CAVITATION PROBLEMS AND THEIR EFFECTS UPON THE DESIGN OF HYDRAULIC TURBINES

by

J. M. Mousson

Rustless Iron and Steel Corporation

Baltimore, Maryland

INTRODUCTION

At the present time, a new peak in hydraulic-turbine production has been reached. The key note is unit capacity, rather than hydraulic or mechanical perfection. It should be emphasized, however, that units recently installed or now in process of manufacture may be expected to have also better performance characteristics than those attained but a few years ago. Progress of propeller turbines in particular has been stimulated through the successful exploitation of many low-head sites, previously thought undesirable from an economic point of view. Since 1931, when axial-flow turbines of high capacity were first introduced in this country, units totaling more than 2 million horse power have been built by American manufacturers. With this rate of production as an incentive for extensive laboratory practice, a stage in development has been reached at which the limitations previously imposed by general cavitation are no longer as critical. While but a few years ago cavitation made a compromise between efficiency and capacity inevitable, research has provided the essential basis for improvement to such an extent that this once-typical characteristic may no longer be regarded as an inherent weakness.

Since the limits of general cavitation, and the laboratory procedures for their determination, have been discussed only recently [1], the object of this paper is to show the progress made in turbine design, construction, operation, and maintenance to eliminate, or at least minimize, cavitation phenomena and their varied, detrimental effects.

1 References appear at the end of the article

146
CAVITATION AND TURBINE DESIGN FEATURES

General Trend in Propeller-Turbine Construction

While at the outset the higher specific-speed characteristics of propeller turbines were exploited to the limit, with a consequent tendency to over-speed the units, a more conservative policy is now being pursued. The compactness of blade-operating mechanisms for units of the movable-blade type has also advanced to such an extent that runners may now be equipped with as many as seven or eight blades. This feature has materially widened the range of head under which these turbines may be operated satisfactorily. With increasing head, the possible number of blades becomes a decisive factor, as more adequate overlapping must be provided to hold the water to the suction faces of the blades. Design characteristics now generally accepted are summarized in Table I.

<table>
<thead>
<tr>
<th>Typical Design Characteristics of Propeller Turbines</th>
</tr>
</thead>
<tbody>
<tr>
<td>Operating Head ft.</td>
</tr>
<tr>
<td>---------------------</td>
</tr>
<tr>
<td>10-65</td>
</tr>
<tr>
<td>70-115</td>
</tr>
<tr>
<td>150-180</td>
</tr>
</tbody>
</table>

For the typical range of low heads, it is usually found that with three-bladed propeller runners some sacrifice in efficiency is necessary. Furthermore, three-bladed runners invariably show a higher cavitation coefficient than units equipped with four or five blades. Hence, also lower settings with respect to tail water are required.

Although units for the highest range of head referred to in Table I have not as yet been installed in this country, they have found repeated application abroad. Due to their success, more serious consideration should be given to propeller units under similar conditions of available head in preference to the conventional Francis turbines with less favorable operating characteristics and larger space requirements for identical unit capacities.

Conduciveness to cavitation has also been diminished by means of improvement in flow conditions immediately ahead of the runners. One of the desirable features, only recently fully recognized, is the possibility of off-setting the center line of intakes from the center line of units. There has been considerable reluctance to
adopt this practice due to the inevitable increase in construction costs imposed by more complicated contraction joints and concrete forms. However, since with adequate dimensioning of the offset unfavorable vortex conditions in the intakes can be avoided through a more complete development of the scroll, the higher costs can be justified due to the possible gain in efficiency over the entire load range and the decrease in the tendency to cavitate when operating at high capacity.

Recent developments in the design of speed rings and wicket-gate mechanisms are noteworthy. Flow obstructions have been reduced to a minimum by decreasing the number of stay vanes to half that of the guide vanes. Furthermore, both of these vanes are no longer designed with straight longitudinal axes, but are curved in plan to impart the entrance whirl with a minimum of turbulence. The formation of large unstable low-pressure zones on one side of these vanes may thus be prevented, or at least reduced in size, and the danger from vortex trains originating in these zones and aggravating general and local cavitation conditions becomes less acute.

Another improvement in design provides for less overhang of the wicket gates. By “overhang” is meant the distance between the downstream edge of the wicket gates and that point where the lower distributor surface tends to fall away from the bottom of these gates. Although this feature necessitates wider and flatter distributor rings, it gives the advantage of eliminating cavitation in the wake of the wicket gates when in wide-open position. Specifications, now being prepared for one of the largest low-head plants ever to be built, contemplate a design with no overhang of wicket gates whatever.

Most important, of course, are developments in runner design. The adoption of conical hubs benefited efficiency as well as cavitation characteristics. Attempts to eliminate leading-edge cavitation have also been successful. This phenomenon has been troublesome, as it caused pitting on the pressure and suction faces of the turbine blades immediately downstream from the leading edges. These edges are particularly conducive to cavitation, as they constitute the uppermost portion of the rotating element, and consequently the cavitation coefficient applicable to these locations is bound to be more critical as compared with that of the installation as a whole. In addition, the proper selection of the blade angle at the entrance
to the runner is of paramount importance in obtaining a minimum of shock. Experience has also shown that the leading edges must be carefully shaped to avoid cavitation. Blade noses found suitable under service conditions are shown in Fig. 1.

Experience has taught that reversals or discontinuities in radial blade sections are liable to cause serious pitting, despite the fact that these zones may be only of limited extent. While the first high-capacity units of the adjustable-blade type manufactured in this country were particularly subject to this weakness, the progress made in the understanding of this problem has been helpful. More careful shaping of blades to conform to those of the pattern in every detail has been stimulated through the introduction of machined blades. One American manufacturer has adopted the Keller type of milling equipment for this purpose, which handles one blade at a time (see Fig. 2). Profiling machines of the boring-mill type with horizontal turntables and accommodating four or five blades simultaneously are employed abroad. The availability of machining equipment to certain manufacturers made others refine their

---

Photo, courtesy S. Morgan Smith Co.

**Fig. 2.—Keller-Type Profile Machine.**

**Fig. 1.—Suitable Profiles for Blade Noses.**
shop methods through closer spacing of templates to keep a more accurate check on the blade profiles. Although machined blades have been widely used abroad, nevertheless there are not as yet sufficient data available to fully justify their adoption.

In some instances, shroud bands on runner blades have been introduced to eliminate or at least minimize pitting caused by gap cavitation [2] (see Fig. 3). For the same purpose, clearances between runner-blade peripheries and throat rings have been materially reduced. It should be recognized, however, that there is a definite desirable minimum of clearance, below which one may not dare to go. On the other hand, there are no restrictions imposed on the depth development of the peripheral shroud which could prevent full effectiveness. The meager comparative data at hand would make a decision as to the preferable solution premature.

Rational Approach to Further Improvement in Propeller Turbines

In view of the substantial progress made, additional improvements become more and more difficult. This has been fully realized by the industry, as is apparent from the various laboratory procedures now being employed to achieve further refinement. Some of these methods are extremely cumbersome and expensive, as, for instance, the testing with painted runner blades to detect local cavitation zones through paint removal, visual examination by means of stroboscopes, and spark photography. There is serious doubt, moreover, as to the merit of these methods. Unless cavitation is active or visible, the results are bound to be negative, despite the fact that the pressure distribution may not be uniform and far from ideal. Therefore, a more rational approach which may give promise of useful results, even though the pressure at any part of the turbine has not been lowered below the vapor pressure, is
deemed essential. In other words, it is no longer a question of only eliminating cavitation, but turbine designs must be refined to such an extent as to avoid low-pressure zones out of line with the desired average pressure distribution.

In view of the above, nothing short of actual measurement of the pressure on the various parts of a model turbine in operation can give promise of a satisfactory solution. Past experience has demonstrated that hydraulic model tests are out of the question, because piezometer readings would not be of sufficient accuracy. The only possible approach, therefore, is aerodynamic testing. This method has been employed with great success by one manufacturer for the detailed study of individual turbine parts [3] set in a wind tunnel and also by actually substituting air for water to operate turbine models [4, 5]. As air is compressible, it is essential to evaluate any possible effect upon the pressure readings and the reactions on the blades. Analysis shows that for an air velocity of 300 feet per second the density differs less than 4 percent from that of motionless air. This would indicate that corrections on pressure readings will be small, provided actual and relative velocities are kept within reasonable limits. Regarding the reaction of the turbine blades, it can also be demonstrated that at the same air velocity of 300 feet per second the corrections required to bring results on the basis of an incompressible fluid will not exceed 3 to 4 percent.

Model testing of any kind is, of course, predicated on the assumption that similitude prevails between the laboratory apparatus and the prototype installation. This prerequisite must be fulfilled by the mechanical parts and the flow passages, and also by the characteristics of the flow itself. In consequence, the ratio of the Reynolds numbers for model and prototype installations should equal, or at least closely approach, unity. It is, therefore, interesting to note that this requirement can also be met in using air for propulsion of models instead of water.

Two general arrangements suitable for aerodynamic turbine testing are shown in Fig. 4. While for one of these air is sucked through the turbine, it is pressed through the unit in the other where a scroll case is installed ahead of the speed ring. In both of these laboratory set-ups, the quantity of air is measured by a calibrated nozzle placed downstream from an equalizing tank. One of
the main advantages of aerodynamic laboratory equipment lies in the small space requirements. Less than half the room is needed when compared to low-head hydraulic model equipment. Furthermore, the easy accessibility of all flow passages and their light construction simplify many problems.

Equipment of this kind lends itself favorably to the exploration of the velocity distribution immediately upstream and downstream from the runners, and, which is more important, to the actual determination of the pressure distribution on the turbine blades under operating conditions. In this connection, it has been found advantageous to place piezometers on radial sections and to assign one section to each blade. If the pressure at any location reaches a value below the vapor pressure, cavitation would actually occur in this region on the prototype. From the aerodynamic tests, the hydrodynamic pressure distribution can be calculated. Whereas for aerodynamic installations there is no limit to the negative pressures, the vapor pressure for hydrodynamic tests sets a minimum. This makes it necessary to cut away the negative pressure peaks indicated in the course of the aerodynamic tests.

As favorable results have been obtained with wind-tunnel experiments and air-propelled model turbines, it is believed that this method should now be more generally considered to assure further progress in the design of hydraulic prime movers.

**Air Injection to Improve Turbine Operation**

*Air Injection to Dampen Vibration Due to Cavitation*

Operating experience with propeller units of the fixed- and adjustable-blade type has shown that vibrations may occur in the
range of from near full-load to over-load conditions. The intensity of the observed phenomena depends in a large measure upon the location of the cavitation limits with respect to the physical capacity of the units. With the former well below the capacity limit, vibration is likely to be far more severe. Aside from the shape of the runner blades, other design features are also of importance. For instance, under the prevailing velocity conditions at full load, it is practically impossible for the water to follow the profile of the runner deflector. In consequence, unstable cavities of considerable size may form beneath the runner, and bumping of units may occur due to the collapse of these cavities. Curiously enough, model tests carried out about three years ago at the Holtwood Hydraulic Laboratory indicated that shorter deflectors tend to improve the efficiency at part load—that is, in the very range where peak efficiencies are an all-important issue for propeller units of the adjustable-blade type. Hence, deflector designs, advantageous from an efficiency point of view, may tend to make conditions more conducive to cavitation and the resulting vibration in the maximum-load range.

Vibration may be so severe as to become the limiting criterion for allowable maximum gate setting. At Safe Harbor, for example,
conditions were particularly annoying due to the proximity of the control room to the first group of the initial units, subject to phenomena of this kind when operating at high capacity. As a result, vibration measurements were made at various locations in the power house. A special vibrograph was used for this exploration which recorded the vibration waves in all three planes of a space-coordinate system (Figs. 5 and 6). The turbine gate openings were finally limited to that for which vibrations were thought to be tolerable. It is but natural that this restriction on unit and station capacity led to a detailed study to overcome these limitations.

Tests with varying quantities of air injected through twin openings in unit head covers proved very successful. For this study, an electric vibration-velocity meter was used (Figs. 7 and 8). Keeping the magnitude of allowable vibration the same and injecting air at a rate of 125 cubic feet per minute, a capacity gain

![Records of vibration phenomena](image)

**Fig. 6.—Records of vibration phenomena obtained with Shihadey vibrograph located on turbine head cover and draft-tube bridge.**
in excess of 1,000 kilowatts per unit was indicated (Fig. 9), or a total of more than 6,000 kilowatts for all units then installed. In the light of these results, injection of air was adopted as a standard operating procedure for the range of higher gate openings, and increased maximum-gate limits were put into effect. The unit most recently installed was provided with five equally-spaced openings for air admission through the head cover. There are preliminary indications that better results may be obtained with but two openings located diametrically opposite each other—

Aside from air injection through the head cover, other tests were made with air admitted through the stay vanes. This was made possible by connecting the steam-piping assembly for frazil-ice protection to the 100-pound air-supply system of the station. Holes of 3/16-inch di-
ameter were drilled from the downstream edge of the stay vanes close to the lower distributor plate through into the hollow core of the vanes. Comparative results on the intensities of vibration with and without air injection through the head cover and stay vanes are also shown on Fig. 9. It is interesting to see that in using both facilities simultaneously an additional gain in capacity can be realized.

The latest unit to be installed was also equipped with air-injection facilities through the guide vanes. To bring air to the inside of these vanes, flexible metal hose connects a receptacle mounted on top of each wicket gate with a circular air header above the turbine head cover. Compressed air can be discharged into the water passage through a diagonally drilled hole which leads from the hollow core of each wicket gate to its underside. The air outlet is located downstream from the gate stem. Comparative results for all three types of air injection—that is, through head cover, stay vanes, and guide vanes—are not as yet available.

With vibration phenomena an all-important factor at high gate operation, it was thought advisable to install, at least on one unit, an experimental device to assist in proper loading without exceeding allowable vibration limits under all operating conditions. Since the cavitation coefficient depends on the total head on the unit, as well as on the suction head, the gate limits for constant vibration level must obviously vary with change in available head (Fig. 10). In addition, the intensity of vibration depends upon the water
temperature, due to the marked change in vapor pressure with varying temperatures; hence, the gate limits are also subject to seasonal variations, and may be higher in winter than in summer for identical levels of vibration velocity. Furthermore, there are effects superimposed due to the fluctuations in barometric pressure and the seasonal variations in the dissolved-air content of the water.

The experimental unit-loading apparatus consisted of a remote-indicating vibration-velocity meter installed on one of the unit instrument-board panels in the control room which was connected to a vibration-pick-up device mounted on the draft-tube-manhole door of the same unit (Fig. 11). Under maximum-capacity operating conditions, this apparatus proved of valuable assistance in loading this particular unit correctly to the proper limits in a minimum of time. In contrast, the conventional procedure had to be followed on all other units in finding the correct gate openings indirectly based on the predetermined power output and in using the indicating watt meter.
as a guide. Valuable time was lost, as 15 to 30 minutes elapsed before near-proper gate settings were found through trial and error in averaging readings from the indicating watt meters. Since it was difficult to adjust the loads on the units by this means within 500 kilowatts, hourly integration of unit loadings had to be resorted to. After several hours had passed, and provided the head on the station remained constant, this procedure was successful, though extremely cumbersome.

In view of the various and important operating advantages to be gained with remote-unit vibration-velocity meters, the decision was made to equip, as soon as possible, all units with suitable apparatus of this kind.

**Air Injection to Prevent or Reduce Pitting Due to Cavitation**

Perhaps the first commercial installation to be equipped with air-admission facilities to prevent attack from cavitation was made abroad almost ten years ago on a butterfly valve where severe pitting had occurred downstream from the heaviest section of the valve disk. This pioneer installation proved to be very successful and demonstrated the usefulness of air injection in practice. As laboratory tests have actually confirmed the effectiveness of air [6], measures of this kind may be equally feasible on turbines, provided a carefully planned approach is used. Since the suction faces and the peripheral edges of propeller-type runner blades, as well as the throat ring and draft-tube liners, are the most vulnerable locations for cavitation, air injected through the head cover may likely be of little value for the reduction of pitting. Furthermore, air ejected at the root of stay vanes and guide vanes may help to reduce pitting only in the wake of these vanes. In view of this, it would be worthwhile to consider for new turbine installations an unbroken curtain formed by air escaping from a perforated pipe ring inserted flush with the circular wall section of the water passage near the downstream edge of the lower distributor plate. An installation of this kind may be helpful not only on propeller-type turbines, but also on Francis units where in many cases severe pitting has occurred at the junction of the blade spoons with the runner band.

Up to the present time, no installation for air admission has been made to give protection to the most vulnerable trailing halves
of the blade suction faces of propeller turbines. It is realized that
the mechanical solution may present some difficulties, particularly
for Kaplan units in which the turbine shaft must accommodate the
operating mechanism of the blade. However, for fixed-blade prop-
eller turbines, and also for adjustable-blade units of the Terry
type, air could be fed into the runner through a hollow turbine
shaft and from the runner hub through suitable passages in the
blades to the areas on the suction faces particularly susceptible to
attack by cavitation.

Due to its possible detrimental effect on efficiency, air injection
should not be used without discretion. It can also be expected that
the necessity of air injection may decrease with progress in de-
sign as well as with more extensive use of materials which are
highly resistant to the attack by cavitation.

Cavitation and Metallurgical Aspects

Operating Experience and Research

Although pitting seriously impaired operating characteristics of
Francis turbines in many instances, little was made known to the
engineering profession in general for many years. This policy
can only be explained by the apparent inconsistencies in the ob-
served phenomena. While certain installations showed no signs of
pitting whatever, others of a like design were seriously affected.
In addition, some turbines developed the first signs of damage at
the outset while for some inexplicable reason other units remained
unattacked, only to fall subject to the same destructive disease
after a period of perhaps many years of apparently complete im-
munity. Since the nature of the hydraulic phenomenon and the
metallurgical aspects of pitting were not yet explained, there was
ample reason for the reluctance of the industry to make any state-
ments whatever.

Two factors were responsible for bringing the problem to the
attention of the profession as a whole. As the urgency of relief for
the affected units of the Francis type became more acute with time,
water-power utilities pooled their experiences and made the first at-
ttempt to solve the problem, particularly that of maintenance,
through a joint effort [7, 8]. The other factor coming into play
was the advent of the propeller-type turbine, which is inherently
more conducive to cavitation than a unit of the Francis type. Thus, turbine users and manufacturers alike began seriously to explore the many aspects of the problem.

Perhaps no other field of hydraulics has produced as many different and independent researches in such few years as has cavitation. The variety of approaches and, in some instances, the unique design of testing apparatus are noteworthy. These laboratory equipments included complete model turbines equipped with blades consisting of different materials to be investigated [9]; Venturi [10] or nozzle [11] type cavitation test stands, water sirens [12], apparatus employing plunger or piston arrangements [13] to produce water hammer or pressure waves, rotary disks with specimens mounted on the periphery to cut water jets at high speed [14], and vibratory equipment built with magnetostriction oscillators [15, 16].

The wide variety of laboratory apparatus reflects the underlying thoughts in approaching the problem. While in one instance a solution was attempted by means of turbine models under conditions identical to those of the prototype, a slow and cumbersome procedure, others experimented with cavitation of such intensity that but a few minutes were necessary to cause the severest of pitting on highly resistant materials. Neither of the two extremes is thought desirable. The method employed for accelerating the tests should under no conditions change the fundamental characteristics of the cavitation attack. For instance, there is serious doubt in regard to the similitude of cavitation generated by vibratory equipment as compared with that occurring in turbine installations, because stroboscopic observations have shown that the formation of cavities with this type of apparatus follows a consistent pattern, and that the attack occurs repeatedly on any location in an identical fashion [17]. This method is radically different from field conditions in which successive blows fall irregularly without following any definite pattern both as to time and space, which results in a kneading type of stress application. In view of this, it is believed that only cavitation test stands of the Venturi or nozzle type may produce experimental results under accelerated conditions which may be regarded as truly comparable to prototype conditions.
In the light of the above, the results obtained in the course of the various investigations may not be directly compared with each other and the validity of particular findings must of necessity rest on the merits of the type of stress application produced by the laboratory apparatus employed. In addition, as the vast majority of studies confined themselves to mere surface examination, caution is deemed advisable in following some of the theories evolved in explanation of the mechanism of pitting. However, some researches have been carried out with the aid of complete metallographic analyses, which alone may provide an adequate basis for the determination of the characteristic phenomena [6].

Pitting due to cavitation is indeed a very complex mechanical process. Detailed microscopic and photo-micrographic examinations of a large variety of ferrous and non-ferrous materials in various conditions have indicated that many different factors influence the resistance. Some of the most important may be mentioned:

Chemical composition: Amount and kind of alloying elements; magnitude of free carbon content; presence of free-machining agents such as phosphorus, sulphur, or selenium; purity of metal and size of non-metallic inclusions.

Physical characteristics: Ductility, tensile strength, original hardness, susceptibility to strain hardening, maximum of induced surface hardness, fatigue properties, yield point, capacity to yield, and limit of yield exhaustion.

Characteristics of structure: Grain size, grain shape, amount of interlocking between grains, stratifications providing little bond between successive layers, presence in ductile matrix of brittle substances such as formations of eutectoids, etc.

There is also indication that under certain conditions corrosion may play an important part. As the casting skin on runners usually shows a higher hardness and a denser structure than the parent metal beneath, active corrosion for a sufficient length of time may expose less resistant material. This may explain the fact that some turbine runners have shown severe damage due to cavitation only after a long period of operation with complete immunity to pitting. The removal of the casting skin by corrosion alone may not be the
only factor in such instances. The roughening of the surface may be an important contributory cause in making local conditions in the wake of these corroded areas more conducive to cavitation.

The importance of the various material characteristics is readily seen from the nature of the punishment. Blows, varying in intensity and covering very small surface areas, follow each other in rapid succession. Due to this type of attack, the surface layer, perhaps a few thousandths of an inch in depth, tends to yield. Rockwell hardness tests have demonstrated that the kneading type of stress application may appreciably increase the surface hardness. Furthermore, microscopic examinations have disclosed slip bands, and in other cases twinning of crystals, and these visual findings substantiate the results of the hardness tests. Under attack of a severity sufficient to cause repeated stress applications in excess of the fatigue limit, failure in the surface layer will eventually take place due to fatigue or cleavage cracks. The fatigue limits are, therefore, one of the predominant factors for materials which do not tend to yield. With excellent yield properties, a continuous plastic deformation in the surface layer is likely to occur, and failure may then take place through exhaustion of the capacity to yield.

Photomicrographs have also shown that the progress of cracks, either inter- or intra-crystalline, may be retarded by grain boundaries. Thus, otherwise identical materials may have varied pitting resistances simply due to the difference in grain size. On the other hand, break-throughs between impurities or non-metallic inclusions may facilitate more rapid progress of the fatigue cracks. Likewise, stress concentrations on the tips of carbon, phosphorus, or sulphur flakes may be responsible for earlier initial failure. If cracks or crevices are formed of sufficient size to admit fluid, failure may also take place due to water-hammer or pressure-wave phenomena. This type of stress application can be particularly effective when these openings show a progressive reduction in cross section similar to a cone.

It is to be expected that the initial failure and the pitting progress depend upon the frequency, as well as the magnitude, of stress application. Failure may not take place if cavitation is of insufficient intensity either to exhaust the capacity of surface yield
or to reach the fatigue limits. This has also been demonstrated by operating experience with turbines. In some cases, cast carbon-steel propeller blades have seriously pitted during a short period of service, yet repairs of a permanent nature were possible by means of resurfacing with welded stainless steel.

Experience of this kind is also valuable from a research point of view, as it gives some indication regarding the magnitude of stresses exerted by cavitation on prototype installations. Keeping in mind that the hydraulic phenomenon is one of high frequency, the stress in these particular instances under relatively severe conditions may most likely be of the order of tens of thousands pounds per square inch, rather than hundreds of thousands or even a million pounds per square inch as has been suggested at one time or another. However, final judgment should be withheld, as future metallurgical research may indicate that the physical characteristics of single crystals or individual grains are sufficiently different from those of entire grain complexes or structures to warrant a reappraisal of the possible stress magnitude.

**Metallurgical Design Considerations**

Prior to discussing the manufacturing procedures of hydraulic turbines, it may be well to give some general comparative data on the pitting resistance of materials most commonly used. The results presented in Table II were obtained by means of a high-head Venturi-type cavitation test stand a few years ago. The wide range of the losses sustained during 16 hours of exposure illustrates the importance of material selection with respect to the relative life expectancy under cavitation conditions. In the light of these laboratory data, it would appear desirable to improve casting procedures of runners through the adoption of cement molds rather than to continue the use of the now customary casting sand. With cement molds, blades may be cast to close tolerance, and finish grinding, if at all necessary, may be kept to a minimum. This procedure leaves the casting skin practically intact, and, therefore, the metal most resistant to cavitation is preserved.

Although cast iron, commonly used in the past for Francis runners, has been found advantageous in regard to corrosion, steel castings are now preferred, due to their higher pitting resistance. Furthermore, in using steel castings far simpler repair procedures
### Table II

**Pitting Resistance of Various Turbine-Blade Materials**

<table>
<thead>
<tr>
<th>No.</th>
<th>Type of Material</th>
<th>Condition</th>
<th>Cavitation Loss in 16 hours mm³ at 20° C.</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>Cast Iron</td>
<td>as cast</td>
<td>636.0</td>
</tr>
<tr>
<td>2.</td>
<td>Cast Iron with Casting Skin</td>
<td>as cast</td>
<td>396.0</td>
</tr>
<tr>
<td>3.</td>
<td>.33% Carbon Steel of Turbine Grade</td>
<td>as cast</td>
<td>62.4</td>
</tr>
<tr>
<td>4.</td>
<td>Stainless 14% Cr., 1% Ni. Steel</td>
<td>as cast</td>
<td>32.0</td>
</tr>
<tr>
<td>5.</td>
<td>Stainless 14% Cr., 2% Ni. Steel</td>
<td>as cast</td>
<td>12.9</td>
</tr>
<tr>
<td>6.</td>
<td>Stainless 14% Cr., 3% Ni. Steel</td>
<td>as cast</td>
<td>9.8</td>
</tr>
<tr>
<td>7.</td>
<td>Stainless 18-8 Cr.-Ni. Steel</td>
<td>as cast</td>
<td>8.8</td>
</tr>
<tr>
<td>8.</td>
<td>Stainless 18-8 Clad Carbon Steel</td>
<td>rolled</td>
<td>11.2</td>
</tr>
<tr>
<td>9.</td>
<td>Stainless 12% Cr., 5% Ni. Steel</td>
<td>Two layers welded on wrought iron 8.4</td>
<td></td>
</tr>
<tr>
<td>10.</td>
<td>Stainless 12% Cr., 5% Ni. Steel</td>
<td>Two layers welded on boiler plate 8.1</td>
<td></td>
</tr>
<tr>
<td>11.</td>
<td>Stainless 18-8 Cr.-Ni. Steel</td>
<td>Two layers welded on wrought iron 8.2</td>
<td></td>
</tr>
<tr>
<td>12.</td>
<td>Stainless 17-7 Cr.-Ni. Steel</td>
<td>Two layers welded on wrought iron 1.3</td>
<td></td>
</tr>
</tbody>
</table>

are possible, because no studding is required prior to welding. On small turbines, and when cavitation conditions are particularly severe (such as with some early horizontal units placed too high above tailwater), solid stainless steel runners have offered good solutions as replacements became necessary. A few medium-sized Francis runners have been made of solid stainless steel, and their installation was justified by reason of the importance of unit availability and dictated by the remoteness of plant locations with consequent difficulties of repairs or renewals.

In Europe, where the adjustable-blade propeller-type turbines enjoyed a more intense development at the outset, solid stainless steel blades containing 14 to 15 percent chromium and 2 to 4 percent nickel have been used for the largest of runners and for heads in excess of 35 feet. In this country, stainless steel casting were not thought feasible for the first installation of large Kaplan units manufactured for Safe Harbor during 1930 and 1931. Each of the blades weighed more than 10,000 lbs. Even if the metallurgical difficulties could have been overcome, the costs at that time were believed to be prohibitive. Accordingly, these blades were manufactured of cast carbon steel of turbine grade. After only one and one-half years of operation, serious damage had taken place on the trailing halves of the blade suction faces, on the blade peripheries, and on areas downstream from the leading edges. Later on, the
throat rings were also affected. Based on experimental field repairs with various types of welded stainless steels and in the light of the results obtained with the cavitation test stand at Holtwood, 18-8 stainless chromium nickel welding wire was adopted for all resurfacing [18].

Welding with 18-8 stainless chromium nickel steel may not always be satisfactory. The proper selection of the welding wire depends upon the cavitation severity. While under prevailing conditions at Safe Harbor this type of stainless steel was found to withstand the attack, under more severe conditions 17-7 or even 16-6 stainless chromium-nickel steels would be preferable (see Table II). The better resistance of these particular materials is due to their higher initial hardness and the increased susceptibility to work hardening. However, the costs for grinding are also materially higher and, therefore, these alloys should only be used when cavitation conditions are unusually severe.

It may also be mentioned that satisfactory turbine repairs have been made with straight 18-percent stainless chromium welding wire. With this material extreme care is necessary, as hair cracks or larger check cracks may be formed after the metal is deposited. To guard against the development of cracks in depth which could weaken the runner, a stainless 18-8 chromium nickel steel may be used for the first welding layer in contact with the parent metal.

The inconvenience of field repairs from an operating point of view made it advisable to introduce shop welding of units prior to installation of the turbine. Blade areas of one large Kaplan runner protected with 18-8 stainless steel are outlined in Fig. 12. The finished runner blades are shown in Fig. 13. This method of manufacture proved very advantageous and is now widely used.
Two different methods of shop welding have been developed. Great care is necessary, otherwise harmful distortion of the blades may take place to the detriment of the essential turbine characteristics. In the first method, the blades are immersed in a water bath with only the area to be welded out of water. This arrangement provides for rapid dissipation of heat generated in the course of welding. In the other method, welding is limited at one particular location to give sufficient time for air cooling between laying of adjacent beads (see Fig. 14).

The different possibilities in manufacture of propeller units with movable blades make it imperative to explore thoroughly the economic aspects. Manufacturing costs for various runner sizes based on the normal competitive conditions in 1938 are given in Table III. The sizes of assumed pre-welded areas are identical to those indicated on Fig. 12. From Table III it is apparent that, when regarded exclusively from a manufacturing point of view, shop welding procedures are more economical for large-size runners than solid stainless steel castings. However, for units of 80 inches or less in diameter, solid stainless steel blades are as economical as shop welding. Furthermore, the thinner blades of small runners may make it impossible to apply the required two layers of welding.
material. Stainless construction of fixed-blade propeller and Francis units is somewhat more limited at the present time due to the restrictions imposed by the capacity of casting facilities, though fixed-blade stainless steel propeller runners of 60-inch diameter have been made in this country. Once larger furnace capacities and more extensive casting experience with stainless steel become available, the manufacturing costs on castings of this type are bound to be reduced.

In view of the fact that rolled plates can be used for throat rings of riveted or welded construction, there are no manufacturing limitations today which could prevent the use of stainless steel to pro-

Photo. courtesy I. P. Morris Dept., Baldwin Southwark Division, Baldwin Locomotive Works.

Fig. 14.—Prewelding Turbine Blade with 18-8 Chromium Nickel Steel Using Method of Intermittent Welding at Various Locations.
Comparative Cost Index for Different Propeller Turbine Blades of the Movable Type Using Various Materials

Cost of Finished Ground Blades in Percent of Cast Carbon-Steel Blades

<table>
<thead>
<tr>
<th>Type of Blade Casting</th>
<th>220 in. Diameter Runner</th>
<th>100 in. Diameter Runner</th>
<th>60 in. Diameter Runner</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbon Steel</td>
<td>100</td>
<td>100</td>
<td>100</td>
</tr>
<tr>
<td>Stainless 18-8 Chromium-Nickel Steel</td>
<td>250</td>
<td>210</td>
<td>180</td>
</tr>
<tr>
<td>Stainless 14% Chromium-2% Nickel Steel</td>
<td>220</td>
<td>150</td>
<td>140</td>
</tr>
<tr>
<td>Carbon Steel prewelded with Stainless 18-8 Chromium-Nickel Steel</td>
<td>136</td>
<td>140</td>
<td>160</td>
</tr>
</tbody>
</table>

tect these stationary parts from attack by gap cavitation. Only restrictions imposed by economic aspects must be given careful consideration. Here it may be very advantageous to adopt 18-8 stainless clad or a stainless steel of the straight chrome variety heat treated to desired physical characteristics.

Finally, it should be pointed out that no general rules can be made as to procedure on type of construction or materials to be used. The proper solution is that which satisfies the hydraulic and the metallurgical as well as the economic aspects. As location conditions may vary over a wide range, each installation demands its individual solution. For instance, while expenditures for labor and materials for field repairs may be identical at various power developments, the over-all costs may be vastly different due to a possible loss in capacity and energy during unit outages. At generating plants with a maximum power-house discharge in excess of the average river flow, turbine maintenance can usually be scheduled for the seasonal low-flow period so that no monetary loss will be incurred due to the non-utilization of available water. On the other hand, for power plants of a relatively limited capacity compared to that potentially available at the particular site—that is, where the river flow is consistently in excess of the maximum possible power-house draft—the actual expenditures for repairs may be considered as but a minor item of expense. Assuming that three weeks would be needed for maintenance work on a turbine rated at 50,000 horsepower capacity at the average operating head and that the load factor is 80 percent, the loss in revenue due to this outage
would exceed $40,000 based on 2.5 mills per kilowatt hour of re­
placed steam energy. From this it may be concluded that monetary
losses may be of such magnitude as to justify economically not only
the most extensive surface protection of all vulnerable turbine parts,
but even the adoption of solid stainless steel construction for the
largest of turbine runners.

Conclusions

Important progress in design and manufacture of hydraulic
prime movers, particularly for those of the propeller type, has been
made. Further refinements towards the ideal are possible. To
accomplish this goal economically, a sound strategic approach is,
however, deemed essential. Future development costs must be
kept within reasonable limits to be commensurate with the improve­
ments yet to be attained. This, no doubt, is a challenge to all
engineers interested in hydraulic turbines.

References

pp. 203-209.
Factor by Air Tests," Escher Wyss News, Vol. XII, No. 1-2,
17, 1930, pp. 418-419.
No. 11, March 17, 1934, pp. 349-351.
bines," Escher Wyss News, May-June, 1933, pp. 77-84.


