibration and increased the output of the units. While the chipping was in progress on the first runner it was necessary to put the unit in service because of a flood on the New River in the summer of 1940. Only 12 of the 16 buckets had been chipped and were still rough before they were ground smooth, but when the unit was put on load there was no vibration at any gate opening.

At Drop 4 Power Plant of the Imperial Irrigation District, about 1942, vibration occurred in a fixed-blade-propeller turbine rated 13,300 hp at 51 ft head. This is a 5-bladed propeller with a specific speed of about 127. This unit had vibration periods at 28, 42, and 63 per cent gate opening. It was so severe that it caused the railings to rattle outside the powerhouse. The remedy used here was to reduce the thickness of the discharge edge from about $1^1/_4$ to $^3/_4$ in. for about $^2/_3$ of the length from the outer circumference toward the hub. The chipping was done on the face of the blade and then faired to a long gradual taper. After the blades were ground smooth the unit was put back in service and there was no vibration.

At Grand Coulee Dam the first three units L1, L2, and L3, developed vibration at about 60–65 per cent gate opening. These units are rated 150,000 hp at 330 ft head with a specific speed of about 34. The vibration could be felt on the top cover plate and railings in the turbine pit. The remedy used here was to reduce the thickness of the discharge edge from $1^1/_4$ in. down to about $^{11}/_{16}$ in. The chipping was done on the face of the bucket and the surface was faired to a long gradual taper and ground smooth. The vibration was eliminated and the output increased about 5000 kw. The increase in output at Grand Coulee and Claytor Dam occurred because the discharge openings of the buckets were slightly increased by the chipping of the face.

The Bureau of Reclamation also experienced vibration on the units at Parker Dam which resulted in cracks in some of the runner blades as described in a paper by John Parmakian.[5] These units are rated 40,000 hp at 80 ft head. Here the final remedy was obtained by chipping the discharge edge to about $^3/_4$ in. and a gradual taper on the face of the bucket. The output of the unit was also increased about $6^1/_2$ per cent.

In the units in which vibration was corrected by the writer the thickness at the discharge edge was reduced to about 55 per cent of the original thickness. At the same time the openings between the buckets were slightly increased. If the frequency of the exciting forces caused by vortexes at the discharge edge varies with velocity divided by thickness V/T, then by reducing T to about 55 per cent and increasing V about 5 per cent, the exciting frequency would be almost doubled. It is possible that in each of these cases this carried the exciting forces out of resonance with the natural frequency of the buckets and thus reduced the vibration so it was no longer critical.

The method of sharpening the discharge edge of the bucket proposed in the paper is a rather simple process and suggests that the bucket edges could be finished in this manner in the shop before the runner is shipped to the job.

G. D. Johnson.[6] Naturally, this subject is of interest to all manufacturers and users of hydraulic turbines. A paper[5] by Messrs. Parmakian and Jacobson dealt primarily with runners furnished to the U. S. Bureau of Reclamation by the writer's company. Consequently, we are particularly interested in this further discussion of the problem.

Theoretically, the bucket-discharge edge should have zero thickness (i.e., a knife-edge), but a fairly thick edge is required to obtain a satisfactory casting. Gratifying results were obtained at the Parker and Keswick Plants of the U. S. Bureau of Reclamation by thinning the discharge edges of the turbine-runner buckets to approximately $^3/_4$ in. (0.004 $\times$ runner discharge diameter). The only disadvantage of this obvious solution is the expense of chipping and grinding to fair the thinned edge shape gradually into the upstream bucket contours.

Presumably, the main attraction of the various edge shapes shown in the author's Fig. 1 is the small amount of metal that must be removed from the bucket edges as cast. It is difficult to explain the relative vibration amplitudes shown by Figs. 3 and 4, especially in the case of configuration No. 11 with no thinning and just a groove along the discharge edge, as was employed on Unit 2 at Canyon Ferry. The fact that it proved to be satisfactory in that particular case is not too significant, since Unit 1 required no modification at all. Referring again to Figs. 3 and 4, it is extremely difficult to explain the high vibration amplitude of configuration No. 5 relative to No. 1, although it seems plausible that configuration No. 8 (as well as Nos. 9 and 10) would have a smaller amplitude.

At the Denison Dam Plant (in Texas) of the U. S. Army Corps of Engineers, audible bucket vibration was satisfactorily eliminated at minimum cost on the 192-in. discharge diameter runner of Unit 2 by welding a hoop of $1^1/_2$-in-diam stainless-steel bar stock to each bucket near the middle of the trailing edge. Struts were used between all of the buckets of Unit 1, but the ring in Unit 2 has proved to be just as durable and effective.

It is also interesting to note that, although vibration pickups and analyzers give more complete information, the existence of definite runner-bucket vibration is evident to the human ear as a loud and disturbing hum, so that elimination of the noise may be construed as satisfactory treatment of the problem. In this connection it is also worth mentioning that a small amount of compressed air released upstream of the runner buckets effectively eliminated the noise and the measured vibration amplitude on all three runners at Keswick, before thinning the discharge edges. The beneficial effect of compressed air has also been demonstrated

[5] "Measurement of Hydraulic Turbine Vibration," by J. Parmakian and R. S. Jacobson, Trans. ASME, vol. 74, 1952, pp. 733–741.

[6] Chief Hydraulic Engineer, S. Morgan Smith Company, York, Pa. Mem. ASME.

in the case of vane vibration resulting from high velocities in a large 90 deg vaned elbow.

Another significant observation is that we have not been able to establish definitely any connection between runner-bucket vibration and cracking of the runner buckets where the trailing edges join the runner crown or band. At Denison, there was definite bucket vibration without any evidence of cracking whatsoever; at Parker and Keswick, there was definite bucket vibration as well as fairly extensive cracking where the bucket-discharge edges join the runner crown. At several other plants there was no observed bucket vibration but cracking of the discharge edges near the crown was rather extensive.

Comments on these points would be appreciated.

W. J. Rheingans.[7] The runner-blade vibrations described by the author are not a recent development, although probably due to a trend in recent years to larger physical-size runners, vibration problems of this type have been on the increase. While blade vibration is usually associated with low-head units of large size, such vibrations have occurred on large-size units under heads as high as 330 ft.

The writer's first experience with runner-blade vibration was in 1930 on a 6500-hp, 48-ft-head, 125-rpm Francis turbine at the Upper Notch Power Plant, Canada. Loud singing noises were present for narrow gate ranges at several different gate positions. After locating the source of the vibrations, they were completely eliminated by placing streamlined struts between the runner blades. These struts have been in place for 25 years and have shown no cavitation or other detrimental effects. The vibration was eliminated before the runner had been in operation long enough to produce cracks, and no cracks have developed in the 25 years of operation with the struts installed.

The writer had no further experience with this type of vibration until quite recently. In 1930 six Francis units were installed at the Osage Power Plant, rated 35,400 hp, 90 ft head, 112.5 rpm. There was no evidence at that time, nor during subsequent 25 years of operation of any serious vibration. In 1953 two additional units of exactly the same design were installed. One unit was equipped with the spare runner which had been furnished with the original six units in 1930, and which had never been used. This unit with the spare runner showed no evidence of blade vibration. The runner for the second unit, installed in 1953, was made from the same pattern that had been used for the original runners. However, the pattern required some repair work, and it is possible that the discharge edges of the runner blades were somewhat thicker. This last unit showed considerable vibration. The vibration was not serious enough to prevent operation, but was quite annoying to the operating personnel in certain locations in the powerhouse. An inspection of the runner showed no blade cracks, but it was decided to groove the discharge edges of the runner blades similar to shape 11, Fig. 1 of the author's paper.

This modification showed a remarkable improvement in the operation of the unit, which was evident as soon as it was placed in operation. The high-frequency vibration had disappeared completely. The experience at this plant shows how sensitive runner-blade vibration apparently is to small variations in runner castings.

Runner-blade vibration is not always accompanied by unusually loud singing noises, and the physical evidence of the vibration may be slight. This is probably due to the absence of any other parts of the unit or powerhouse structure with a resonant vibration frequency to pick up the blade vibration. However, the actual blade vibration can be serious enough to eventually cause runner-blade cracking.

[7] Manager, Hydraulic Department, Allis-Chalmers Manufacturing Company, Milwaukee, Wis. Mem. ASME.

A situation of this type occurred at the Wilson Power Plant of the T.V.A. Four units rated 35,000 hp, 92 ft head, 100 rpm were installed in 1942, two in 1943, one in 1949, and three in 1950.

The six units installed in 1942 and 1943 showed some slight cavitation and were repaired by welding in 1948. Shortly thereafter, cracks appeared in the runner blades. These were repaired by welding in 1950, but kept on reappearing. The units had never produced any loud singing noises, nor any unusual vibrations. However, a careful investigation showed that there was a slight high-frequency vibration at full gate which could be produced by runner-blade vibration. Accordingly, struts were inserted between the runner blades. This eliminated the vibration at full load and stopped any further cracking of the runner blades.

Thus, as stated, runner-blade vibration is not a recent problem. However, up to the time of the investigations described by the author, the remedy has been either the installation of struts between the runner blades, or trimming down the thickness of the discharge edges. Therefore the results obtained by the author with varying shapes of trailing edges are a decided contribution to the art. His investigations comprise the first published data for this type of approach to the problem.

Further research along these lines is indicated, to determine exactly how the von Karman trails perform and how they produce the forces to cause blade vibration.

G. J. Vencill.[8] This type of paper is of great value to an operating organization in need of a solution to a problem. There is sometimes a tendency to pass over reports of trouble as subjects for technical papers, and while such matters are often freely discussed, they may not be made available for general reference.

When our organization was faced with a turbine-vibration problem recently, we found the literature on the subject exceedingly sparse. At the Osage Plant vibration developed in two new units when they were placed in service in 1953. Six original units which had been operating since 1931 had given no trouble. Unlike the case described in the paper no "loud singing noise" was observed, but vibration was felt and the noise was a loud grinding noise with a pulsating characteristic corresponding to each revolution of the shaft. The noise occurred in two ranges of gate opening between 0.65 and full gate with a relatively quiet spot between. It is doubtful that an analysis of the noise would have shown any predominant frequency. The noise was not noticeable with low tailwater levels, but increased with rising tailwater and reached a maximum when the level approached the bottom of the turbine runner. The same characteristic was observed at Keswick Power Plant mentioned in the discussion of the paper by Parmakian and Jacobson.[5]

Shaping the trailing edges of the runner buckets similar to configuration No. 11 in Fig. 1 of the paper eliminated the noise and vibration.

It was observed while the work was being done that a corner of the trailing edge was being pitted and that in places it had been beveled as much as $^1/_4$ in. If this action continued, the shape of the edge would approach that of configuration No. 10, and might have eliminated the vibration without assistance in due course.

Studies of this kind are carried out much more readily in the laboratory and in the shop than after a turbine has been placed in operation. It is to be hoped that a full understanding of vibration can be reached which will enable these troublesome cases to be eliminated in the design stage, and not become problems in operating units. This paper is evidence of progress being made.

[8] Union Electric Company of Missouri, St. Louis, Mo.

Author's Closure

One explanation that has been advanced for the effectiveness of configuration No. 11 is that a vortex was stabilized in the cusp formed in the discharge edge. The experiments described by Professors Hazen and Kittredge lend credibility to this theory.

As suggested by Mr. Rheingans, many cases of runner-blade vibration may exist that are not evidenced by noise or vibration of the accessible parts of the units. Such cases may or may not be severe enough to cause eventual cracking of the runner blades. Conversely, it is possible that some of the noisiest cases of blade vibration are quite harmless.

Several of the discussions mentioned the successful use of struts and hoops to suppress runner-blade vibration. Such cures, which are often effective, do not remove or reduce the periodic exciting forces, but merely limit the amplitude of blade vibration. One objection to this type of cure is that there may be significant drag forces associated with a fully developed von Karman trail. These drag forces could cause an appreciable power loss which would be unaffected by the use of struts or hoops, but which might be eliminated by a change in edge shape. The author concurs with Mr. Rheingans that further research on trailing-edge phenomena is desirable.