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ASSESSMENT OF DESIGN CONSIDERATIONS AND FIELD REPAIR
PROCEDURES TO MITIGATE CAVITATION PITTING
IN HYDRAULIC TURBINES

by

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ABSTRACT

Cavitation erosion in hydraulic turbines is of major importance to all members of the hydroelectric community. This erosion is caused by the collapse of vapor bubbles against or very near turbine surfaces. These bubbles are formed in regions where the absolute pressure falls below the vapor pressures of the fluid. Nuclei flowing through these regions experience tension and quickly develop into vapor bubbles. These bubbles then rapidly collapse, creating very high pressure waves emanating from the center of collapse. When the bubble collapse occurs near a material surface, a minute crater may be formed. Over a period of time this can lead to the spalling of material, which will eventually require remedial measures in order to insure continual successful turbine operation.

II. INTRODUCTION

Cavitation erosion in hydroelectric turbines has been in existence since the development of the reaction turbine. In the early years, little was known about its causes, and, hence, attempts made to correct the problem were of the trial and error variety. With time, advances were made in the hydroelectric turbine industry to reach present day knowledge. Today, few if any cases exist where a correctly designed, placed, and operated turbine is eroded to the point of destruction.

While the intensity of cavitation erosion has been reduced, the problem has become of increased concern in recent years. The value of electricity has risen dramatically in recent years. With this inflation has come an effort to produce power as economically as possible. Hydropower has the advantage of having no fuel component expense as in other types of electrical production such as fossil fuel plants. Major expenditures of hydropower plants are initial capital investment along with the operation and maintenance costs. Many new plants, particularly small installations are being remotely operated, requiring personnel only for inspection and repairs.

Repair of cavitation damage, along with generator repair are two of the major reasons for increased downtime in hydroelectric installations. The cost of repairs continues to rise, and although numerical values of increased costs cannot be obtained from the study, an estimate of future trends can be made using the Engineering News Record periodical. During the ten-year period from January, 1973, to January, 1983, the cost of skilled and common labor increased 207 percent and 218 percent, respectively, with overall costs rising approximately 215 percent. Irrespective of the actual amount of the increase. It is anticipated that further increases will occur in the future. The actual expense is also dependent on future electrical rates. This expenditure can be further reduced if a repair crew does not need to be employed full time by the electric utility.

Another major expense of cavitation erosion is that of lost revenues due to lack of power production. The theoretical cost can be approximated at 1 percent of gross revenues by using an average of owner estimates of additional down time of from 0 to 7 days typically. This leads to an average of approximately 3-1/2 days or 1 percent of yearly energy production, which oftentimes could be more than the actual repair costs. Actual lost revenues may be far less or even nonexistent if the repairs are scheduled at a time when the energy produced by the turbine is not being used. However, expenses may rise further if the lost energy must be replaced by higher priced energy such as oil fired generators.

This survey was undertaken by the Electrical Power Research Institute to determine, if possible, the causes of the cavitation and to develop repair procedures and preventative measures to best mitigate the problem. This paper deals extensively with the reasons for the occurrence of cavita-

tion. In the data analysis it was possible to correlate relationships between design trends and operational procedures with erosion intensity. The data base for the study was developed through use of a 10 page, 87 item questionnaire. Prior to mailing, the questionnaire was sent to several owners and operators for their input regarding completeness and answerability. When feasible, trips to some owners were undertaken to aid in assuaging their concerns. Inclusion in the survey was determined by the following criteria.

- 1) Reaction turbines placed since January 1, 1950.
- 2) Power output of at least 20 MW or a discharge diameter of more than 120 inches.

Response to the survey was outstanding with over 220 of the 270 questionnaires mailed and returned. These questionnaires represent 602 units totaling 40,000 MW of conventional turbine capacity or more than 67 percent of total U.S. conventional hydropower capacity.

Several surveys of this nature have been made. The last analysis of practice in the U.S. was made thirty years ago (1). These data cover the performance of units in place prior to 1950 and do not provide much guidance concerning the current state of the art. More recent studies have been made in other parts of the world, such as Canada (2), Japan (3), and Europe (4, 5, 6).

In some cases funding limitations prevented full examination of field experience. The Canadian study, for example, was intended to be the first part of a two part study. As it stands, this study contains standard trend correlations. Little has been done to examine the basic factors in the cavitation erosion problem. Where appropriate, the data base in this study has been used to supplement the U.S. data. In the case of design correlation, there is general agreement between the U.S. and Canadian experience.

The Japanese study (3) was carried out on a site specific basis. Particular attention was paid to the nature and causes of the observed erosion, i.e. whether due to cavitation, corrosion, or the abrasive action of suspended solid material in the flow. Sediment is a particularly acute problem for hydroturbine operation in both Japan and China. Surprisingly similar conclusions to those in this study are reached vis a vis mechanisms of erosion repair procedures, the effect of operational procedures etc., the exception being the relatively more important problem of erosion due to suspended sediment, which, of course, is not a cavitation problem.

The extensive surveys of de Siervo and de Leva on turbine selection and design (4, 5, 6) were of considerable interest to this study. Although cavitation erosion was not considered in any depth, the extensive correlations of design trends support the conclusions reached on U.S. turbine design practice.

The recent work of Simoneau (7) is also of interest, since a similar, but dimensional version of I_n is used in his analysis. Again trends similar to those observed in this study were found.

III. THEORY OF CAVITATION

Cavitation, while a complex phenomenon, is an understandable and important branch of fluid mechanics. Very simply, it is the formation of vapor bubbles in a liquid caused by reduction in the hydrodynamic static pressure on the fluid. Cavitation resembles boiling with one major difference: in boiling the vapor pressure of the liquid is raised to the atmospheric or surrounding pressure, whereas in cavitation, as stated above, the pressure on the fluid is reduced to vapor pressure.

In nearly all cases of concern, the pressure drop is associated with an increase in velocity. Few systems are designed to permit the occurrence of cavitation on a regular basis. Aside from its erosive effects, it may also produce noise and vibration which can be of a damaging nature to the machinery and have a negative impact on workers subjected to these conditions. Therefore, the occurrence of cavitation is often limited to valves, bends, or control devices which can cause high localized velocities. As the velocity increases, Bernoulli's equation provides for a corresponding pressure decrease. If the pressure falls to vapor pressure, cavitation may occur. On hydraulic structures such as spillways, cavitation may occur in regions well away from control structures and at mean velocities which are not high enough to lower pressure to vapor pressure. Closer analysis, however, shows that the cavitation is associated with surface roughness or isolated asperities in the structures' surfaces.

Hydraulic pumps and turbines are interesting in that they can present all of those concerns previously mentioned, along with the rapid removal of static head through the turbine which is caused by power production. Constant rotation adds an additional variable to the problem. By design, high velocities occur even in properly designed turbines. For example in regions near the entrance edge of the band on a Francis turbine velocities are high due to the rapid turning required of the fluid in this region. Poor welds or mismatched joints can raise problems as can the surface finish of blades and other parts. As fluid flows toward the discharge edge of the blade, conditions become ripe for cavitation occurrence as head has been removed allowing for less safety before critical conditions (the reaching vapor pressure) occurs.

Cavitation is not automatic once the pressure drops below vapor pressure. Tension occurs in the fluid after the pressure falls below vapor pressure, and in order for cavitation to develop, a number of other conditions must exist. Nuclei must be present to initiate bubble formation; these nuclei may be of many forms but all share the common quality of being weak spots in the fluid. For a nuclei to trigger cavitation it must be subjected to enough tension for a sufficient period of time. The amount of tension and time required are dependent on the nuclei size. The number of cavitation bubbles is, therefore, proportional to the amount of tension, the duration below the critical pressure, and the number and size of nuclei.

$$\text{No. of bubbles} = f(T, t, \text{No. of nuclei}, \text{size of nuclei}) \quad (1)$$

While understanding the formation of cavitation is of much importance to its reduction, the actual damage producing mechanism is the bubble collapse. As the bubble moves into a region of higher pressure, it collapses rapidly, on the order of micro-seconds, with intense shock waves emanating from the collapsed center. The intensity of collapse relates to the bubble size, static pressure in the collapse region, the dissolved gas content, compressibility, viscosity, and surface tension of the fluid.

$$I_c = f(R, P_s, \alpha, E_w, v, S) \quad (2)$$

Increased bubble size and static pressure in the collapse zone both lead to increased collapse intensity. Increases in the gas content, viscosity, and surface tension all tend to reduce the intensity of collapse. The gas content acts as a cushioning agent to collapse while viscosity and surface tension reduce intensity by slowing the collapse velocities.

Cavitation erosion requires that vapor bubbles collapse near the material surface as the pressure waves emanating from the collapse center decay rapidly, as r^3 . When a bubble collapses near the surface, the pressure waves can create small craters. These pits can also be formed when a micro-jet caused by asymmetrical collapse due to a nearby surface impinges on the surface. When enough craters have formed, stress caused related cracking may occur between pits leading to the spalling away of material.

The rate of material removal is dependent on the intensity of cavitation and the resistance of the material to cavitation attack. Two basic hypotheses to material resistance have been put forth in the past. One idea is that a very hard surface finish protects the material by being able to resist impact. The second states that a very soft surface absorbs impact pressures, thereby reducing erosion.

A method of determining the relative likeliness of cavitation is available using the cavitation index, σ .

$$\sigma = \frac{P_0 - P_v}{\frac{1}{2} \rho U^2} \quad (3)$$

Sigma is basically an Euler number referenced to vapor pressure. It can also be thought of as the ratio of a free-stream reference pressure to dynamic pressure. Critical values of sigma, σ_c , can be determined for all types of hydraulic equipment. Critical sigma is the value at which a device will begin cavitating. Higher values of critical sigma represent a greater likeliness for a device to cavitate.

$\sigma < \sigma_c$ No cavitation

$\sigma > \sigma_c$ Cavitation occurs

IV. METHODS OF ANALYSIS

The general program of analysis was designed to enable correlation of any desirable parameters as the needs arose. The questionnaire (Appendix A) was separated into two categories consisting of design and operational information.

In analyzing the data, information was grouped into three categories. The first category represents the correlation of various design parameters. This body of information provides a perspective on past and current design trends. The following correlations were made:

- Plant Sigma vs. Specific Speed
- Critical Sigma vs. Specific Speed
- Power Coefficient vs. Specific Speed
- Unit Speed vs. Specific Speed
- Specific Speed vs. Net Head
- Power Coefficient vs. Net Head
- Non-dimensional Head vs Non-dimensional Flow
- Rotational Speed vs. $P/H^{1.25}$
- I vs. Plant Sigma/Crit Sigma
- I vs. $(\sigma_p - \sigma_c) * H$

The second grouping of information provides information concerning erosion rate as a function of design parameters. The correlations consist of the following:

- I_n vs. Specific Speed
- I_n vs. Chop (Ratio of Blades to wicketgates)
- I_n vs. Coefficient Power
- I_n vs. Unit Speed

- I_n vs. $NS\sqrt{H}$
- I_n vs. H
- I_n vs. D

Finally, the third grouping of data provides information on the effect of variations in operational procedures on erosion. This grouping of data consists of the following:

I_n vs CO (Coefficient of overoperation)

I_n vs operational/rates Ns

I_n vs. (Max H - Min H)/Ave. H.

An additional set of correlations was aimed at testing the assumptions used in non-dimensional intensity that was developed as part of this study. The following correlations were made during this sub-task:

Area of Cavitation vs. D^2

Non-dimensional Volume vs. D/H

Non-dimensional Volume vs. H each for various Ns Ranges

Non-dimensional Volume vs. D each for various Ns Ranges

Non-dimensional Volume vs. P/D^2 each for various Ns Ranges

Non-dimensional Volume vs. Power

Non-dimensional depth of penetration vs. P/D^2

Non-dimensional depth of penetration vs. Y

All parameters originally plotted were based on operating rather than rated conditions. The operating conditions were felt to more accurately represent the conditions critical to the occurrence of cavitation. The cavitation intensity used unless otherwise stated was based on metal removal from blade surfaces only as this region was believed to be more closely tied to turbine parameters correlated, whereas other areas, such as draft tube or nose cone cavitation were more site specific in nature.

The nomenclature used in this report is that agreed upon between Acres and St. Anthony Falls Hydraulic Laboratory unless otherwise specified. A description of the parameters used in the initial correlations is as follows. The units are nondimensional unless noted.

Sigma, Critical, σ_c

The value of critical sigma was obtained from model tests. The value of $\sigma_{1\%}$ was chosen for use with Francis units, while σ_s was used for propeller and Kaplan machines as recommended by Allis-Chalmers Fluid Products Company. Both values are as defined by IEC 193A.

Plant sigma - σ_p

Plant sigma was computed using net head, barometric pressure based on plant elevation, and a vapor pressure of 0.7 ft of water in the English system of units. This vapor pressure was chosen for the study and is valid for water at approximately 65°F.

$$\sigma_p = \frac{H_b - H_v - Z}{H} \quad (4)$$

Net Head, H (ft)

Net head was determined from the normal operating head at the plant less system losses. Losses were assumed to be a minimum of 1.0 ft for heads less than 50 ft and two percent of gross head for higher heads.

$$H = H_G - 1.0 \quad (H_G \leq 50 \text{ ft}) \quad (5)$$

$$H = 0.98 H_G \quad (H_G > 50 \text{ ft}) \quad (6)$$

Reference Diameter, D (ft)

The runner discharge diameter was chosen as the reference diameter.

Nondimensional Head, H_{ND}

Nondimensional head assumes that hydraulic efficiency is the same in model and prototype. This requires that Reynolds number, Re, must be high enough for losses to be independent of Re. It also requires that geometric similarity and equivalent relative roughnesses exist between model and prototype.

$$H_{ND} = \frac{gH}{\Omega^2 D^2} \quad (7)$$

Nondimensional Flow, Q_{ND}

Nondimensional flow is the ratio between kinetic energy and the centrifugal force potential energy.

$$Q_{ND} = \frac{Q}{\Omega \Omega^3} \quad (8)$$

Specific Speed, N_s

A basic parameter of turbine design and selection, specific speed, is a combination of nondimensional head and nondimensional flow. Specific speed describes a particular combination of operating conditions which ensure similar flow patterns in geometrically similar machines.

$$N_s = \frac{\Omega P^{0.5}}{\rho^{0.5} (gH)^{1.25}} \quad \text{Turbines} \quad (9a)$$

$$N_s = \frac{\Omega D^{0.5}}{(gH)^{0.75}} \quad \text{Pumps} \quad (9b)$$

Specific speed was computed using both operational and rated parameters to enable the determination of off design operation on cavitation. The value computed with operational values was used in all other correlations, however. The specific speed parameter used in the report may be converted to dimensional units using the following conversions.

$$n_s = 43.5 N_s \quad \text{English units}$$

$$n_s = 193.1 N_s \quad \text{Metric Hp}$$

$$n_s = 166 N_s \quad \text{Metric using KW for power}$$

Speed Coefficient, ϕ

The speed coefficient is proportional to the ratio of the tip speed of a runner and the maximum water velocity under normal net head conditions. The actual ratio can be obtained by dividing by a coefficient of 2.828.

$$\phi = \frac{\Omega D}{(gH)^{0.5}} \quad (10)$$

Power Coefficient, C_p

The power coefficient is a parameter which expresses power loading on a turbine blade. A decrease in the power coefficient should lead to a decrease in cavitation.

$$C_p = \frac{P}{\rho D^2 (gH)^{1.5}} \quad (11)$$

Over-operation Coefficient, C_o

This coefficient expresses the amount of time the turbine operates above best gate.

$$C_o = 1 + \frac{\% \text{ of time above best gate operation}}{100} \quad (12)$$

Chop

Chop is the ratio of the number of blades to the number of wicket gates.

$$CHOP = \frac{\text{Number of Blades}}{\text{Number of Wicket Gates}} \quad (13)$$

This parameter was used to detect occurrence of any shadowing effects caused by the wicket gates.

Safety Over Critical Sigma₁, σ_p/σ_c

The first method of calculation of safety over critical sigma involves the ratio of the plant sigma based on operational conditions and its corresponding critical sigma determined from manufacturers model tests. Higher values of this parameter indicate increased safety over critical sigma.

$$\text{SAFETY 1} = \frac{\sigma_p}{\sigma_c} \quad (14)$$

Safety Over Critical Sigma₂, $(\sigma_p - \sigma_c)H$ (ft)

This method of computing safety determines a value in feet that the turbine is set below the critical turbine setting.

$$\text{SAFETY 2} = (\sigma_p - \sigma_c)H \quad (15)$$

Amount of Metal Removed

The amount of metal removed between repairs was determined using two methods. The primary method involved the use of the area of cavitation and the average depth of erosion as estimated by repair personnel. This was then converted to weight loss using a specific weight of 0.282 lbs per cubic inch (487 lbs/ft³).

$$W = 0.282(A \cdot \bar{y}) \quad (16)$$

A secondary method was used when average depth and cavitation area were not available. This method computed weight loss from the amount of weld rod and wire used in repair. An assumption was made that the weight of material removed was one half that used in repair.

$$W = \frac{(\text{wt. weld rod} + \text{wt. weld wire})}{2} \quad (17)$$

The second method could only be used for the determination of cavitation erosion within the entire turbine, and was unable to be of assistance for the determination of just blade surface cavitation alone.

Cavitation Intensity, K_1 (lb/ft²yr)

Cavitation intensity K_1 is a long used standard cavitation coefficient of the hydroelectric industry. K_1 relates the amount of metal loss proportional to runner area per year.

$$K_1 = \frac{W}{D^2 \cdot \text{Hours in operation}} \quad (18)$$

Cavitation Intensity, K_2 (lb/ft²yr)

K_2 is cavitation intensity K_1 which has been adjusted for a turbine load factor.

$$K_2 = K_1 * \frac{\text{Maximum Power at Normal Head}}{P_{\text{ave}}} \quad (19)$$

where $P_{\text{ave}} = \frac{\text{Average Annual Generation}}{\text{No. of Units} * \text{Hours of Operation per Year}} \quad (20)$

Cavitation Intensity, I_n

The nondimensional cavitation intensity, I , was developed to relate equivalent cavitation conditions. It was developed using three basic assumptions.

$$I \sim ySe \quad (21)$$

The first is that the intensity of erosion is equivalent to the mean depth of penetration per time y , multiplied by a representative material strength parameter Se . This gives units of power per unit area, since the strength parameter has the units of F/L^2

$$I \sim y Se \quad (21)$$

A second assumption is that intensity is proportional to the power loading on the blades.

$$2) \quad I \sim \frac{\text{Total Power}}{D^2} \quad (22)$$

This is based on the idea that, for a given cavitation intensity, a certain portion of the total available energy is used in the formation of cavitation bubbles. Another way of looking at this assumption is that, for a given machine operating under one set of conditions, an increase in power loading will lead to an increase in intensity. One problem inherent in

this assumption is the fact that, while the idea is sound, a critical value of power loading must be reached before cavitation occurs. The determination of this critical value of power loading from the data obtained in this study is not possible, however, as it is dependent on many variables such as σ_p/σ_c , N_s , head variation, etc. This assumption, however, appears valid from other cavitation research.

The third assumption in the development of the normalized intensity I_n is that the area of cavitation is proportional to the diameter squared.

$$A_c \sim D^2 \quad (23)$$

This assumption has also been used in the development of the more familiar definitions of cavitation erosion; K_1 and K_2 .

A non-dimensional cavitation parameter can be developed by combining the three assumptions in the following manner. The depth of penetration per unit time \dot{y} , can be determined as below.

$$\dot{y} = \frac{W}{T_e \cdot A_c \cdot \gamma} \quad (24)$$

since

$$A_c \sim D^2 \quad (25)$$

$$\dot{y} \sim \frac{W}{T_e \cdot D^2 \cdot \gamma} \quad (26)$$

Since

$$I \sim \dot{y} S_e \quad (27)$$

$$I \sim \frac{w S_e}{T_e \cdot D^2 \cdot \gamma} \quad (28)$$

I can then be normalized by using the second assumption in the denominator of the equation:

$$I \sim \frac{w S_e}{T_e \cdot D^2 \cdot \gamma \cdot \frac{P}{D^2}} \quad (29)$$

$$I_N \sim \frac{wSe}{T_e \cdot \gamma \cdot P} \quad (30)$$

The material strength parameter of ultimate resilience was chosen based on cavitation erosion research work by Thiravengadam (9). This parameter was also stated to give the best correlation to erosion intensity by H. Kato (10). Ultimate resilience is basically a weighted area under the stress-strain curve of a material (Fig. 1).

$$U.R. = 1/3E (2S_m + S_y) \quad (31)$$

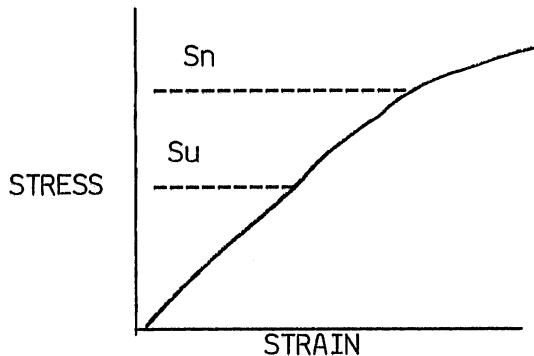


FIGURE 1

The parameters used in correlations permitted the observation of several trends expected from previous cavitation research. One such trend is the increase in cavitation erosion with velocity. In past research it has been found that cavitation erosion is proportional to V^6 . On this basis cavitation should also be proportional to head. The assumption that cavitation erosion is a function of power loading is based on the idea that cavitation uses a certain proportion of the total energy in its formation. As the power loading increases, the amount of cavitation should, therefore, also increase. One additional trend expected is the increase in cavitation with diameter. The exact relationship is unknown, since there does not appear to be any information available on this point in the literature. Owing to these considerations, it should at the very least be expected that I_N is functionally related to n_s and σ .

V. SURVEY DISCUSSION

As mentioned previously, all major design parameters were considered in this study. When plotted versus one another, most design parameters exhibit good correlation. This is due in a large part to the interdependent relationships between parameters. Very little scatter is evident. Much of that which appears in these correlations can be attributed to variance from design trends due to site specific considerations. Correlations involving the non-dimensional cavitation intensity I_n showed both expected and unexpected results; however, scatter was several orders of magnitude.

An example of a good correlation between parameters is given in Figure 2. This is a plot of speed coefficient versus specific speed. The speed coefficient is a measure of peripheral (and maximum velocity as well) normalized with respect to head. This is one parameter in which designers apparently have little latitude in varying for a given design specific speed. The significance of this parameter in relation to cavitation pitting will be discussed subsequently.

If one looks at the linear correlation of speed coefficient ϕ with specific speed, i.e.

$$\phi = CN_s \quad (32)$$

a relationship between specific speed and head is easily deduced. Using the definition of speed coefficient, Eq. (32) can be written as

$$N_s = \frac{wd}{C \sqrt{gH}} \quad (33)$$

From Figs. 2 and 3 it is apparent that manufacturers are forced to limit peripheral speed, wd , due to the occurrence of cavitation. It can be noted in Fig. 3 that specific speed is inversely proportional to the square root of head. This is precisely what one would expect from inserting constant peripheral speed, wd , in Eq. (33).

Figure 4 is the correlation between power coefficient and specific speed. Again, this parameter increases monotonically with specific speed. This parameter is also of interest to our interpretation of cavitation pitting rate. Noting the inverse relationship between specific speed and the square root of head, a linear relationship between power coefficient and specific speed implies a linear relationship between power loading, P/d^2 , and head.

It can be seen in the interparameter relationships of Figs. 2 through 4 that designers are trying to minimize turbine diameter. Theoretically, it is possible to develop a turbine of any specific speed for a given head. From a practical point, however, using the highest possible peripheral

speed along with the highest power loading leads to the smallest allowable diameter which in term develops the specific speed, head relationship. Recent designs have noted an increase in specific speed for a particular head. This relates to higher peripheral speed, higher power loading and a corresponding reduction in diameter.

It should be noted in Fig. 4 that power coefficient is noticeably higher for Francis as opposed to propeller-type machines at an equivalent specific speed.

Cavitation Pitting Experience

Figure 5 shows the relationship between plant sigma, σ_p , and specific speed. It can be seen in this figure that the Bureau of Reclamation experience curves developed many years earlier are still a good indication of recent turbine setting criteria. It should be noted in Fig. 5 that while much scatter is apparent at high specific speed, this is in part due to these machines being low head. Being low head machines, high specific speed units will exhibit marked changes in plant sigma with only a slight change in tailwater levels. Further insight is given in Figure 6. Here the best fit curve is compared to model test data for performance breakdown, σ_c , and the theory of Morel and Arndt (11) for the cavitation inception index, σ_i . Note that plant sigma falls somewhere between the inception index and the point for performance breakdown over most of the N_s range. In other words, the safety factor used by the manufacturer, $\sigma_p - \sigma_c$, is apparently a compromise between totally cavitation-free operation and economical settings.

Figure 7 is a plot of I_n versus specific speed. The data displayed must be interpreted carefully. As shown in Fig. 6, there is clearly a relationship between both the inception and critical cavitation indices and specific speed. The recommended plant settings are a compromise between cavitation-free performance and some acceptable rate of erosion. In a perfect world all plant settings would result in relatively the same rate of erosion, resulting in no correlation between erosion rate and specific speed. In other words, the determinant part of the problem (i.e. the relationship between σ and N) should have been factored out with only the indeterminants remaining in the data. Hence, it would be no surprise if there was no correlation between I_n and N_s . However, as seen in Fig. 7, even though there is a large amount of scatter in the data, two facts emerge. The upper limit on I_n appears to be about 2. In the case of Francis units, $N_s < 2$, the lowest achievable value of I_n appears to be strongly dependent on specific speed. The reason for this can be inferred, at least qualitatively, by examining the assumptions inherent in assuming a constant value of I_n . These assumptions are tested later in the report.

An increase in cavitation erosion, as previously mentioned, is expected with power coefficient. The trend is much stronger than expected for Kaplan, propeller and pump turbine units (Fig. 8), and shows near independence with the Francis machines. One possible reason for the apparent behavior of the Francis units relates to more stringent manufacturing tolerances of higher head and hence higher power loading machines. This will be discussed more fully later in this report.

Francis and pump-turbine units exhibit an increase in cavitation with speed coefficient (Fig. 9). This may in part be due to speed coefficients relationship with net head. As with the correlations of I_n vs. N_s earlier, a minimum value of I_n based on speed coefficient can be obtained. This value varies directly with the relationship between speed coefficient and head. For Francis turbines, however, the data indicate a slope of approximately 2.5 on log log plotting.

$$I_n \propto \phi^{25} \quad (34)$$

This is much stronger than that expected from dimensional analyses of the equations involved.

Contrary to expectations, obvious increases in cavitation erosion did not occur with increased head. This may be suppressed by manufacturer's increased effort toward better surface finish on units with higher heads.

As can be seen in Fig. 10, it is very difficult to ascertain whether or not there is a trend with the size of the machine. If there is a trend, the normalized cavitation area, A_c/d^2 , varies considerably with diameter. This would not be unexpected when certain facts concerning bubble dynamics are taken into account. The maximum pressures created during collapse is independent of the size of collapsing bubbles. However, the energy of a given collapse scales with the product of the cube of bubble size and the pressure in the region of collapse. It can be shown that the size of cavitation bubbles scales directly with the size of the machine when all other factors are held constant. Thus, as the size of the machine increases, relatively more bubbles are energetic enough to create a damaging blow during collapse. In other words, as the energy of collapse increases, relatively more remote bubbles can damage the surface of a runner.

Correlations with operating parameters yielded expected results yet the dependence of cavitation on operating characteristics was surprising. The trends seem to show a definite increase in cavitation erosion with off design operation.

Graphs of cavitation intensity with safety over critical sigma show a definite decrease in erosion with an increase in safety for all types of machines (Fig. 11).

As seen in Fig. 12, propeller units show an increase in cavitation erosion with increased time spent operating above best gate.

The amount of head fluctuation directly affects the amount of cavitation experienced by the turbine (Fig. 13). While the strength of this relationship was surprising in its clarity and dominance regardless of turbine type, it is not unexpected. When head varies significantly from design, leading edge cavitation occurs. With head well above design, leading edge cavitation will occur on the suction side of the blade while at heads below rated head, leading edge cavitation will appear on the pressure side of the blade.

In general, cavitation trends do appear although not necessarily in agreement with expected results. One of the more obvious reasons for this discrepancy is that the manufacturers may be compensating for these trends and, thereby, affecting the correlation by the accuracy of their correction factor. One of the most likely correction factors is the use of better blade finishes for higher head units. Another is the use of stainless steel or stainless steel overlay in the construction of a turbine to be used in more critical situations such as higher flow velocity through a turbine. One uncontrollable factor in the analysis is the amount of scatter in the data. This scatter can be of such a magnitude as to smother any trend which may be in existence.

Due to the extensive scatter apparent in the early analyses, additional correlations were run testing the basic assumptions of I_n , the nondimensional cavitation intensity. One such correlation was done for carbon steel plotting the average depth of erosion per time, y , versus power loading (Fig. 14). For a given material this is equivalent to the correlations between erosion intensity and P/d^2 . The correlation is not linear as would be assumed from a simplistic model. However, using Knapps Law

$$I \propto U^6 \quad (35)$$

it can be shown that y should be proportional to the square of P/D^2 which is exactly the trend noted in Fig. 14. The power of a flowing fluid gives

$$P \propto \rho U^3 \quad (36)$$

or

$$PD^2 \propto \rho U^3 \quad (37)$$

Inserting in Eq. (35)

$$I \propto (\rho U^3)^2 \quad (38)$$

Hence,

$$I \propto (P/D^2)^2 \quad (39)$$

Not shown is the trend for stainless steel overlay which was found to be eradict. This is indicative of the general lack of consistency in the effectiveness of overlays for minimizing pitting. The lack of consistency may be due to several factors including, but not limited to, poorly finished welds and galvanic corrosion. There are too few data to draw conclusions for solid stainless steel runners. Although the U^6 law is dimensional, and a more general relationship does not exist, the sixth power trend can be explained qualitatively using the same arguments regarding energy of collapse that were used in predictions of size effects. Much more work is

necessary to further quantify these factors. However, further refinement of these ideas in this study would probably not be warranted in light of the statistical variation inherent in a field study of this type.

"Scatter" is the word chosen to describe lack of correlation. It was hoped after entering this study that due to a large data base scatter involving the cavitation erosion may have been suppressed. Unfortunately, this was not the case, as the scatter turned out to be of such magnitude that trends which may have otherwise appeared were disguised. Several explanations for the scatter exist, each of which fits its own niche in the analysis. One problem arises from the variability of recording methods for cavitation damage repair from one utility to the next: an inconsistency within a utility, as the factor of individual judgement depends on the particular person recording the data.

Due to the many variables involved in turbine design, placement, and operation, all variables cannot be eliminated without encountering a large reduction in the data base. This reduction unfortunately reduces plots to just one or two points, too few from which to make any meaningful judgements. When several plots of a particular correlation are made for small ranges of a third parameter, the trends interpreted from the few data samples remaining on each plot often contradict themselves. This leads to the conclusion that the variation in parameters not obtained in the survey questionnaire must also be a factor in the cavitation conditions of a turbine.

Many variables exist which are site specific; that is, they are dependent on the particular site. The most prominent of cavitation affecting parameters which are site specific is the entrance flow patterns to the turbine. Poor inflow will often create problems which continue through the turbine. Unfortunately, a method does not exist for easily recording inflow patterns to relate to survey data. Another parameter which could have substantial influence on the scatter of the data is water quality. While this was not mentioned as a major problem by the operators, it does introduce an additional variable in that it was only obtained in a qualitative manner.

By far the most surprising of the correlations to come from the study was that of cavitation erosion with turbine manufacturers. Elimination of the variability with respect to the manufacturer is also difficult. Development of correlations for a single turbine manufacturer was discussed, however, the returns from such an attempt were thought to be minute compared to the time required. The reasoning for this decision was that, while one more variable would be removed from the data, those variables still remaining, as discussed earlier, along with a significant reduction in the data base, would reduce any foreseen benefits. It was decided; however, to do further analysis of the cavitation erosion characteristics with respect to turbine manufacturer.

Variance between manufacturers can be attributed to three major causes; turbine design, turbine manufacture and recommended turbine setting. A comparison between manufacturers can be seen in Table 1. While it is true that the site specific conditions still exist, an attempt was made to minimize the effect by omitting the best and worst 20% of the

units with respect to the amount of cavitation erosion for each manufacturer (Table 2). By eliminating the extreme turbines, manufacturers are less likely to be prejudiced by conditions beyond their control. It is apparent from this analysis that a large variation in cavitation characteristics exist between manufacturers, which cannot be attributed to site specific conditions (Table 3).

Subtle variations in turbine design philosophy would be difficult to determine due to "inhouse" information and are beyond the scope of this study. It was possible, however, to take a further look at the effects of turbine setting and manufacturing tolerances of a couple of major manufacturers. A comparison of the turbine setting of Manufacturer 1 and manufacturer 8 is shown in Fig. 15. It is apparent from the curves that differences in recommended setting occur between manufacturers. The number of turbines above and below the point of equivalence is given for the two manufacturers in Table 4.

Not surprisingly, significantly more units in operation were in the specific speed ranges where Manufacturer 8 had a higher recommended sigma, i.e. lower turbine setting. From Table 2 it can be seen that the units of manufacturer 8 were better in regard to cavitation erosion.

The third suspected cause of cavitation damage variation with manufacturer includes manufacturing capabilities along with a hidden factor of design recommendations regarding surface finish and acceptable tolerances. While it is impossible to determine the actual factors involved, it was possible to determine the magnitude of variation for a specific location. The plant chosen had units of both manufacturers 1 and 8 placed in an alternating manner with identical conditions; i.e. inflow patterns, setting, etc., for all units. From the analysis of this single site, the cavitation erosion was found to be 43% higher for those units manufactured by manufacturer 1 as opposed to those designed and built by manufacturer 8.

VI. ACTUAL COST OF STAINLESS STEEL RUNNERS

Referring to Fig. 4, we note that there is a good correlation between power coefficient, C_p , and specific speed, N_s . For fixed H , C_p is proportional to power loading, P/d^2 . Referring to Figs. 4 and 16 we see that, for fixed head, the power loading can be increased by using a stainless steel runner. An equal power comparison would then indicate that a stainless steel turbine is smaller. Thus, the additional material cost may be somewhat offset by cost savings due to reduced size. As an example, consider a 50 MW Francis machine designed to operate under a head of 400 ft. Using Figs. 3 and 4, the design values of N_s and C_p are

	SS	CS
N_s	0.84	0.77
C_p	0.144	0.134

Note that the diameter of the machine can be calculated from this information (see Eq. 11):

$$D^2 = \frac{P}{C_p \rho (gH)^{3/2}} \quad (40)$$

Thus a runner of 8.0 ft can be used with stainless steel, compared to an 8.3 ft diameter carbon steel runner. Assume the following:

- stainless steel runner is 25 percent more costly than carbon steel
- the runner accounts for 20 percent of the turbine cost
- turbine cost varies with $d^{1.85}$.

From the above, the ratio of turbine costs for stainless steel runners and carbon steel runners is:

$$\frac{\text{Stainless}}{\text{Carbon}} = (8.0/8.3)^{1.85} \times (1.0 + 0.25 \times 0.2) = 0.981$$

Stainless is therefore more economical. It must be recognized, however, that this analysis may be somewhat academic, as it applies to a given erosion rate. In actual practice, stainless steel may be selected for the same diameter to give reduced erosion rate.

VII. THE COST OF CAVITATION

The most obvious cost to the industry because of cavitation is the cost of necessary repairs. Considerable scatter in the data was found during our attempts at quantifying repair costs. After several attempts, it became apparent that the best approach would be to correlate the cost per pound, expressed in man-hours, with the number of pounds of material removed per year. Reasonable correlation could be achieved after correlations were made in given size ranges. It can be conjectured that small machines will be less expensive to repair because of easy removal. On the other hand, large machines might be relatively cheaper to repair in place than mid-size machines because of easy access to the eroded area.

Secondly, it must be recognized that the cost per pound should increase with a decrease in the total number of pounds of material removed. This is due to the fixed costs of removal and/or preparation necessary to carry out the repairs. The data so obtained can be integrated to yield a total repair cost as a function of the pounds of material removed. Sample results are shown in Fig. 17. Data of this type give an indication of how much amortization of increased capital cost for tighter guarantees can be borne by savings in repairs. Propeller and Kaplan units show similar trends. Only in the case of Francis units were there sufficient data to draw conclusions about repair costs for stainless steel units. The data do indicate that, although pitting damage is considerably attenuated through the use of stainless steel, any pitting that occurs is relatively expensive to repair.

If only repair costs are considered, the industry experience as a whole is that cavitation repairs cost a fraction of a mil per kilowatt hour produced. Using an average of \$20 per man-hour the direct costs average approximately 1.5 cents per megawatt hour. Outages would be a significant cost if they would occur. However, most respondents to the questionnaire found that repairs could be scheduled during low water periods. There was little, if any, report of spillage due to outages caused by cavitation. If direct repair costs and loss of revenue due to outages only are considered, it would appear that cavitation erosion should not be of significant concern to the hydroelectric industry. However, the most significant loss may be an embedded cost due to the reduction in the useful life of the machine. Utilizing only the data available to this study, it was not possible to quantify costs of this type. However, simple economics can show that a reduction in the lifetime of a 50 MW runner from 50 years to 35 years represents a loss in excess of \$400,000, assuming 5% inflation and a 10% interest rate. Obviously this is an oversimplification of a complex problem and further study is warranted. There is also a lack of hard data concerning what reductions in hydraulic efficiency can accrue over the useful life of the machine. Repeated repair of runners and other components can create subtle changes in the geometry of the flow passages, which in turn can have a significant effect on efficiency. Even very small reductions in efficiency can have a serious impact on revenues. Further research into the problem area is also warranted.

VIII. RECOMMENDATIONS FOR FURTHER RESEARCH

While it is obvious that cavitation is still far from completely understood, this study did enable further pinpointing of future research endeavors. Future research can be directed in two directions, the first being basic cavitation research, the second specifically directed towards the hydroturbine industry. It is impossible to draw a distinct separation between the two, however, as basic research may lead to a direct application in the hydroelectric industry.

Four specific areas involving basic research could prove useful to the turbine and pump industry. It became even further apparent during this study that surface finish directly effects the amount of erosion. Research has been done regarding the effect of surface roughness on the occurrence of cavitation (Arndt, 12). A similar study could be undertaken to determine the effects of generalized surface roughness and isolated asperities, i.e. welds and joints, on cavitation erosion intensity.

Evaluation of the overall diagnostics used in the determination of cavitation erosion intensity would aid in the standardization of testing procedures. Two such techniques are the ductile probe technique and the use of an erodible paint. In the ductile probe test a series of inserts of easily eroded material such as pure aluminum are placed in areas of expected cavitation. In the paint test, an erodible paint is placed in areas susceptible to cavitation. Many additional methods exist to determine cavitation erosion intensity. However, the two methods mentioned represent both the qualitative and quantitative measurement techniques. Due to the wide variation in testing methods, it would be advantageous to compare the techniques in present use to determine those methods most accurate and useful to the hydro turbine industry.

Correlations between noise and cavitation erosion may be difficult to achieve, but benefits from such research could prove very valuable. Difficulties may arise from the fact that cavitation occurring in the flow and away from any surface may cause noise but no erosion. Another problem is that noisy cavitation is not necessarily erosive cavitation even when occurring near a material surface (Fig. 18). Any correlations of noise and erosion could be valuable to the hydroelectric industry as it may be possible to place a noise intensity recording device, similar to a watt meter, which could aid personnel in determining when repairs are needed, thereby eliminating periodic shutdown for cavitation damage inspection.

Testing of cavitation erosion mitigation procedures such as air injection and the various non-metallic coatings in laboratory conditions would be research of a basic nature yet with direct applicability to the turbine industry. Under controlled conditions it would be possible to observe the resistance mechanism of the non-metallic materials. Such a study may enable development of a simple procedure to determine which, if any, of the non-metallic materials may be effective at suppressing erosion

at a given location. Similarly, it may be possible to systematically determine the benefits of air injection on reduction of the hydraulic turbine cavitation. Air injection can be effective in suppressing cavitation erosion as air injection reduces attack pressures which scale with the waterhammer pressure, p_{cv} , by causing a substantial reduction in the speed of sound, C , with a slight increase in air content.

In light of this study, an extensive investigation of the current computational procedures and manufacturing tolerances of pump and turbine manufacturers would be a definite benefit to the hydroelectric industry.

Lastly, of benefit to the hydropower field would be the standardization of the recording of repair procedures. It was noticed during this study that the recording of repair procedures varied greatly from one utility to another. Standardization of recording methods would help to reduce the scatter obtained in any surveys of similar nature attempted in the future.

VIII. CONCLUSIONS

Cavitation can affect the performance of turbines and can cause rough and noisy operation. Cavitation pitting necessitates periodic repair of many of the machines currently in service. Although the direct costs of repair can be calculated, the more insidious costs associated with down time, reduced lifetime of the equipment and a gradual reduction in efficiency with time cannot be as easily accounted for. These hidden costs can far exceed the direct costs for materials and repair personnel.

From experience with the 602 machines in the data base the following conclusions can be drawn:

- 1) The overall design parameters for given conditions of power and head did not vary much among machines of different manufacturers. The general trend is toward the smallest machine (highest tip speeds) for a given application.
- 2) The use of stainless steel can substantially reduce the pitting problem. The general trend in the industry is to design for higher tip speeds (smaller machines). The newer machines are also generally of stainless steel construction. The increased tip speed appears to correlate with the allowable increase in tip speed that would be predicted by the U^6 law. However, much of the increase in tip speed is probably due to better design. Although stainless steel is a dramatically superior material in terms of resistance to pitting, any major pitting that occurs is substantially more expensive to repair.
- 3) Although there was consistency in design, there is substantial scatter in the pitting data. Much of this scatter is due to variations in setting, quality control, construction details, and operational history.
- 4) The use of a normalized pitting rate does appear to improve the correlations. However, there are definite scaling problems with both assumptions in the normalization. The U^6 law indicates that pitting rate should correlate with the square of power loading, P/d^2 . Further, the area of cavitation does not appear to scale with d^2 . This latter scaling problem can be explained qualitatively utilizing the theory of growth and collapse of cavitation bubbles. Thus, even if cavitation number and specific speed were held constant, I_n would vary in a qualitatively predictable manner with size and speed. Size effects are especially pronounced.
- 5) As noted in (4) the d^2 law which is apparently universally used in establishing guarantees does not appear to be a reliable predictor of increases in erosion rate with increases in the size of the machine. This definitely warrants further investigation.
- 6) Forty percent of the units installed since 1970 with a guarantee have experienced erosion exceeding the average allowable rate specified in

that time period. Whether or not this is of concern to the industry is not known. The reasons for exceeding the limits have not been identified. It could be a case of recommended settings that are too generous or that the guarantees themselves are unrealistic. In any event, recommending tighter guarantees at this point is unwarranted until further information is available on scaling cavitation erosion, on methods for minimizing erosion and on the costs of improved quality control that will be necessary to minimize the problem.

7) Direct costs (man-hours and materials) for cavitation repair average 1.5 cents per megawatt hour produced. At first blush, the costs due to cavitation erosion do not appear to be excessive. Exceptions to the rule are there, but apparently are due to individual problems of cost control within a given utility. A major factor is the possible reduction in lifetime of a given unit. A reduction in useful life from 50 to 35 years represents a loss of about \$400,000 for a 50 MW machine.

8) Operation above best gate definitely aggravates the erosion problem. This effect is especially pronounced in propeller units. A tradeoff between the extra revenue from operation above best gate and increased costs due to excessive erosion has not been established.

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