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exist for the publication of theoretical and experimental investigations of all aspects of the mechanics and thermodynamics of fluid-flow with special reference to fluid-flow machinery

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Diagnostics of Cavitation in the Hydraulic Machinery*

A survey of the state of the art is given in the paper for the diagnostics of cavitation in the hydraulic machinery.

At the beginning a general description of cavitation phenomena occurring in the hydraulic machinery is given. The purpose and methods of cavitation diagnostics are explained. Exemplary results of authors' own cavitation studies are presented together with a discussion of relations between different symptoms of cavitation in water turbines and pumps. Prediction of cavitation and cavitation erosion is discussed in a separate chapter. The problem of selection of hydraulic machinery operating parameters from the viewpoint of cavitation is also taken into consideration. Finally future goals of cavitation diagnostics in the hydraulic machinery are shown.

1. Cavitation in the hydraulic machinery

The pressure in a hydraulic machine may locally drop down to the evaporation pressure of the liquid at the existing temperature. Cavitation occurs as a result of such pressure drop—vapour-gas bubbles form and develop, to collapse afterwards in the higher pressure regions.

Cavitation may appear at stationary or moving parts of a machine. Most often this phenomenon occurs at the back face of rotor and guide vanes (fig. 1) and at the rotor casing surface.

When cavitation takes place at surfaces which are in contact with the flowing liquid then local damage to the material may occur. Obviously the extent of damage depends on the cavitation intensity and duration as well as on the resistance of the particular structural material to the cavitation erosion. Cavitation is accompanied also by such phenomena as pressure pulsation in the stream of liquid, vibration of walls limiting the flow field, cavitation noise, vibration of the machine as a whole and changes (deformation) of its performance curves. The level on which these phenomena occur depends on the cavitation development stage as well as on pressure gradients in the flow and design of the machine.

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The susceptibility of hydraulic machines to cavitation depends on their specific speed, that is

$$n_q = n Q^{1/2} H^{3/4} \quad [\text{min}^{-1}; \text{m}^3/\text{s}; \text{m}].$$

In low and medium specific speed machines, that is those with radial and mixed flow through the rotor, the critical pressure appears, as a rule, on the suction side, namely at the inlet to an impeller and at the outlet from a turbine runner. In high specific speed machines, i.e. machines characterized by axial or nearly axial flow through the rotor critical pressure may occur even on the pressure side: at the outlet from a pump impeller or at the inlet to a turbine runner.

Working conditions, including conditions of flow into the rotor region, affect considerably the incipience of cavitation. Ranges of favourable conditions are limited — see as an example the ranges of flow rate, Q/Q_{opt} , and heads, H/H_{opt} , defining cavitation region boundaries for a Francis turbine with the cavitation specific speed equal to $n_{q\text{opt}} = 70$ (fig. 2).

The state of a machine is characterized from the viewpoint of cavitation by the Thoma number

$$\sigma = \frac{h_b - h_{va} - h_s}{H} = \frac{\text{NPSH}}{H},$$

where h_b is the atmospheric pressure measured on the tail water free surface level, h_{va} is the evaporation head, h_s is the geodetical suction head i.e. the difference between the rotor

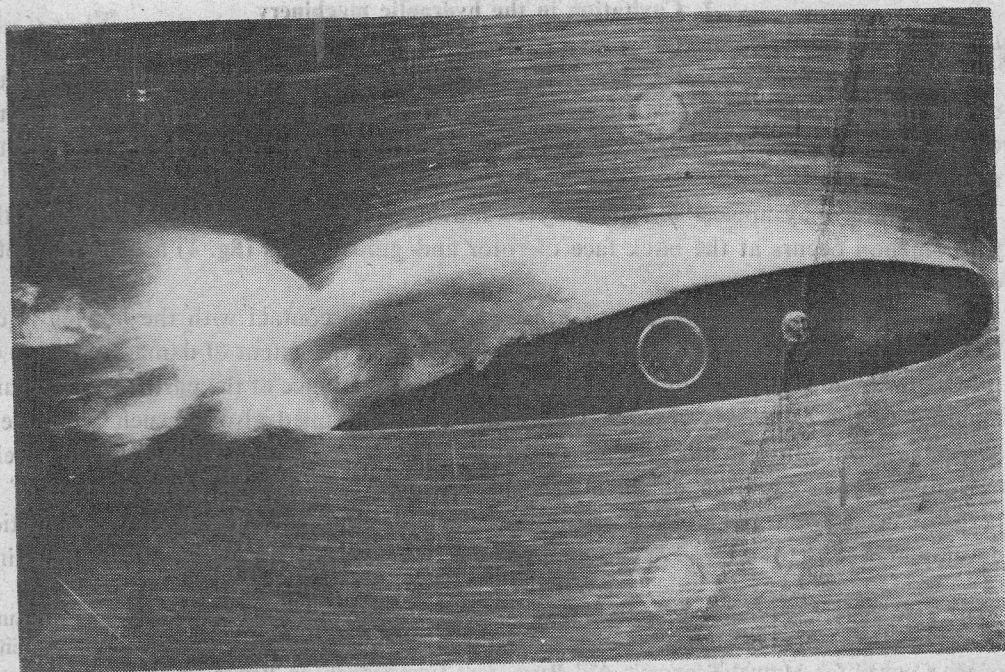


Fig. 1. Cavitation cloud over a hydrofoil for the angle of attack α equal to 9° and the cavitation number σ amounting to 2.5

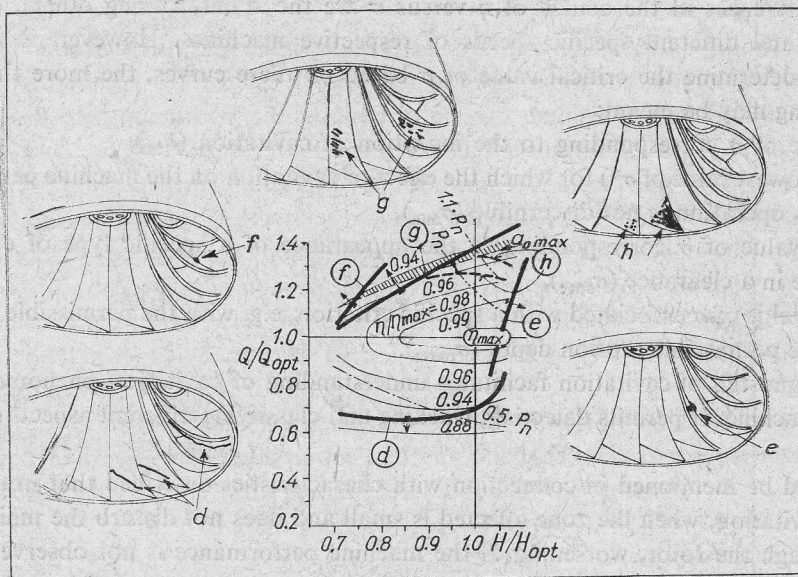


Fig. 2. The course of cavitation at the runner of a Francis turbine characterized by the specific speed n_q of 70 (the boundaries of the cavitation region are defined by lines d, e, f, g, h on the set of iso-efficiency curves) [27]

datum and the tail water free surface datum, H is the total head for the particular working point of the machine, and NPSH is the net positive suction head.

The effect of cavitation number σ is not identical for all cavitation region boundaries as shown in figure 2. The boundary designated by h depends clearly on σ while for the remaining boundaries (d, e, f, g) cavitation does not depend, or depends only slightly on the value of σ .

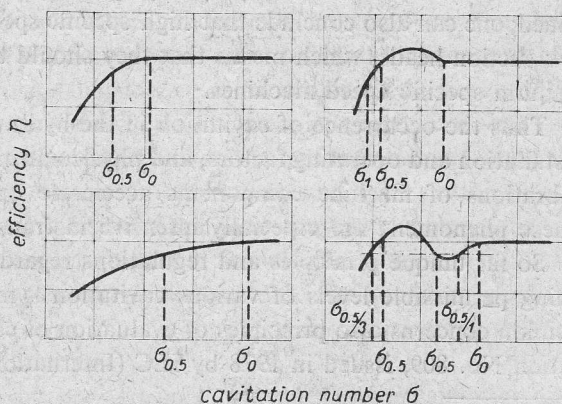


Fig. 3. Typical cavitation characteristics of impeller pumps

The influence of cavitation on the machine performance — flow rate, total head, power and efficiency — shows as a deformation of appropriate characteristics. Figure 3 presents typical $\eta=f(\sigma)$ curves illustrating the effect of cavitation on the efficiency of impeller

pumps. Differences in the course of η versus σ are the result, among others, of design differences and different specific speeds of respective machines. However, it would be difficult to determine the critical value of σ based on these curves, the more that by σ_{cr} the following may be meant:

- value of σ corresponding to the inception of cavitation (σ_{in}),
- the lowest value of σ^* for which the effect of cavitation on the machine performance curve or its operation is not discernible (σ_{min}),
- the value of σ corresponding to the appearance of a specific type of cavitation, for instance in a clearance (σ_{spec}),
- the value of σ associated with a specific criterion, e.g. with the permissible efficiency drop, or the permissible erosion depth (σ_{perm}).

The diagnostics of cavitation facilitates understanding of cavitation phenomena occurring in a machine. It permits detecting, tracking and classifying different aspects of cavitation.

It should be mentioned in connection with characteristics described that in the initial stage of cavitation, when the zone affected is small and does not disturb the main flow of liquid through the rotor, worsening of the machine performance is not observed. Sometimes even a slight increase of the head and efficiency may occur, caused by a favourable change of the pressure distribution over blades or reduced friction in the rotor. In the second stage of cavitation, when the zone affected is sufficiently large to obscure the flow, hydraulic losses increase and the flow rate drops. The performance curves of the machine exhibit distinct deformations. Simultaneously vibrations of the machine and noise increase.

It was already said that properties of a machine concerning cavitation depend on its design and specific speed. This dependence is expressed by the following formula

$$\sigma = \kappa \eta_q^{\frac{4}{3}},$$

where κ is a factor allowing for the specific design of the machine and its efficiency.

From this formula not only the susceptibility of a machine to cavitation may be determined; one can also conclude that high specific speed machines should operate at possibly low suction heads, which means that they should be founded more deeply than low and medium specific speed machines.

Thus the occurrence of cavitation in the hydraulic machinery is a resultant of design, installation and operating factors, and physicochemical properties of the working medium. Vibrations of machine components accelerate or intensify cavitation in the machine. These phenomena are especially intensive in transient conditions of machine operation.

So far unique principles and regulations regarding the evaluation and methods of defining permissible levels of various cavitation symptoms have not been developed. This opinion concerns also principles of evaluation of cavitation damage as presented in publication No. 809, issued in 1978 by IEC (International Electrotechnical Commission) [10].

*) According to the IEC standards σ_0 denotes the lowest value of σ at which the influence of cavitation on the efficiency of a machine is not discernible, while σ_x denotes the value of σ for which the efficiency is $x\%$ lower than that corresponding to σ_0 (e.g. $\sigma_{0.5}$ denotes the value of σ at which the efficiency is 0.5% lower than the efficiency at σ_0). For particular cases of local increase of η with decreasing σ the nearest value of σ corresponding to the same efficiency as for σ_0 is denoted by σ_s (see fig. 3).

It was recommended in that publication that appropriate procedures should be based on agreements between the machine supplier and the customer (user), as it is known that substantially high values of σ are required to prevent damage to the machinery or keep it on a low level, and this means founding the machine sufficiently deep, which is associated with high costs of construction works. The following is subject to an agreement: the warranty period and the number of hours of machine operation to the damage check (8000 hours or 3000 hours), the mass- or volume loss of the material, or the area of surface damaged and the damage depth, the number of working hours needed for repair of the damage or the mass of electrodes used for regeneration of components damaged in the process of cavitation. All of these details to be agreed upon concern, of course, a specific value of the suction head or the stipulated σ number.

Also the permissible influence of cavitation on the performance characteristics of the machinery, and especially their power and efficiency, as well as the permissible level of vibroacoustic phenomena are the subject of bilateral agreements.

2. Cavitation diagnostics

The term „diagnostics”, which is equivalent literally to „recognition” or „discernment” is assigned various shades of meaning. It is understood as the detection of a phenomenon evaluation of its intensity, evaluation of the effect of the phenomenon on an object, or evaluation of the state of an object. The diagnostics of cavitation consists, first of all, in detecting the phenomenon and evaluating its intensity.

The diagnostics of cavitation in the hydraulic machinery is aimed at identifying the phenomenon and evaluating the state of the machine under consideration based on measurements of selected quantities characterizing its operation. Two areas may be distinguished in the domain of diagnostics investigations: those belonging to pure research and investigations aimed at specific applications. The first group covers investigations consisting in seeking parameters characterizing the operating conditions of a system corresponding to the inception of cavitation, cavitation development or to its maximum influence on a machine. The second group concerns in situ diagnostics of particular machines, carried out by means of diagnostic methods development in investigations of cognitive nature.

The majority of cavitation diagnostics methods used at present has been developed as a result of laboratory investigations of flow round isolated blades and other stationary obstacles. These methods may not be suitable for application in hydraulic machinery, due to the somewhat different development of cavitation (caused, among others, by nonuniformities of the stream field along the leading edge of blades and different boundary conditions) and differences of measurement conditions in a rotating system compared to those in a stationary one.

Three aspects of the phenomenon are of importance when describing cavitation and its effect on the hydraulic machinery. One should mention, first of all, localization, spatial dimensions, shape and structure of the cavitation cloud. Secondly, there is the disturbing influence of cavitation on the characteristics of energy transfer between the liquid and the blading of a machine. The dynamic effect of the phenomenon on the machinery, whose

result is erosion of flow passage walls and vibrations of machinery components, represents the third aspect. Relations between characteristics concerning the respective aspects are not unique. Also relations between parameters defining characteristics of each aspect are individual features of a machine.

Diagnostic methods may be direct or indirect. Direct diagnostic methods consist in observing the aspects of the phenomenon directly in a way permitting one to evaluate the actual intensity of the process. Usually these methods allow for qualitative as well as quantitative evaluation.

Indirect diagnostic methods make use of observations of phenomena accompanying the process under investigation and depending in some way on it. Conclusions as to the intensity of the process can be drawn, to some accuracy, based on characteristic properties of the phenomenon under investigation and the object in which it occurs. Indirect methods are usually easy to apply; they give, however, a qualitative picture only. The quantitative evaluation of the process investigated is possible only when the transmittance of the system for this process is known. Accurate determination of the actual transmittance is very difficult, the more so that it constitutes an individual feature of an object.

The methods of cavitation diagnostics used in practice may be divided into three groups:

- a) methods evaluating spatial properties of the cavitation cloud,
- b) those determining the effect of cavitation on power characteristics of the hydraulic machinery,
- c) methods defining the intensity of dynamic effect of cavitation on the machinery.

Methods belonging to group (a) are useful when one expects to find cavitationless working conditions of the hydraulic machinery. They may be useful for evaluation of the cavitation intensity, understood as an action on the object, when relations between parameters corresponding to the incipience of the phenomenon and those for which the respective aspects of cavitation have the maximum intensity are known.

Methods of group (b) may be considered as most useful when parameters defining the performance of a machine: the efficiency, total head and capacity of pumps or output of water turbines are the main criterion defining the usefulness of a machine to operation under given conditions.

Methods of group (c) are most appropriate when the parameters of machine operation should be determined for which erosion and vibrations of the machine are not harmful. Thus they are useful when durability and noiseless operation of a machine are important.

3. Exemplary results of diagnostic research

As it was already mentioned investigations of simplified fluid-flow arrangements subject to cavitation are not sufficient for development of effective methods of cavitation diagnostics in the hydraulic machinery. Thus it became necessary to conduct extensive cavitation research in model machines. Usually the methods of diagnostics developed in such a way are versatile enough to allow for applying them to diagnostics of any machine. This is not the case for the results of investigations. Frequently extension of results of model investigations onto the working conditions of prototypes fails, especially when the ratio of sizes of two geometrically similar machines, the prototype and its model, is very large.

Presented below are examples of diagnostic investigations of cavitation in the hydraulic machinery. First, results of laboratory investigations of centrifugal and diagonal pumps are discussed. These have been aimed at developing diagnostic methods. Next, results of cavitation investigations of a model reversible hydraulic machine are given. Finally application of diagnostic methods to investigations of Kaplan and Francis water turbines as well as pump/turbines is described. The results presented have been obtained at the laboratory of the Institute of Fluid-Flow Machinery of the Polish Academy of Sciences in Gdańsk and during field investigations ordered by the power industry authorities.

3.1. Investigations of cavitation in pumps

Centrifugal pump. Cavitation is likely to develop in centrifugal pumps under working conditions different from the rated ones (especially for the flow rate Q exceeding Q_{rat}) and in the presence of decreased pressure on the suction side. In such conditions cavitation results mainly in pump efficiency and total head drops and a rise of the vibration level. The first two symptoms show the way in which cavitation affects the conditions of energy exchange between the impeller and the liquid, while the latter symptom is an indicator of the intensity of dynamic influence of cavitation on the machine. All those symptoms appear in a case of fully developed cavitation.

Figure 4 shows diagrams of efficiency (η), total head (H), and acceleration of pump casing vibrations (a) versus the pump capacity (Q) for two different values of the suction

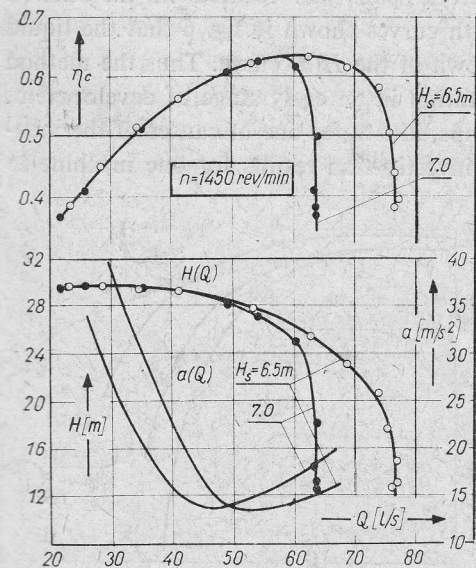


Fig. 4. Acceleration of the pump casing vibrations $a(Q)$ shown together with the head/discharge curve $H(Q)$ and the pumping set efficiency/discharge curve $\eta(Q)$ [20]

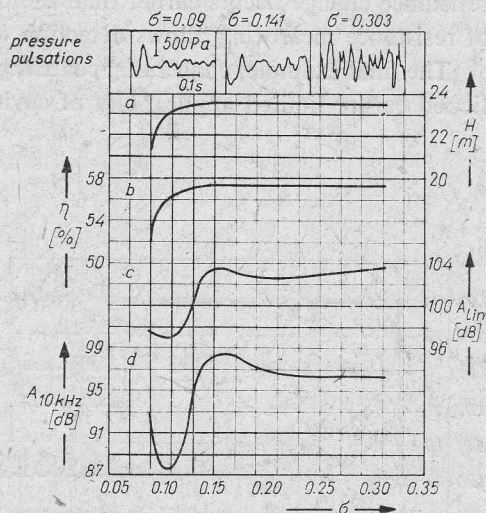


Fig. 5. The influence of cavitation number σ on the course of pressure pulsations in the flow and the total head (a), efficiency (b) and acceleration of vibrations (c and d) of a centrifugal pump [22]

head (H_s) [20]. The breakdown of the H and η curves indicates a highly developed cavitation. This is accompanied by a rise of the vibration acceleration value. The larger suction head H_s (which means lower pressure on the suction side) the lower is the value of pump capacity corresponding to the curve breakdown point.

Figure 5 illustrates, in turn, the influence of cavitation number (σ) changes on the pressure pulsation in the flow and the total head of a pump (a), its efficiency (b), acceleration of vibrations for the entire frequency spectrum (c) and at a frequency of 10 kHz (d). It can be seen from this figure that the rise of vibration level (especially marked for curve (d)) occurs before the breakdown of performance curves i.e. in an early stage of cavitation development. For cavitation intensive enough to disturb conditions of energy exchange in the impeller the vibration level decreases. The same is valid for pressure pulsations. Such a state is caused by large content of the gaseous phase in the cavitation cloud and decreased level of the dynamic intensity of cavitation (fig. 33).

Sometimes it is expected of the cavitation diagnostics to determine the working parameters of a machine that correspond to the appearance of cavitation in the flow and to allow evaluation of the cavitation region extent. In such a case methods based on measurements of cavitation effects in the machine become inadequate, in opposition to those utilizing measurements of certain physical properties of the liquid inside the cavitation cloud, such as absorption of ultrasound, light, gamma radiation or changes of the electric resistance.

Figure 6 shows the change of the resistance of liquid and the total head of a pump as functions of the cavitation number σ . Two pairs of electrodes were used for the measurements. One of them was placed in the suction nozzle of the pump, upstream of the inlet to the impeller. The second, compensating electrode pair was located in the suction reservoir. It is evident from the inspection of both curves shown in fig. 6 that the liquid resistance change occurs earlier than the breakdown of the $H(\sigma)$ curve. Thus the method of resistance measuring allows detection of cavitation in an early stage of development.

The final example, concerning centrifugal pumps, illustrates one of causes of the mentioned earlier limited applicability of cavitation investigation results for one machine to

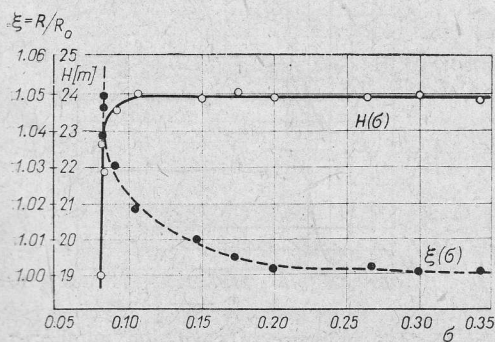


Fig. 6. Vibration of the relative resistance of liquid R/R_0 and the total head (H) of a pump as a function of the cavitation number σ [22]

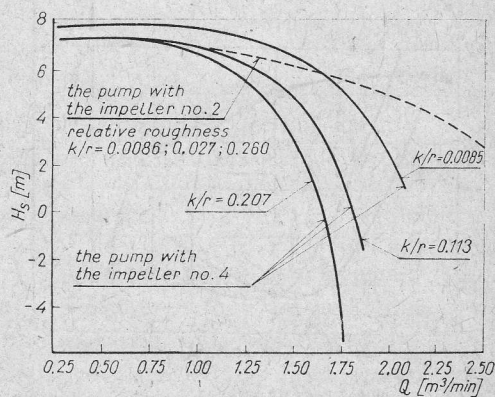


Fig. 7. Maximum suction heads for a pump with two model impellers at different levels of blade passage roughness [4]

another one. Figure 7 shows maximum suction heads for a pump with two impellers and different values of the blade passage roughness. It follows from the performance and cavitation characteristics obtained [4] that the change of the blade passage roughness of the impeller of a pump (in this case a high pressure one with the specific speed $n_q = 20 \div 28$) does not influence significantly its suction properties (impeller no. 2) so far as the inlet section is larger than that providing for high efficiency (most commonly applied). In the case of the impellers designed for high efficiency operation (impeller no. 4) an increase of roughness may influence unfavourably the suction properties of a pump. This is caused by cavitation which develops in the impeller more easily when the passage wall roughness is larger. It follows then also that different surface conditions of blade passage in geometrically similar impellers form one of the causes responsible for the lack of correspondence between cavitation characteristics of similar machines.

Diagonal pump. Complex investigations of cavitation phenomena in a diagonal pump have been carried out. Measurements were done in eleven series at different impeller speeds n and different capacities Q [11]. The working conditions of the pump were chosen so as to obtain kinematically similar flow conditions despite of the rotational speed changes ($\xi_Q = Qd^{-3}n^{-1} = \text{const}$). Figure 8 shows measurement series distributed over the set of iso-efficiency curves.

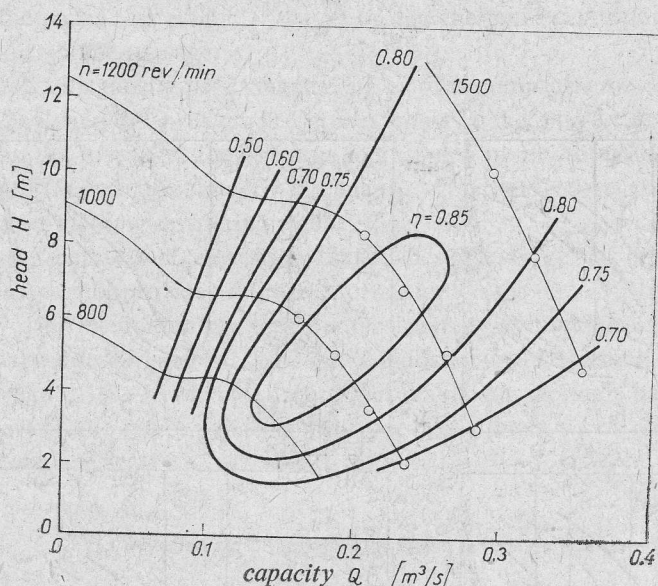


Fig. 8. Distribution of measurement series on the set of iso-efficiency curves for a diagonal pump [11]

Cavitation conditions were changed for the respective series of measurements by changing pressure in the hydraulic circuit.

Investigations consisted in visually inspecting and measuring the length of the cavitation cloud at one of impeller blades (to this end the impeller casing had been provided

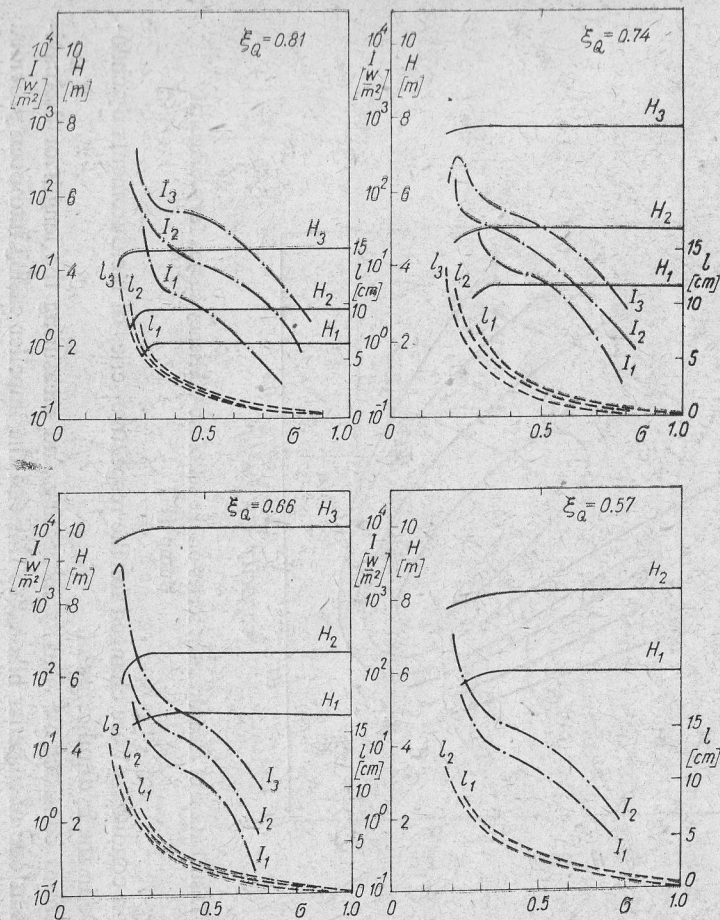


Fig. 9. Diagnostic characteristics for a diagonal pump, obtained by direct methods: the total head (H), the intensity of cavitation pulses (I), and the length of cavitation cloud (l) versus the cavitation number σ [11]

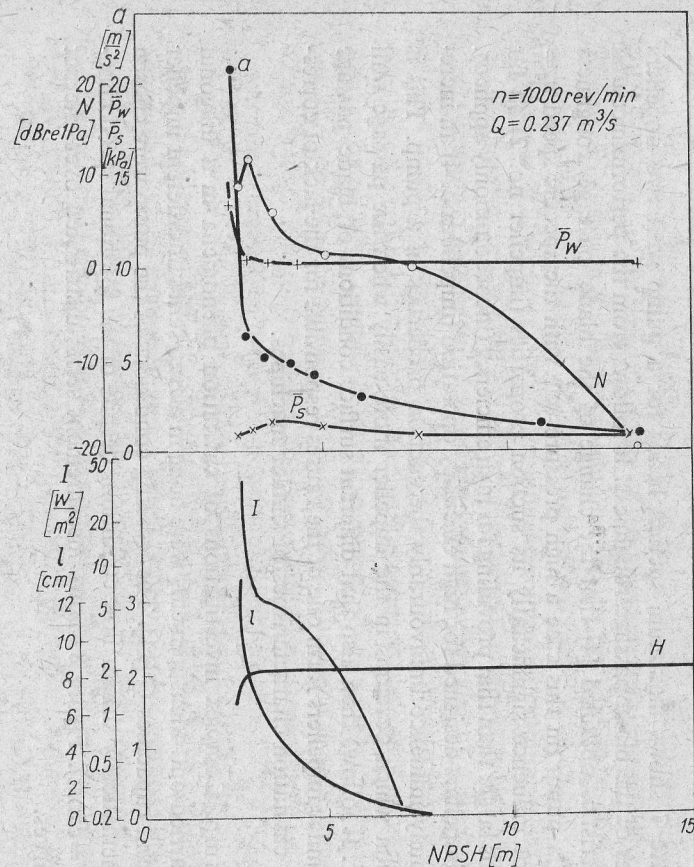


Fig. 10. An example of comparison of diagnostic cavitation characteristic for a diagonal pump at $n=1000 \text{ r.p.m.}$, $Q=0.237 \text{ m}^3/\text{s}$; characteristics obtained using direct methods are shown in the lower part of the figure, the upper part shows characteristics obtained by vibroacoustic indirect methods (a denotes the level of impeller casing vibration acceleration, N is the level of hydroacoustic noise at a frequency of 120 kHz, $\bar{P}_w$ — pressure pulsations within the impeller, $\bar{P}_s$ — pressure pulsations in the suction nozzle) [11]

with sight-glasses), measuring the total head and the intensity of cavitation pulses acting onto the impeller casing, as well as measuring the following vibroacoustic parameters: pressure pulsation, pump casing vibration and noise at the frequency of 120 kHz.

Results of measurements of cavitation pulses acting onto the impeller casing made it possible to estimate the energy of cavitation bubble collapse transferred to the wall defining the flow boundaries. The intensity of this process was expressed by a two-parameter quantity:

$$I = k \frac{1}{2\rho c} \frac{\lambda}{N} \sum_{i=1}^N n_i p_i^2,$$

where n_i denotes the number of cavitation pulses counted within the i th interval of amplitude values, p_i is the pressure amplitude in the i th interval, N is the total number of pulses, λ is the number of pulses per unit time, ρ is the liquid density, c is the speed of sound in the liquid, and k is a correction factor allowing for the fact that the size of the membrane of a transducer used for the measurements is two (three) orders of magnitude larger than the diameter of an average pressure pulse (it follows from measurements that $k \approx 5 \cdot 10^{-5}$).

It was assumed that the impeller casing surface at which pressure pulses were measured is representative for determination of dynamic cavitation loading of the impeller blade passages. The intensity of cavitation effects on the impeller under particular working conditions of the pump was estimated based on the energy of cavitation pulses calculated along the impeller blade passage.

Figure 9 shows diagnostic performance curves of a pump obtained by direct methods for $\xi_Q = \text{const}$, $\sigma = \text{var}$, and $n = \text{var}$. The curves concern the characteristics of the energy transfer between the rotor and the liquid as represented by the total head (H), the cavitation cloud length (l) and, calculated in the manner described above, the energy (I) which was transferred to the flow restricting walls.

Vibroacoustic methods of cavitation diagnostics, which belong to the category of indirect methods, do not provide accurate information on the intensity of the phenomenon. They may only describe, with a good or poorer accuracy, the variation of this parameter.

An exemplary comparison of different diagnostic characteristics of cavitation has been shown in figure 10. Curves based on direct diagnostic methods have been presented in the lower part of the figure, while in the upper part those based on indirect methods have been plotted.

3.2. Cavitation investigations of a model reversible machine

Cavitation investigations of model machines which are geometrically similar to prototypes designed for operation under actually existing field conditions are aimed at predicting cavitation properties of prototype machinery. The possibility of prediction is, however, limited because of the lack of sufficient knowledge about scale effects on cavitating flows.

Actually investigations of model machines reduce to specifying the operating parameters corresponding to appearance of cavitation at the rotor blades and those responsible for the beginning of cavitation influence on the efficiency/discharge curve. Additionally they may be supplemented with a cavitation cloud description.

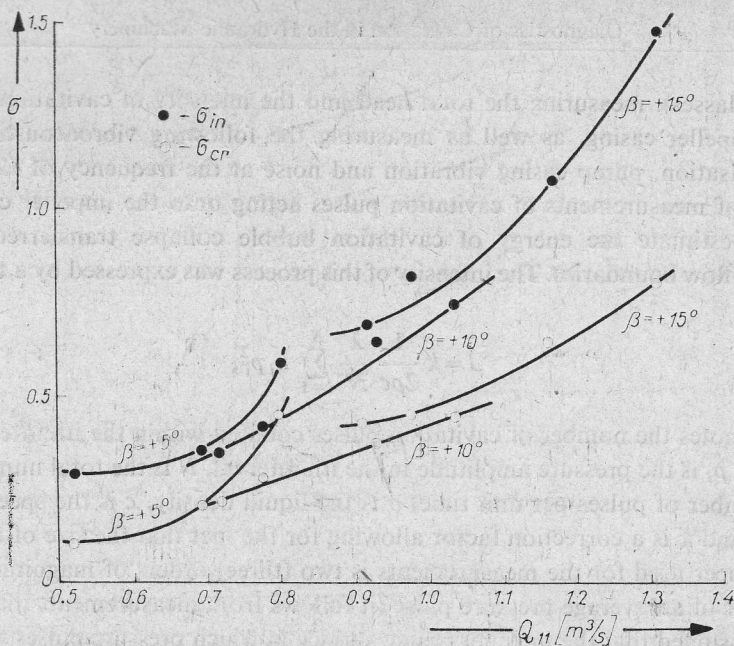


Fig. 11. Variation of the cavitation number values corresponding to the inception of cavitation, σ_{in} , and to first symptoms of influence on the efficiency characteristics, σ_{cr} , vs the flow rate Q and the runner blade angle setting β for a model pump/turbine unit of Deriaz type in the turbine mode of operation [25]

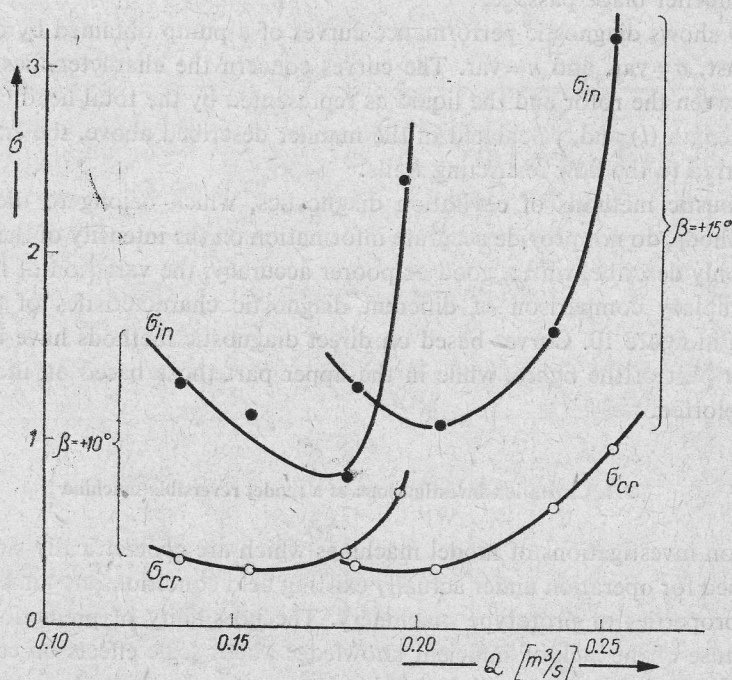


Fig. 12. Variation of the cavitation number values corresponding to the inception of cavitation, σ_{in} , and to first symptoms of influence on the efficiency characteristics, σ_{cr} , vs the flow rate Q and the runner blade angle setting β for a model pump/turbine unit of Deriaz type in the pumping mode of operation [25]

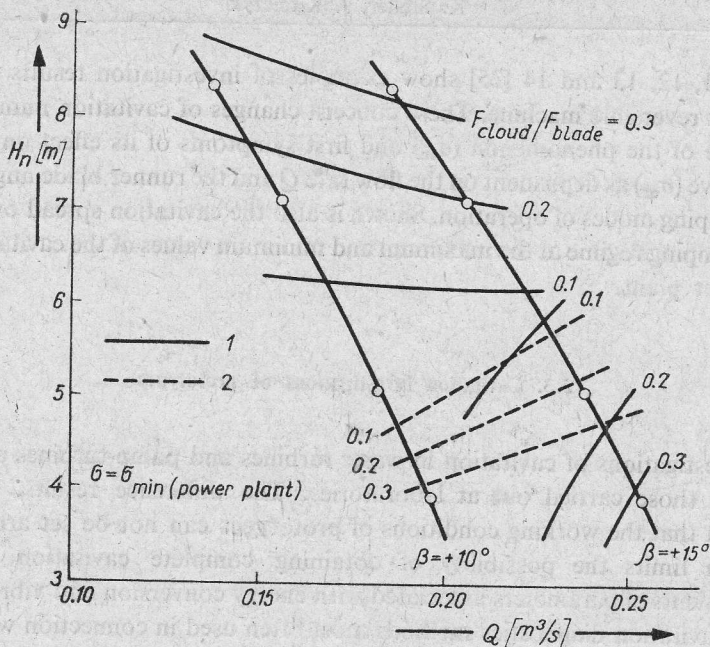


Fig. 13. Spreading of cavitation over runner blades of the model Deriaz pump/turbine unit at the minimum value of σ characterizing the power plant; pumping mode of operation [25]

1 — suction side of the blade, 2 — pressure side of the blade

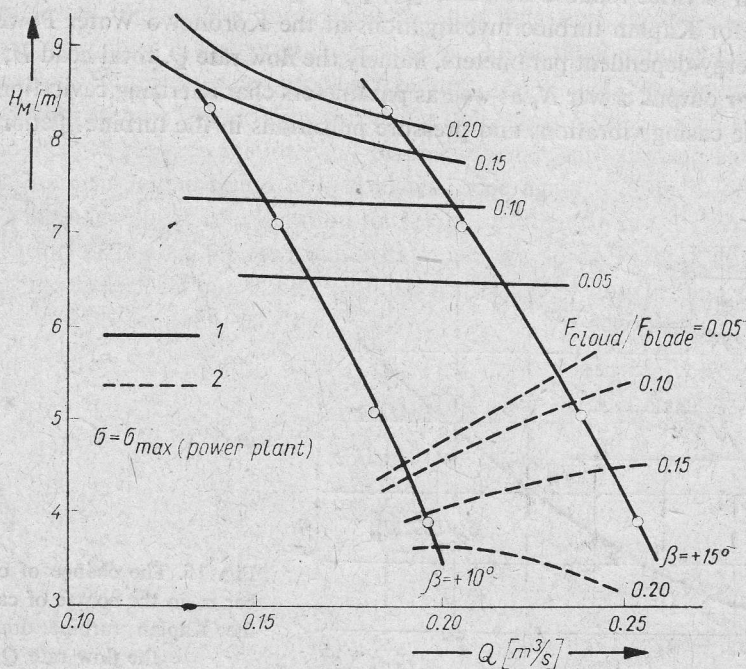


Fig. 14. Spreading of cavitation over runner blades of the model Deriaz pump/turbine unit at the maximum value of σ characterizing the power plant; pumping mode of operation [25]

1 — suction side of the blade, 2 — pressure side of the blade

Figures 11, 12, 13 and 14 [25] show examples of investigation results obtained for a model Deriaz reversible machine. These concern changes of cavitation numbers indicating the incipience of the phenomenon (σ_{in}) and first symptoms of its effect on the efficiency/discharge curve (σ_{cr}) as dependent on the flow rate Q and the runner blade angle β in the turbine and pumping modes of operation. Shown is also the cavitation spread over runner blades in the pumping regime at the maximum and minimum values of the cavitation number σ for the power plant.

3.3. Cavitation investigations of prototypes

Field investigations of cavitation in water turbines and pump-turbines are much more difficult than those carried out at laboratories. The difference results, among others, from the fact that the working conditions of prototypes can not be set arbitrarily at any point, which limits the possibility of obtaining complete cavitation characteristics.

Measurements of parameters associated with energy conversion and vibroacoustic parameters are cavitation diagnostics methods most often used in connection with prototypes.

Kaplan turbines. Localization of cavitation in a Kaplan turbine depends on its intensity. This is illustrated in figure 15 showing the change of cavitation number σ_c as a function of twice reduced flow rate Q_{11} [6]. Figure 16 is a schematic of the measuring set-up used for Kaplan turbine investigations at the Koronowo Water Power Plant. The principal energy-dependent parameters, namely the flow rate Q , total head H , suction head H_s , generator output power N_e as well as parameters characterizing cavitation — the noise level, turbine casing vibrations and pressure pulsations in the turbine suction region were measured.

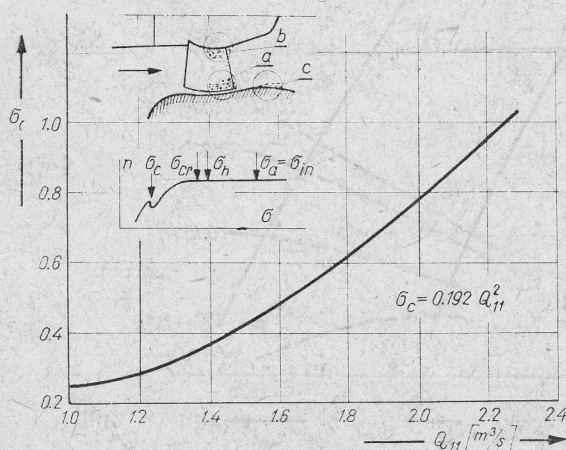


Fig. 15. The change of cavitation number σ_c in the course of cavitation within the Kaplan turbine draft tube versus the flow rate Q_{11} [6]

Figure 17 shows the turbine casing vibration amplitude as a function of the output while in figure 18 turbine vibration spectra are shown for different loads of the hydroelec

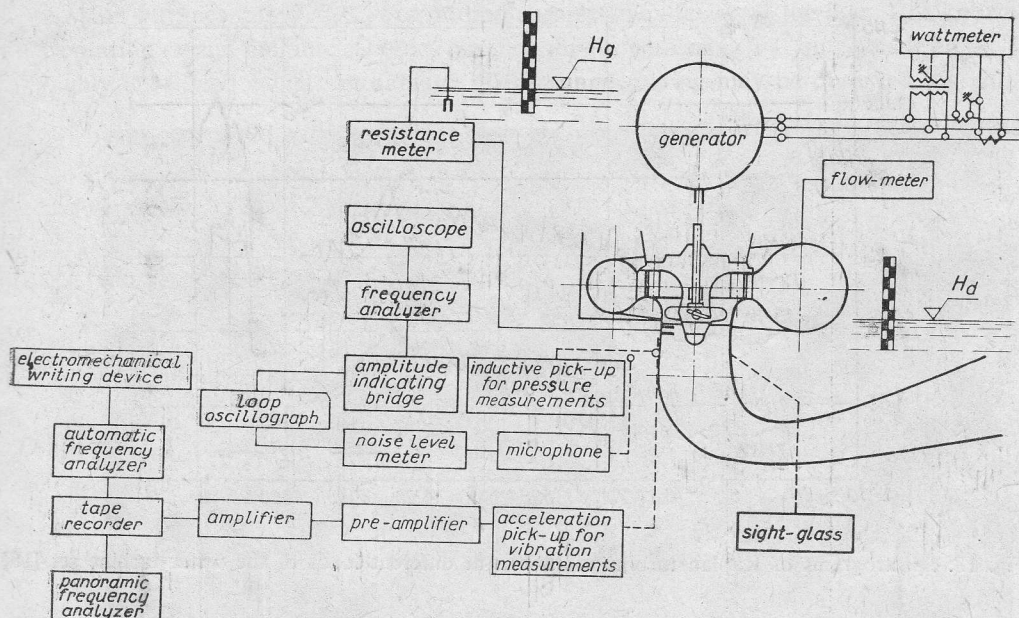


Fig. 16. Measuring equipment used for investigations of a Kaplan turbine [18]

tric generating set. Figure 19, in turn, shows pressure pulsations downstream of the turbine runner as dependent on the generating set loading. All of them indicate high degree of cavitation development for turbine outputs exceeding 13 MW. This was confirmed by a visual inspection of the turbine runner done through a sight-glass in the suction tube.

Correspondence between the intensity of phenomena characterizing cavitation and its actual development in the liquid flow confirms once again usefulness of the cavitation diagnostics methods under consideration for relative evaluation of the cavitation intensity.

Localization of regions affected by cavitation is not so easy. Most effective, however seldom feasible under field conditions, is visual inspection of flow around the runner. For

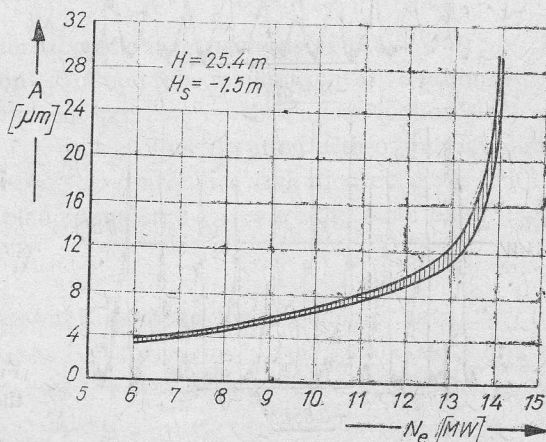


Fig. 17. Amplitude of Kaplan turbine casing vibrations versus the water turbine set output [17]

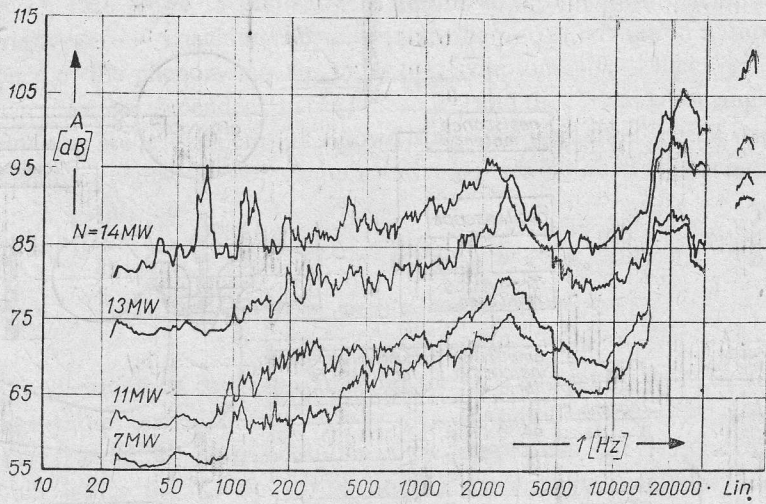


Fig. 18. Spectrograms of Kaplan turbine vibrations at different loads of the water turbine set [18]

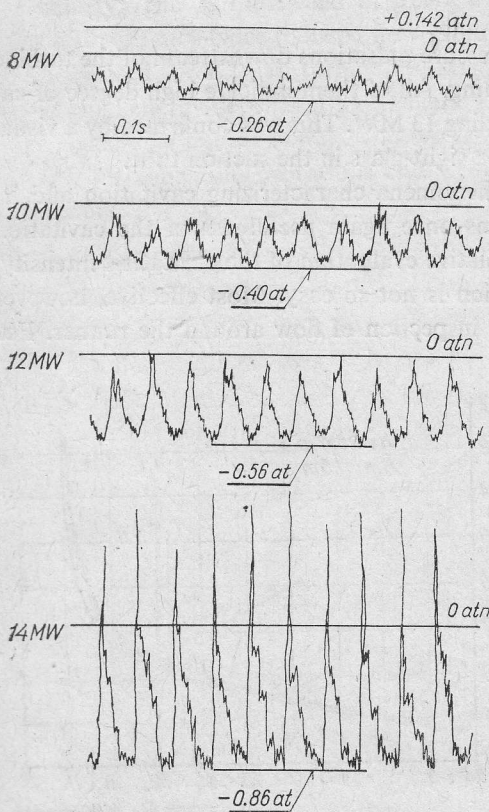


Fig. 19. Pulsations of pressure downstream of the Kaplan turbine runner at different loads of the water turbine set [18]

Kaplan turbines knowledge of cavitation specifics may facilitate the task. It is known that cavitation occurs first in clearances between the runner casing and outer blade edges. For a highly developed cavitation damage to the runner casing may be observed (fig. 20).

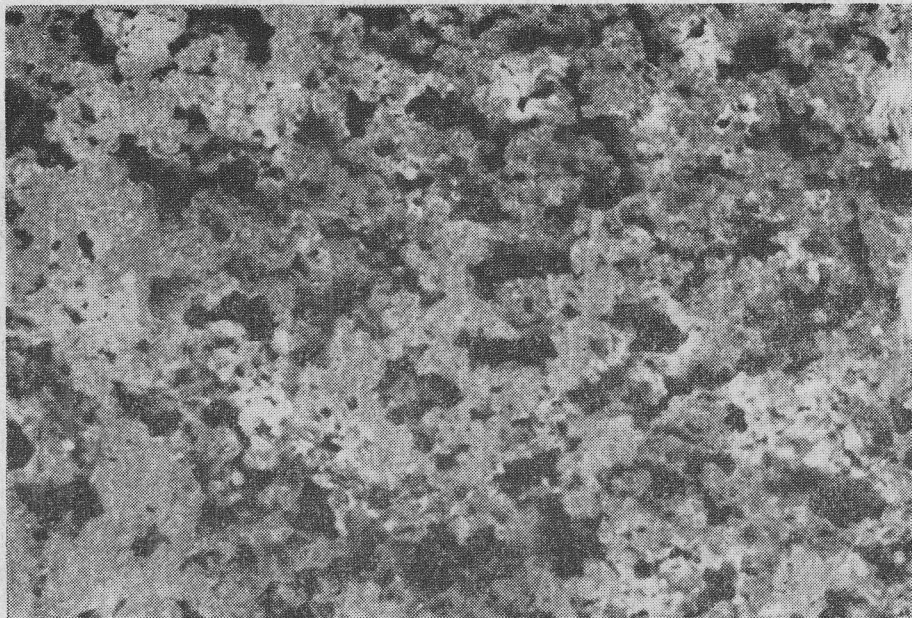


Fig. 20. A part of a Kaplan turbine runner casing damaged by clearance cavitation [21]

Francis turbines. Francis turbines are less susceptible to cavitation than Kaplan turbines. Localization of cavitation regions within the runner is different, too (fig. 2). Inspection of erosion effects shows that the phenomenon is most intense near the outlet edges at the suction sides of the runner blades (fig. 21). There are Francis turbines in which cavitation damage has occurred to both the suction and pressure sides of the runner blades. An example is shown in figure 22. The damage proved to be harmful as cavitation pits, at which stress concentration took place, facilitated the occurrence and propagation of cracks (fig. 23).

Figure 24 gives an example of vibroacoustic investigations of Francis turbine. Shown is the relation between the acceleration of suction tube manhole cover vibrations (the cover acting as a diaphragm sensitive to dynamic pressure on the runner suction side) and the hydroelectric generating set output. The course of acceleration with output is characteristic of Francis turbines and shows that in these machines cavitation occurs below 30 MW and above 40 MW. Relatively large blading material losses are the proof of cavitation occurrence in the machine.

Reversible machines (pump-turbines). In these machines cavitation is most intense in the pumping mode of operation. It occurs mainly on the suction side of the runner near blade edges. Figures 25 and 26 show the distribution of cavitation damage over the runner of a Francis pump-turbine.

In reversible machines guide vanes are affected by cavitation, too. Figure 27 shows cavitation pits that developed on the upstream part of a vane during the pumping operation.

Usually turbine operation gives rise to less intense cavitation.

Figures 28, 29 and 30 present results of vibroacoustic investigations of cavitation in reversible machines installed at the Żydowo Pumped-Storage Water Power Plant.

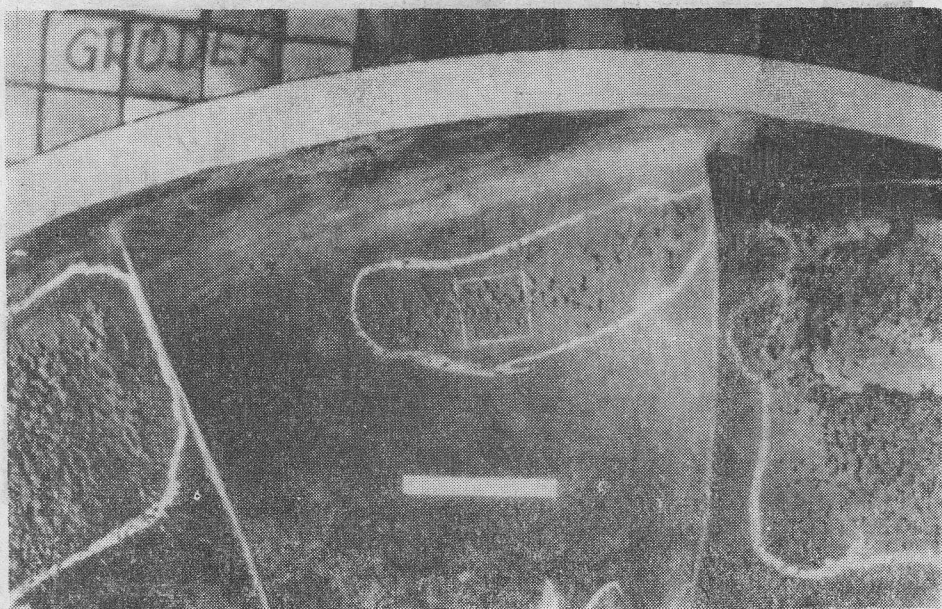


Fig. 21. Damaged surfaces of Francis runner blades [21]

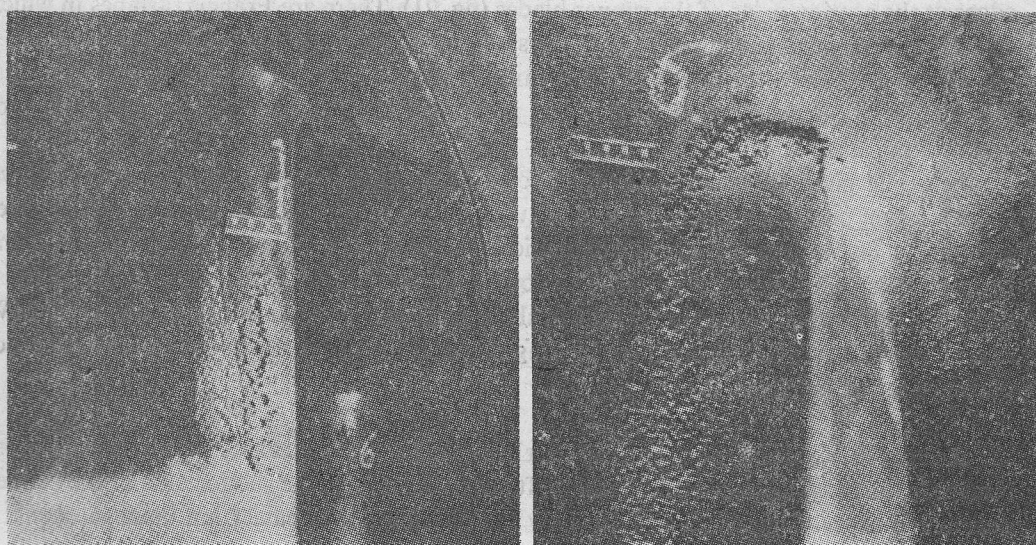


Fig. 22. Parts of a Francis runner with damaged leading (a) and trailing (b) edges [21]

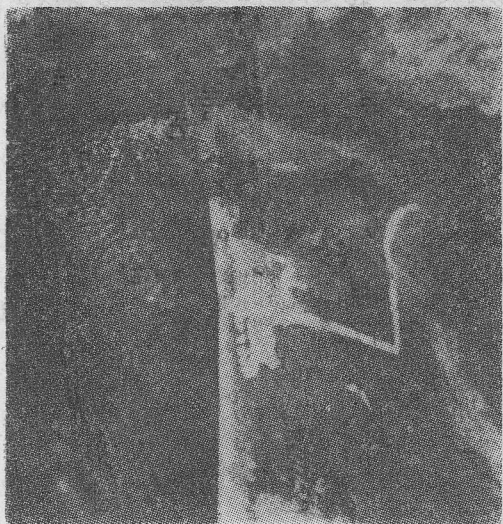


Fig. 23. Cracked blade of a Francis runner [21]

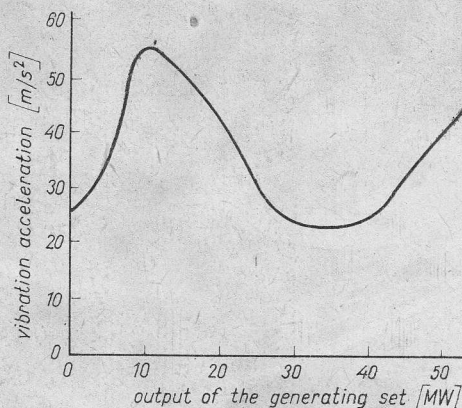


Fig. 24. Vibration acceleration of a manhole in a Francis turbine draft tube as a function of the water turbine set loading [1]

Shown in figure 28 are instantaneous values of pressure pulsations in a guide vane passage for the pumping and turbine modes of operation. The waveform differences are not distinct enough to provide information about cavitation in the machine. It follows then that measurements of vibroacoustic parameters on the pressure side of the runner are not fully useful for cavitation diagnostics.

Figure 29 shows instantaneous values of displacement amplitude measured on vibrating draft tube manhole. The intensity variation of the process is distinct, both for load changes in the turbine operation and after switching to the pumping operation.

Figure 30 illustrates, in turn, curves of vibration acceleration of the draft tube manhole cover and hydroacoustic noise measured under the runner at 100 kHz frequency. These characteristics are not closely related, however certain analogies between them are visible. Both exhibit a drop of values measured at 50 - 80% guide vane opening and rapid rise for opening exceeding 80%. Characteristic of both of them is also a high intensity of vibrations and noise in the pumping mode of operation compared to those in the turbine mode of operation.

Figures 29 and 30 present seemingly contradicting results as to the signal levels in the pumping and turbine modes of operation at 50 - 80% guide vane opening. It should be remembered, however, that in the spectrum of a wide-band acceleration signal the maximum is located somewhere between several and dozen or so kilocycles, while in the displacement spectrum the maximum appears at low frequency [3]. Thus the vibration acceleration level is a better measure of dynamic phenomena occurring as a result of high frequency inputs.

The following interpretation of figures under consideration can therefore be given: the maximum in figure 29 corresponds to development of a vortex under the rotor in the turbine mode of operation, while the level of vibrations shown in figure 30 depends on the cavitation intensity.

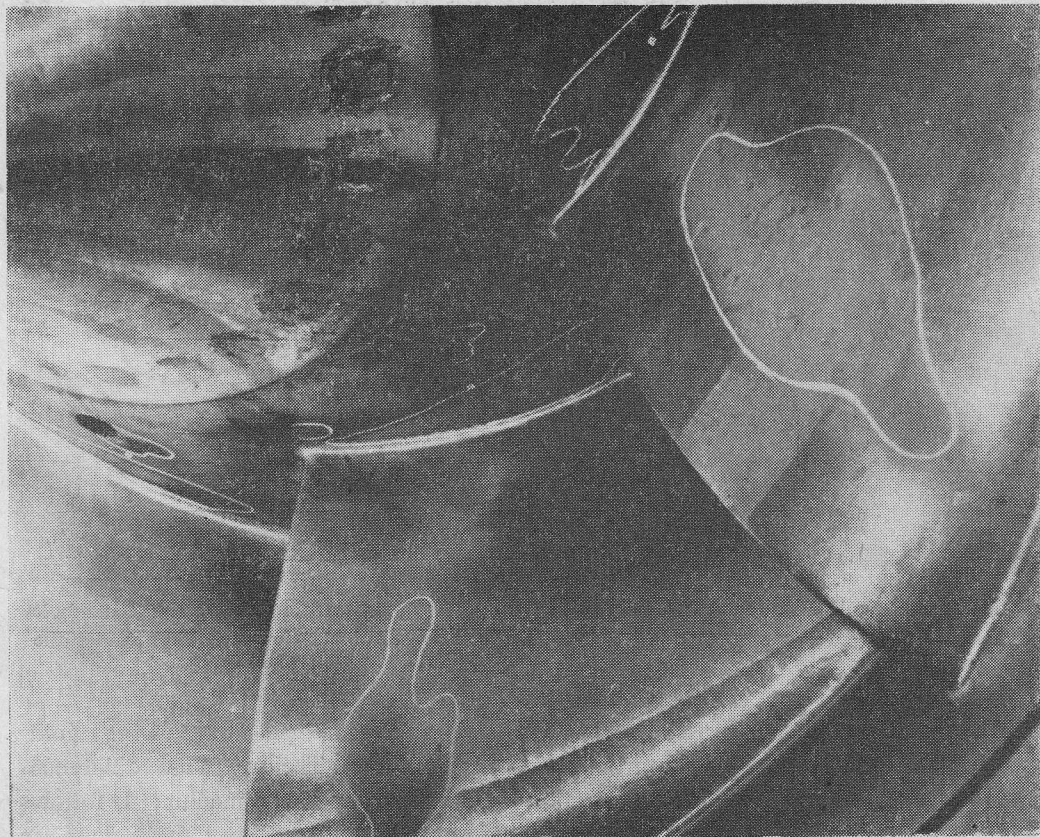


Fig. 25. Damaged surfaces of the runner in a reversible hydraulic machine of Francis type [21]

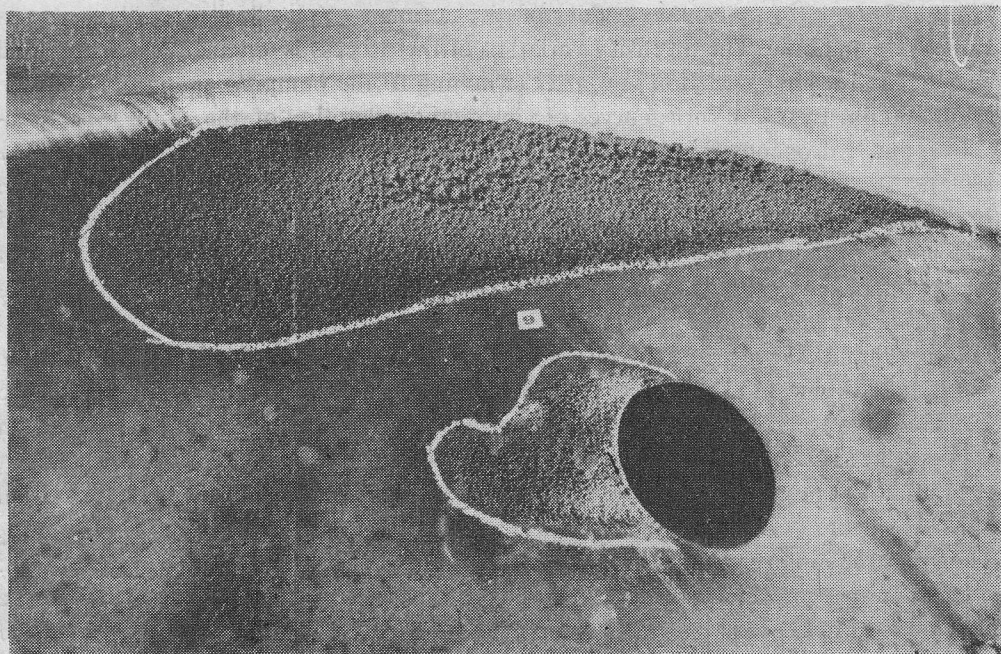


Fig. 26. Cavitation pitting of a reversible machine runner hub [21]

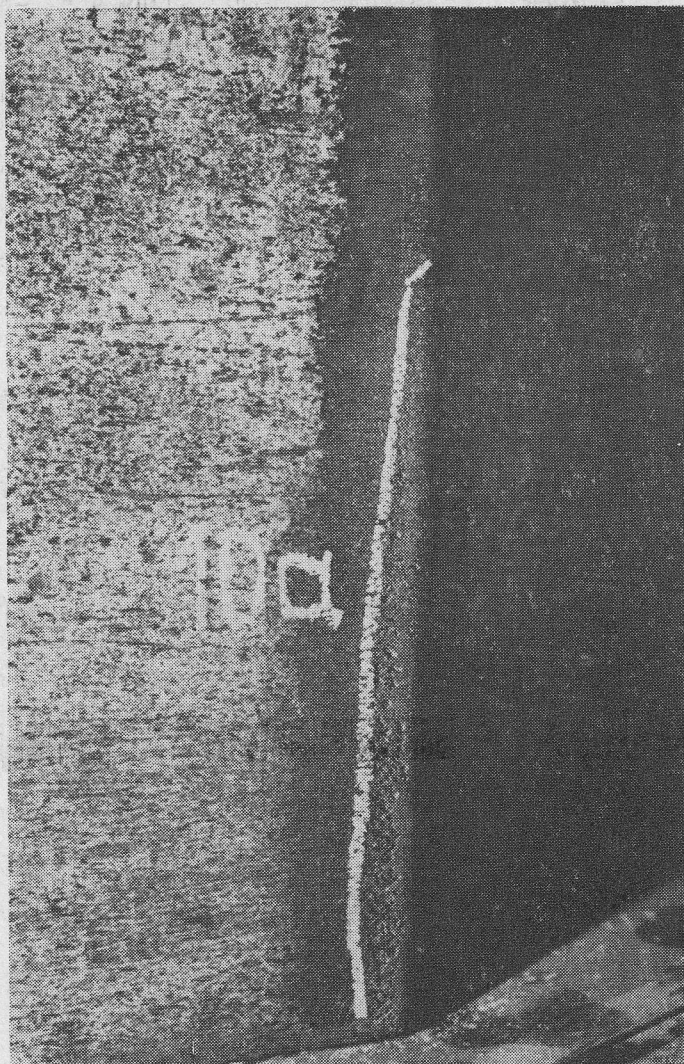


Fig. 27. Cavitation pitting of a reversible machine guide vane [21]

The above example shows that it is not easy to make conclusions about the intensity of cavitation in a hydraulic machine based only on measurements of vibroacoustic parameters. This must be done very carefully and concern at most a qualitative evaluation of vibrations of the dynamic process intensity.

4. Relation between various cavitation symptoms in water turbines and impeller pumps

Determination of permissible working parameters of hydraulic machinery, done from the viewpoint of some established cavitation criteria, is one of the most important purposes of cavitation diagnostics. This requires that relations between different cavitation aspects as discussed previously be defined.

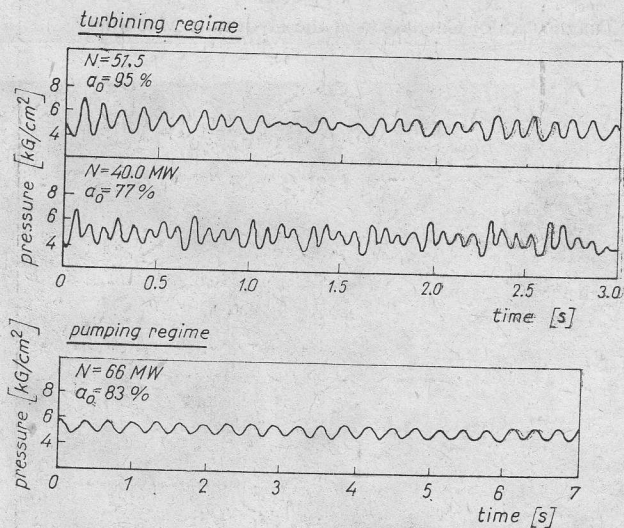


Fig. 28. Pulsations of pressure in a blade passage of the reversible machine guide-apparatus for the turbine and pumping modes of operation [2]

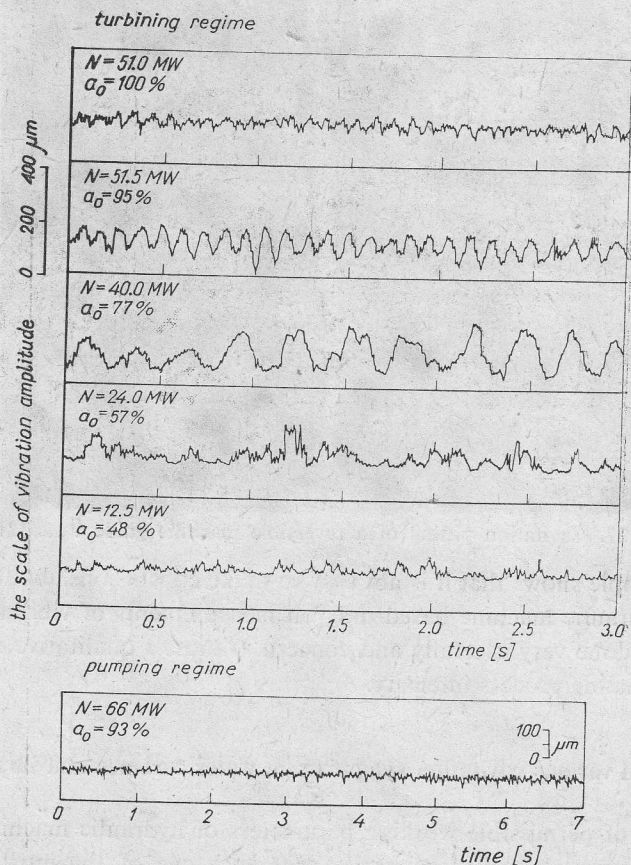


Fig. 29. Displacement during vibrations of a reversible machine draft tube manhole cover for different loads of the turbine set in the turbine and pumping modes of operation [2]

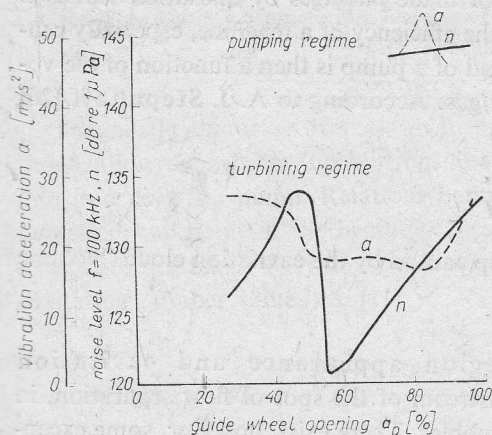


Fig. 30. Vibration acceleration of the draft tube manhole cover and hydroacoustic noise measured under the runner at a frequency of 100 kHz, versus the reversible machine guide vane opening (according to [1])

4.1. Relations between aspects of cavitation phenomena

Any aspect of cavitation, such as the form and extent of the phenomenon, its influence on the energy transfer between the liquid and the blading, as well as the erosive and dynamic effects on the flow arrangement may be described by some specific characteristics. The respective characteristics of a given fluid-flow arrangement are related to each other but experimental investigations are necessary to define these relations.

Relation between cavitation and the machine efficiency. The influence of cavitation on the energy exchange is evident, as was already pointed out, when the phenomenon is strongly developed. This occurs due to a rise of drag and drop of lift (mainly in machines characterized by high specific speeds). The dependence is illustrated in figure 31 showing the lift coefficient c_z and drag coefficient c_x for a profile as functions of the cavitation number σ .

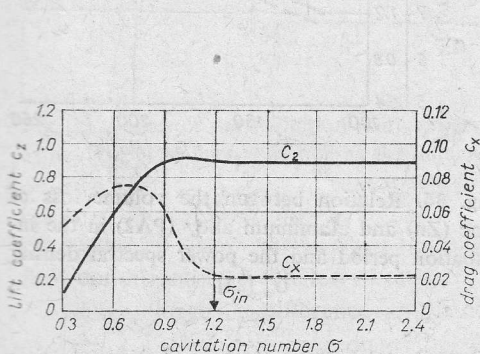


Fig. 31. The change of the lift coefficient c_z and drag coefficient c_x for Clark Y 11.7 profile set at an angle of incidence equal to 3° , as dependent on the cavitation number [14]

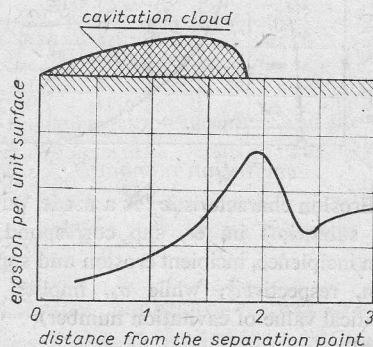


Fig. 32. Relation between erosion and the cavitation zone [12]

Decrease of the effective cross section of rotor blade passages by cavitation clouds is another reason of the influence of cavitation on the efficiency of a machine, especially centrifugal pumps. The drop of efficiency or total head of a pump is then a function of the volume flux of the rotor space occupied by vapour/gas. According to A. J. Stepanoff [26] the following formula is satisfied

$$\frac{\Delta H}{H} = K \frac{Q_s^2}{H},$$

where H is the total head and ΔH is the head drop caused by the cavitation cloud "volume flux" Q_s .

Relation between the cavitation region appearance and cavitation erosion. Material damage occurs mainly downstream of the spot of flow separation, in the "collapse region" of majority of cavitation bubbles. There exist, however, some exceptions, e.g. smooth and transparent cavitation clouds resembling supercavitation cavities (sheet cavitation). In such a case damage to walls restricting the flow may occur upstream of the cloud separation spot.

Figure 32 shows a typical relation between erosion and cavitation regions. It is evident that the most severe damage takes place at the end of the cavitation region.

Relation between cavitation intensity and cavitation erosion. Interesting is also the relation between the material damage rate and the cavitation development stage as expressed by the σ number. The relation, in the form of $\Delta m/t = f(\sigma)$ curve, is shown in figure 33.

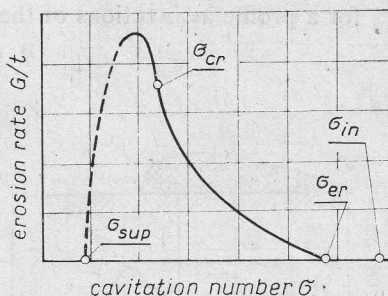


Fig. 33. Erosion characteristic for a model pump [7] (The subscripts in, er, sup correspond to cavitation incipience, incipient erosion and supercavitation, respectively, while σ_{cr} implies the critical value of cavitation number)

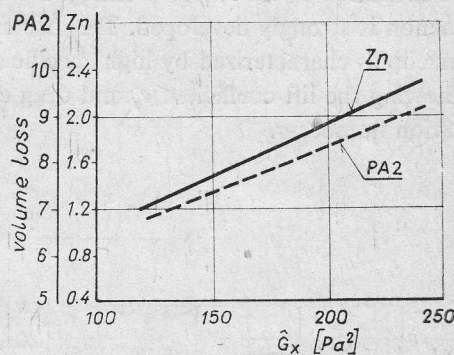


Fig. 34. Relation between the volume loss of zinc (Zn) and aluminium alloy (PA2) in the incubation period and the power spectral density $\hat{G}_*$ [24]

Another example of relation between the cavitation intensity and cavitation erosion is given in figure 40 (for details see Section 5). Similar relation is shown in figure 34 illustrating the dependence of the material damage on the spectral power density in the so called initiation period.

4.2. Relations between characteristic features of cavitation in a hydraulic machine, shown for a diagonal pump as an example

Diagnostic characteristics shown in figure 9, which have been obtained by direct methods, allow one to draw conclusions about relations between respective aspects of cavitation in a diagonal pump. Relations between values of σ corresponding to the cavitation inception, the point of the head-discharge curve breakdown and appearance of pressure pulses are not fixed. They change depending on flow conditions. The ratios of respective cavitation number values are [11]

$$\frac{\sigma_{in}}{\sigma_{0.5}} = 2.75 \div 5.50, \quad \frac{\sigma_{in}}{\sigma_{dyn}} = 1.1 \div 1.8, \quad \frac{\sigma_{dyn}}{\sigma_{0.5}} = 2.5 \div 3.0$$

(σ_{in} is the cavitation number value corresponding to the cavitation inception, $\sigma_{0.5}$ corresponds to the beginning of influence on the head-discharge curve, and σ_{dyn} corresponds to the beginning of dynamic effects on the pump).

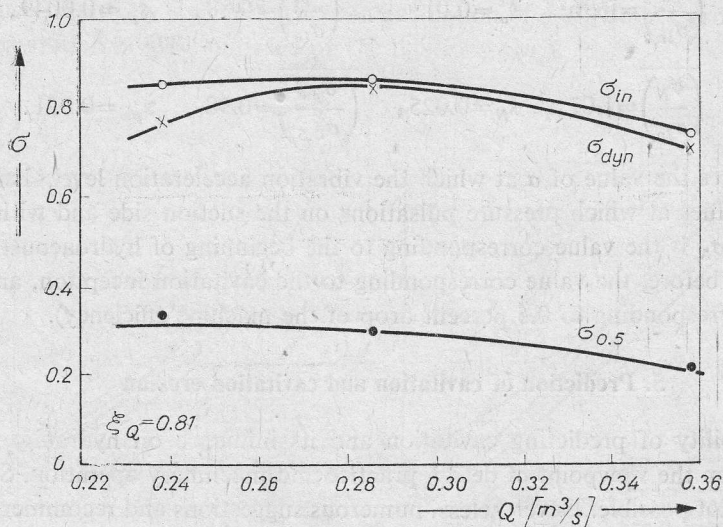


Fig. 35. Values of the cavitation number corresponding to the cavitation inception and the beginning of its dynamic influence as well as the influence on power characteristics, versus the capacity of pump at variable speed ($\xi_Q = 0.81 = \text{const}$) [11]

Figure 35 shows variation of the relations versus flow rate for kinematically constant flow conditions $\xi_Q = \text{const}$, $n \neq \text{const}$.

Equally large are differences in the course of intensity changes for respective cavitation aspects. Dynamic influence on pump walls appears for weak cavitation and rises at variable rate as σ decreases. Highly developed cavitation and rapid rise of pressure pulse intensity correspond to parameters critical with respect to the total head characteristic.

It should be noted that cavitation characteristics of kinematically similar flows are similar.

Worth noting is also the dependence of vibroacoustic pump characteristics on the cavitation intensity. An analysis based on 11 series of measurements has led to the conclusion that the following may give relatively good information on the cavitation inception in a pump: acceleration of the machine rotor casing vibrations, high frequency hydroacoustic noise, and pressure pulses in the suction region, while pressure pulses within the pump impeller indicate the beginning of the efficiency-discharge curve breakdown.

In order to evaluate the accuracy of the respective diagnostic methods mean values of σ_y/σ_x ratios have been calculated (here σ_y denotes the value of cavitation number corresponding to discernible changes in the course of the diagnostic characteristic under consideration, and σ_x is the value corresponding to the appearance of the cavitation aspect having been taken into account) together with the variance s_y^2 . The following formulae were used:

$$\overline{\left(\frac{\sigma_y}{\sigma_x}\right)} = \frac{1}{n} \sum_{k=1}^n \left(\frac{\sigma_y}{\sigma_x}\right)_k, \quad s_y^2 = \frac{1}{n} \sum_{k=1}^n \left(\frac{\sigma_y}{\sigma_x}\right)_k^2 - \left(\overline{\left(\frac{\sigma_y}{\sigma_x}\right)}\right)^2.$$

The results were as follows:

$$\begin{aligned} \overline{\left(\frac{\sigma_a}{\sigma_{in}}\right)} &= 0.96, & s_a^2 &= 0.013, & \overline{\left(\frac{\sigma_{ps}}{\sigma_{in}}\right)} &= 0.88, & s_{ps}^2 &= 0.0049, \\ \overline{\left(\frac{\sigma_N}{\sigma_{in}}\right)} &= 1.07, & s_N^2 &= 0.025, & \overline{\left(\frac{\sigma_{pw}}{\sigma_{0.5}}\right)} &= 0.98, & s_{pw}^2 &= 0.021. \end{aligned}$$

(here σ_a denotes the value of σ at which the vibration acceleration level changes, σ_{ps} and σ_{pw} denote values at which pressure pulsations on the suction side and within the pump impeller rise, σ_N is the value corresponding to the beginning of hydroacoustic noise level rise, σ_{in} is, as before, the value corresponding to the cavitation inception, and $\sigma_{0.5}$ is the value of σ corresponding to 0.5 percent drop of the machine efficiency).

5. Prediction of cavitation and cavitation erosion

The capability of predicting cavitation and its influence on hydraulic machinery is invaluable from the viewpoint of design practice and machinery operation. So far reliable prediction is not possible. Nevertheless, numerous suggestions and recommendations as to the approach to this problem have been formulated. This concerns cavitation prediction for a machine of normal size based on results of model investigations, as well as the possibility of predicting cavitation erosion judging on the behaviour of material specimens in laboratory conditions. One can refer to F. G. Hammitt's opinion expressed in his report "The State of the Art and Predicting Capability" [8]: „Prediction of the onset and extent of cavitation or liquid impact erosion can be attempted using theoretical and experimental techniques. However one of the greatest problems today in the liquid erosion field is our present inability to predict erosion rates in field machines from laboratory tests or theoretical models". The said predicting capability results from doubtful value of cavitation intensity evaluation under field conditions, insufficient knowledge of scale effect influence on the results of cavitation investigations and difficulties in correct modeling of cavitation phenomena.

In what follows our remarks about prediction of cavitation and cavitation erosion will be confined to a brief review of IEC Standard recommendations [9], and presentation of investigation results obtained by other investigators as well as our own results. These latter concern mainly cavitation erosion, which has been the subject of research at the Institute of Fluid-Flow Machinery, Polish Academy of Sciences, in Gdańsk, for dozen or so years.

5.1. Predicting cavitation in field conditions

The Thoma number is the quantity characterizing cavitation properties of a hydraulic machine and the risk of cavitation occurrence inside. The value of σ depends on the machine specific speed (see Section 1), but design and manufacturing differences between machines of identical specific speeds (e.g. workmanship quality, roughness of passage walls) give rise to doubts as to the value of σ as a criterial quantity. This means that conclusions about the cavitation hazard based on the relation $\sigma(n_q)$ are not reliable enough. The diagrams in figure 36 confirm this opinion. They show permissible values of cavitation number $\sigma = \sigma_{\text{perm}}$ as recommended by various authors for Francis and Kaplan turbines of different specific speeds. Obviously, relatively large differences in evaluation of σ_{perm} do not provide a reliable basis for cavitation prediction based on the relations $\sigma(n_q)$.

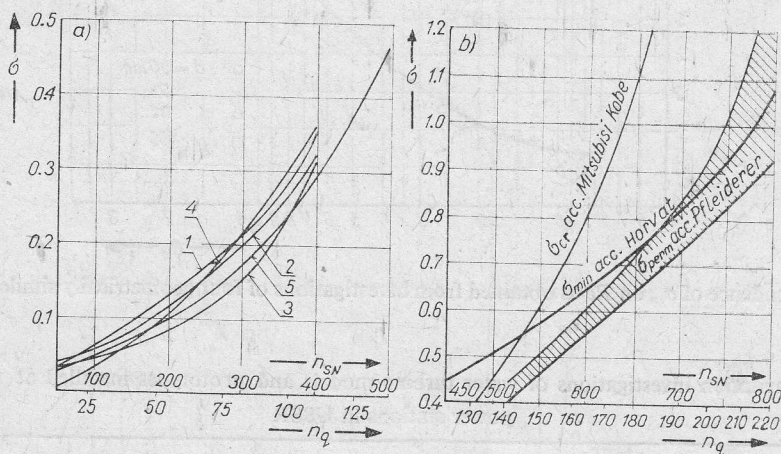


Fig. 36. Permissible values of the cavitation number σ for Francis (a) and Kaplan (b) turbines according to various sources [18]

1 — acc. to LMZ, 2 — acc. to Morozov, 3 — acc. to Thoma, 4 — acc. to French standards, 5 — acc. to Pfeleiderer

One can also try to predict cavitation in a machine on the basis of model investigations. Such approach has been presented in the IEC publication, reference [9]. It should be explained that the method has been based on the equality:

$$(\sigma_{cr})_M = (\sigma_{cr})_P$$

meaning that the cavitation number σ is equal for both the model (M) and prototype (P) provided that the Froude similarity criterion.

$$H_M/H_P = D_M/D_P$$

is satisfied during model investigations. Here H is the total head, and D is the nominal rotor diameter. Where the Froude condition is not satisfied (for $H/D < 3$) it is recommended that appropriate calculations following rules given in [9] be carried out. When the Froude condition is satisfied or when $H/D > 3$ one can assume that $\sigma_P = \sigma_M$. It follows then

$$(\text{NPSH})_P = (\text{NPSH})_M \left(\frac{n_P}{n_M} \right)^2 \left(\frac{D_P}{D_M} \right)^2.$$

It should be emphasized once again that opinions are divided as to the manner of applying results of model investigations to natural conditions. This is so because of the existence of the scale effect, discovered in thorough investigations. Illustration of this effect may be found in figure 37, showing σ_{in} as a function of the Reynolds number obtained from investigations of four geometrically similar rotors [17]. On the other hand one can refer

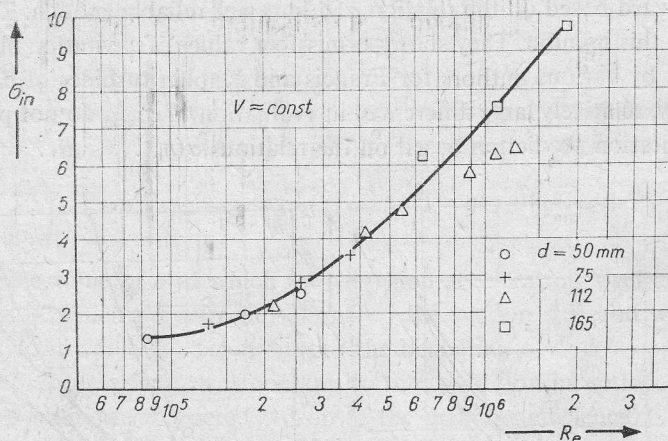


Fig. 37. Dependence of σ_{in} on Re as obtained from investigations of four geometrically similar rotors [17]

Table I

Results of cavitation investigations of water turbine models and prototypes installed at some water power stations in USSR

Water power plant	Turbine data	$(\sigma_{cr})_M$	$(\sigma_{cr})_P$	$\frac{(\sigma_{cr})_P}{(\sigma_{cr})_M}$
Tismlanskaya	PL-495 $H=22.2$ m $H_S=1.45 \div 4.15$ m $P=44 \div 53$ MW	$0.30 \div 0.41$	$0.27 \div 0.39$	$0.9 \div 0.95$
Kaunasskaya	PL-661 $H=17.9 \div 20.7$ m $H_S=0 \div 0.12$ m $P=24 \div 27$ MW	$0.48 \div 0.52$	$0.47 \div 0.55$	$0.98 \div 1.06$
Volzhskaya	PL-587 $H=21.5 \div 24$ m $H_S=-2.6 \div 0.95$ m $P=116 \div 122$ MW	$0.57 \div 0.64$	$0.46 \div 0.58$	$0.82 \div 0.91$
Verebnie Tulomskaya	PL-646 $H=59.1 \div 61.2$ m $H_S=-4.0 \div 3.8$ m $P=70$ MW	$0.23 \div 0.25$	$0.23 \div 0.24$	$0.96 \div 1.0$

to cases confirming the adequacy of cavitation prediction for normal size machinery based on flow observations for model machines. Data given by A. S. Lashkov [13] may be regarded as an example. We quote in table I after his work a comparison of model and prototype investigations carried out for machines installed at some water power plants in the Soviet Union.

5.2. Cavitation erosion prediction

Damage to hydraulic machinery components caused by cavitation (figures 21 ÷ 23, 25 ÷ 27) depends, first of all, on the intensity of the phenomenon, the kind of material, and physico-chemical properties of the liquid. The cavitation intensity is defined as an average of the energy flux generated by cavitation bubble implosion and acting onto a given area.

As a rule the cavitation resistance of materials is evaluated based on specimen testing in laboratory conditions. Prediction of material behaviour in field conditions, different than those during testing, is often inaccurate, if not erroneous, because of differences in flow features and cavitation conditions.

Numerous experiments have shown [5] that the method of testing affects the material evaluation. See for instance results of investigations carried out at the Institute of Fluid-Flow Machinery, P. A. Sci., in Gdańsk, which are shown in figure 38 and tables II and III.

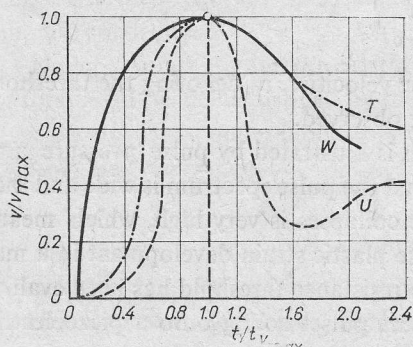


Fig. 38. Change of the relative damage rate for specimens of pure iron (technical grade) vs the conventional time

W – vibratory facility, T – rotating-disk facility, U – jet impingement facility [23]

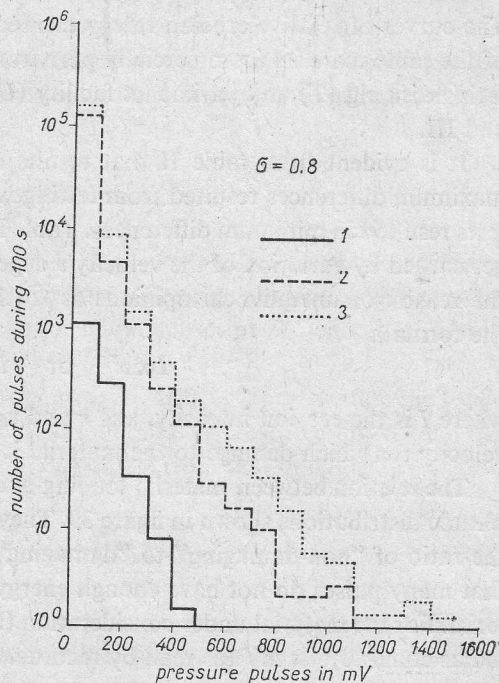


Fig. 39. Histogram of pressure pulses acting upon a piezoelectric transducer or a specimen in different parts of a cavitation cloud [24]

1 – no erosion, 2 – moderate erosion, 3 – strong erosion.

Table II

Cavitation resistance of various metals as related to commercially pure iron, determined by various investigation methods [5]

Material	Relative resistance of the material (as compared to that of commercial grade iron) evaluated using different methods of investigation		
	magnetostriction facility	rotating-disk rig	jet impact facility
Iron E04	1.0	1.0	1.0
Steel St3	1.1	1.4	1.0
Steel 45	3.4	2.3	1.1
Bronze BM 12733	8.5	4.1	1.6

Table III

The effect of the peripheral speed of test pieces in the rotating disk facility on the cavitation resistance of a material [5]

Circumferential velocity V [m/s]	Relative circumferential velocity	Relative material resistance δ_{cav}		
		Cu	Al	Zn
42.1	1.0	1	1	1
32.8	0.78	108	72	13
23.4	0.56	no loss observed after 40 h		30

The curves of relative erosion rate presented in figure 38 show how different is behaviour of the same material (commercially pure iron) when tested on a vibratory facility (W), rotating-disk rig (T) and jet-impact facility (U). Quantitative results are shown in tables II and III.

It is evident from table II that results of material evaluation vary in different way; maximum differences resulted from testing with the vibratory facility (W) while jet-impact tests resulted in minimum differences. Table III shows in turn that material durability changes caused by variation of the velocity v depend on the kind of material. Supposedly this is the reason of unequivocal opinions as to the value of the exponent $n(n^*)$ appearing in the formula

$$I \propto v^n \quad \text{or} \quad I_* \propto (v - v_0)^{n^*},$$

where I is the erosion intensity, and v and v_0 are flow velocities, v_0 denoting the threshold velocity for which damage to the material is still not observed.

The relation between material loading and erosion is illustrated by pulse pressure amplitude distributions shown in figure 39. They show that the pulse spectrum is wide and that the ratio of "non-damaging" to "damaging" bubble collapses is very high, which means that many pulses do not have enough energy to cause plastic strain development in a material. For the material under consideration (lead) the resistance threshold has been evaluated as equal to 600 mV (as read by recorder of pressure pulses acting onto a piezoelectric transducer).

As it was already said differences in evaluation of the same material are caused by imperfectness of cavitation modeling and omission of the scale effect influence on the material damaging. One is thus concerned with the load spectrum which should be known in order to evaluate and predict the resistance of a material to cavitation erosion. For instance, the

following cavitation intensity measures may be adopted: a spectral power density function estimator

$$\hat{G}_*(f) = \frac{1}{B_0 T} \int_0^T x^2(t, f, B_0) dt,$$

or an energy measure

$$ME = \frac{1}{T} \sum n_i p_i^2,$$

where T denotes the averaging time, $x(t, f, B_0)$ is the input signal part filtered out by a band-pass filter having the bandwidth B_0 and the centre frequency f , and n_i is the number of pulses of p_i pressure. Results of investigations presented in figure 40 illustrate the close dependence of material damage on the values of ME and $\hat{G}_*$. Figure 40a is a photograph of an eroded specimen on which locations of a pressure transducer receiving pressure pulses from cavitation bubble implosions are shown. Figure 40b is a diagram showing the distribution of ME for $\sigma = \text{const}$. Figure 40c presents $\hat{G}_* = f(\sigma)$ curves for the spots indicated.

Assume that during the first stage of material damaging, i.e. in the initiation period the impingement energy absorbed by the material is constant (no energy dissipation caused by cavitation pits occurs). It is then possible, based on the value of an appropriate energy measure ($\hat{G}_*$ or ME) in the field and laboratory conditions as well as results of tests (volume loss — ΔV_M) to evaluate the development of erosion in natural conditions using the following formula:

$$\Delta V_P = \Delta V_M \frac{R_{\text{cav}, M}}{R_{\text{cav}, P}} \frac{(\eta ME)_P}{(\eta ME)_M},$$

$t = \text{const}$
 $F_P = F_M$

where R_{cav} is the cavitation resistance, η is the efficiency of energy absorption by the material, subscript P denotes field (prototype) conditions, and subscript M denotes laboratory (model) conditions.

Under assumption that the cavitation intensity measure ME is proportional to the square of total head (which is justified by the formula describing ME as presented above) and that the cavitation field under laboratory conditions (F_M) differs from that in natural conditions (F_P) one has for $t = \text{const}$

$$\Delta V_P = \Delta V_M \left(\frac{D_P}{D_M} \right)^2 \left(\frac{H_P}{H_M} \right)^2 \frac{R_{\text{cav}, M}}{R_{\text{cav}, P}}.$$

It should be remembered, of course, that the method of erosion prediction as presented above is approximate and concerns the initial damaging period only.

Worth noting are results of testing of 45 materials, aimed at determining their cavitation resistance, which are shown in table IV arranged according to the resistance quality. The tests were conducted on the rotating-disk rig at the Institute of Fluid-Flow Machinery, P. A. Sci., in Gdańsk.

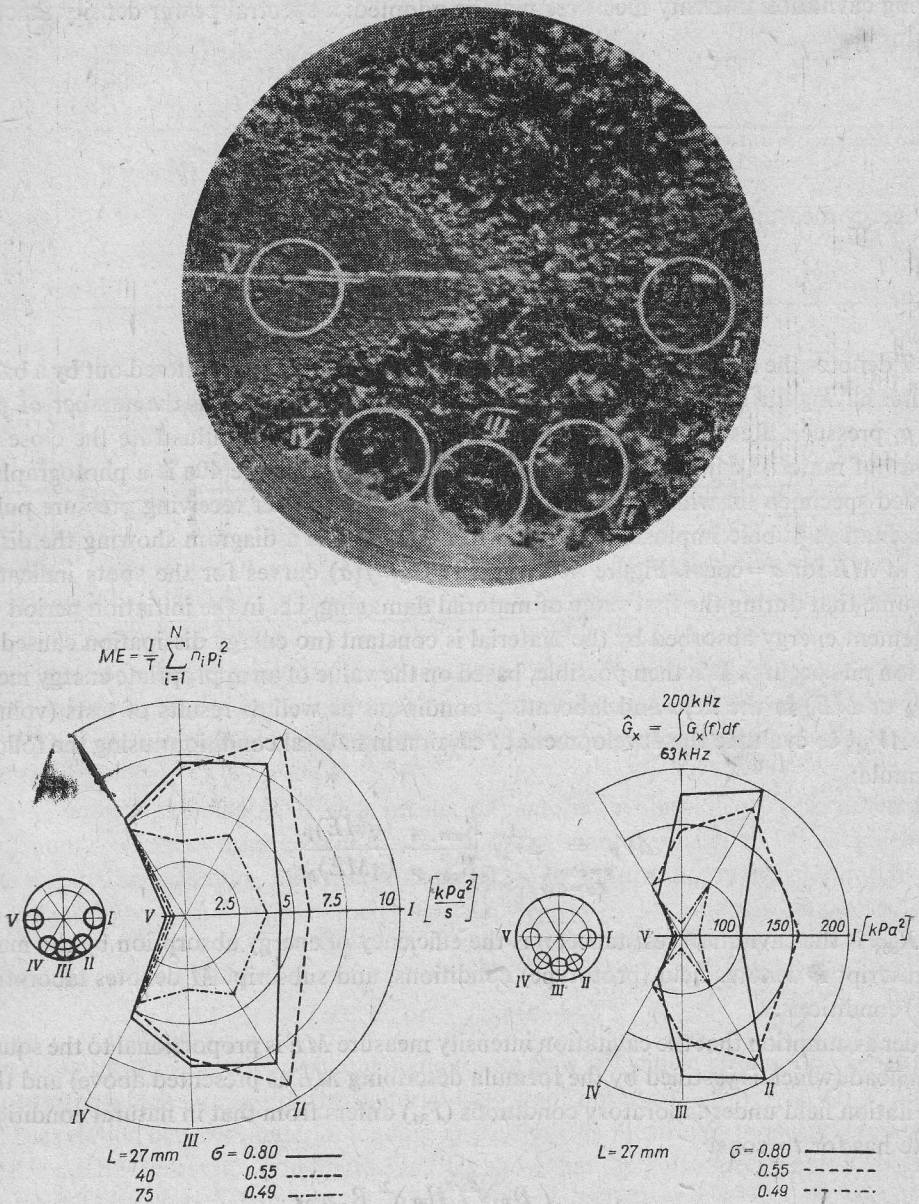


Fig. 40. Relation between the extent of damage to the specimen material and values of ME and $\hat{G}_*$ [15]: a) the surface of eroded specimen and the pressure transducer locations, b) the cavitation intensity distribution for different cavitation conditions defined by ME , c) the cavitation intensity distribution for different cavitation conditions defined by $\hat{G}_*$.

The development of cavitation erosion in a machine may also be estimated based on changes of the cavitation number σ . This is illustrated in figure 41 showing the dependence between the material damage rate and the development stage of cavitation. Such a relation

Table IV

Cavitation resistance classification of 45 structural materials

Resistance	Material (feature)
Very good	G 20, Stellite, S 30
Good	23H12MNF, EStel C ₀ W, ES13Cr, 2 HB (after heat treatment HB 400), 45 (after heat treatment HB 300)
Fairly good	ES-18-8-2, LH17M2, ES-18-8, ES18-8-6, BM 12733, Superston, 2H13 (after heat treatment HB 230), ZlCr28, ES 24-18, S10, Nikalium, 1H18N9T, 45, OH18N9
Moderate	BAM2, 2H13, St5, BK331, L35II, L60I, 1H13, L1H13N, E04, St3, 15G2NbA, L50I, NM55, L25II, L47II
Poor	15GA, M70 (annealed), Zl30, Zl25, Zl20
Very poor	M1, PA2N

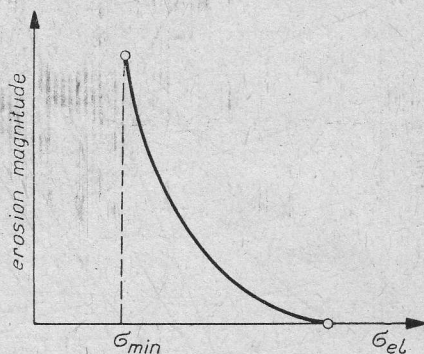


Fig. 41. The course of cavitation damage versus σ_{el} at constant power of the machine [10]

cannot, however, be utilized for volume loss prediction concerning a particular machine. In spite of that it makes possible prediction of changes in the extent of cavitation damage caused by changes of the cavitation number σ .

6. Selection of operating parameters for hydraulic machinery with the cavitation hazard taken into account

The cavitation hazard increases with increasing unit powers of hydraulic machinery. This is caused mainly by increasing liquid flow velocities.

Often one can avoid cavitation by ensuring appropriate value of pressure on the suction side of the hydraulic machine rotor. This means having the cavitation number σ appropriately large. Many a time this is not a useful solution due to high costs of deep foundation of the machine, sometimes several tens metres below the tail water level. For this reason cavitation is usually allowed and therefore periodical machine overhauls become necessary, while investment costs are thus kept on the lower level.

Choice of appropriate working parameters of machinery is another method of cavitation suppression. It is known that the cavitation intensity and its effects on the machine depend on flow conditions. The best range of working parameters may be defined based on the set of iso-efficiency curves for the particular machine, taking into account, among others, also cavitation and dynamic parameters. Figure 42 shows such a "complex" characteristics made for a diagonal pump. The data were taken at constant suction head and different rotational speeds. The following curves are presented: head/discharge curve $H(Q)$,

cavitation cloud length, cavitation number, efficiency and pump casing vibration acceleration. It is evident from the figure that for the pump under consideration the less favourable conditions correspond to the region of inflexion of curves $H(Q)$ that is to the zone of cavitation caused by recirculation. On the other hand, the most favourable operating conditions exist for discharges slightly above the optimum value (that is for $Q \gtrsim Q_{\eta=\max}$).

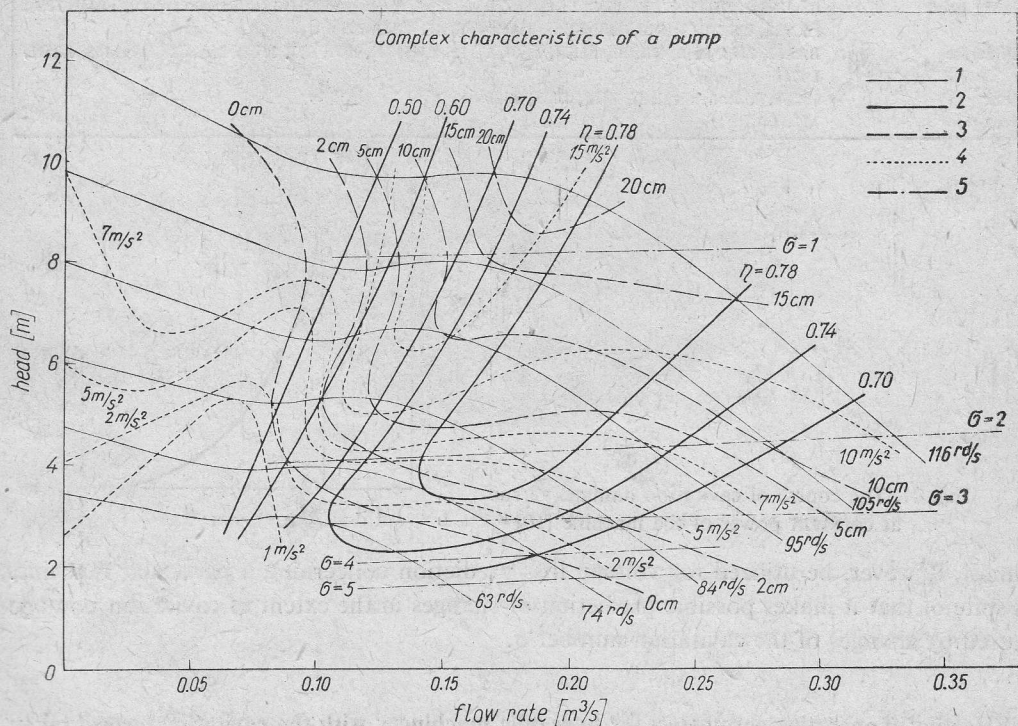


Fig. 42. A "complex" set of curves for a diagonal pump, including the curves of head/discharge $H(Q)$ 1 - throttling, 2 - efficiency (η), 3 - cavitation cloud length, 4 - vibrations of the casing (accelerations), 5 - cavitation (σ)

Usefulness of the above described method for determining machinery working parameters is limited to in situ investigations. Its application to model investigations does not provide reliable information on the prototype machine behaviour as there is no close correlation between symptoms of cavitation in a model and in field conditions. This makes correct prediction of working parameters of a machine that are most favourable from the viewpoint of cavitation difficult, leading to difficulties in selection of parameters optimum with respect to different cavitation aspects.

The manner of running the machine during transient stages of operation should also be taken into consideration when selecting the proper operating conditions of a hydraulic machine. Often a hysteresis of dynamic machine properties is observed, associated partially with cavitation hysteresis. This is illustrated in figure 43 showing vibrations and pulsations of pressure in a Kaplan turbine as functions of the turbine set output changes. In conclusion it should be noted that an appropriate procedure of turbine set start-up, established basing on the experience, may lead to significant decrease of dynamic loading of the machine.

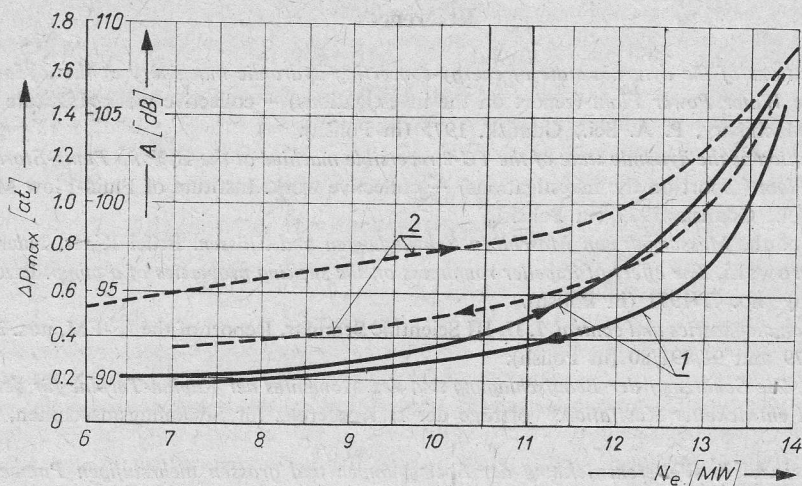


Fig. 43. Variation of vibrations and pressure pulsations in a Kaplan turbine as dependent on the manner of turbine set power changing [19]

1 — vibrations of the casing (A), 2 — pressure pulsations (Δp_{max})

7. Trends in cavitation diagnostics of hydraulic machinery

Present trends of development of hydraulic machinery cavitation diagnostics indicate that the future work in this domain will proceed in two directions. Supposedly these will be seeking for reliable quantitative methods of evaluating the state of a machine with respect to cavitation and investigation of the importance of the scale effect for cavitation diagnostics.

Our opinion presented in this work is that cavitation, being a multiaspect phenomenon, cannot be described by one parameter only. Such an assumption leads in consequence to developing diagnostic methods covering all principal features of cavitation influence on hydraulic machinery instead of trying to find one versatile method.

Most needed, however most difficult to develop, is a method for evaluating the dynamic intensity of cavitation. The method would be aimed at the quantitative evaluation of the erosion hazard. Thus it would be possible to compare erosive cavitation intensities in different fluid-flow arrangements, that is to relate results of laboratory tests of structural materials cavitation resistance to conditions actually existing in the industrial hydraulic machinery.

Future works concerning cavitation detection should also include localization of the phenomenon and determination of its form.

The future tasks of the diagnostics regarding the scale effect will be concerned with relating the intensity of cavitation observed in models to that occurring in prototypes. Thus prediction of cavitation and dynamic characteristics of machines is the goal. Up to now such prediction is based mainly on approximate evaluations.

References

- [1] *Investigations of the dynamic state of energy-converting hydraulic machinery at the Żydowo Pump-Storage Water Power Plant* (report on the investigations) – collective work. Institute of Fluid-Flow Machinery, P. A. Sci., Gdańsk, 1975 (in Polish).
- [2] *Investigation of the dynamic state of the TG-2 reversible machine at the Żydowo Pump-Storage Water Power Plant* (report on the investigations) – collective work. Institute of Fluid-Flow Machinery, P. A. Sci., Gdańsk, 1977 (in Polish).
- [3] J. T. Broch, *Messungen von Maschinen Schwingungen und Stößen*. Brüel Kjaer, Naerum 1970.
- [4] S. Dąbrowski, *The effect of impeller roughness on the sucking properties of a pump*. Przegląd Mechaniczny, no. 7, 1971 (in Polish).
- [5] *Cavitation diagnostics and control*. I, II, III Scientific Seminar, Report of the IF-FM, nos. 31/920/78, 59/973/79 and 91/991/80 (in Polish).
- [6] Á. Fay, *Die Schätzung der Beaufschlagung und des Momentes der Kaplan-Turbine für den Betriebszustand entwickelter Kavitation*. Vorträge der II Konferenz für Strömungsmaschinen, Budapest 1966.
- [7] D. Floriančič, *Weiterentwicklung der Speisepumpen und grossen mehrstufigen Pumpen für die Wasserversorgung*. Technische Rundschau Sulzer, Bd. 55, H. 4, 1973.
- [8] F. G. Hammitt, *Cavitation erosion. The state of the art and predicting capability*. Applied Mechanics Reviews, vol. 32, no. 6, June 1979.
- [9] International Electrotechnical Commission, Publication 497; International Code for Model Acceptance Tests of Storage Pumps, Geneva 1976.
- [10] International Electrotechnical Commission, Publication 609: Cavitation pitting evaluation in hydraulic turbines, storage pumps and pump-turbines, Geneva 1978.
- [11] J. Kirejczyk, *Diagnostics of cavitation in diagonal pumps*. Doctor's Thesis, Institute of Fluid-Flow Machinery, P. A. Sci., Gdańsk 1980 (in Polish).
- [12] R. T. Knapp, *Recent investigations of cavitation and cavitation damage*. Trans. ASME, vol. 77, 1955.
- [13] A. S. Lashkov, *Energeticheskii metod opredeleniya kriticheskogo kavitatsionnogo rezhima gidroturbiny*. Energomashinostroenie, no. 2, 1976.
- [14] F. Numachi, *Effect of static pressure differences on the cavitation characteristics of hydrofoil profiles*. Cavitation in Hydrodynamics, National Physical Laboratory, London 1956.
- [15] E. Partyka, *Cavitation intensity indicators*. III Seminar on Cavitation Diagnostics and Control, Reports of the IF-FM, no. 91/991/80, Gdańsk, 1980 (in Polish).
- [16] Z. Reyman, K. Steller, *Evaluation of the materials resistance to cavitation in flow*. Trans. IF-FM, no. 76, 1978 (in Polish).
- [17] K. Steller, *Investigations of the scale effect on the cavitating flow*. Trans. IF-FM, no. 50, 1970 (in Polish).
- [18] K. Steller, *Symptoms of cavitation in the hydraulic machinery*. Trans. IF-FM no. 59, 1972 (in Polish).
- [19] K. Steller, *Correlation between different symptoms of cavitation observed during the tests of a Kaplan turbine and a rotodynamic pump*. Proc. of the 4th Conf. on Fluid Machinery, Akadémiai Kiadó, Budapest 1972.
- [20] K. Steller, *Scientific activity of the Department of Fluid Dynamics at the Institute of Fluid-Flow Machinery, P. A. Sci., during the years 1970 - 1972*. Warszawa-Poznań 1974 (in Polish).
- [21] K. Steller, *Cavitation damage to hydraulic machinery and associated warranties*. Reports of the IF-FM, no. 23/911/77, Gdańsk 1977 (in Polish).
- [22] K. Steller, *Scientific activity of the Department of Fluid Dynamics at the Institute of Fluid-Flow Machinery, P. A. Sci., during the years 1973 - 1975*. Warszawa-Poznań 1978 (in Polish).
- [23] K. Steller, *Change of the relative material resistance due to the change of cavitation conditions*. Proc. of the 6th Conf. of Fluid Machinery, Akadémiai Kiadó, Budapest 1979.
- [24] K. Steller, *Remarks on cavitation and cavitation erosion prediction*. III Seminar on the Cavitation Diagnostics and Control, Reports of the IF-FM, no. 91/991/80, Gdańsk 1980 (in Polish).

- [25] K. Steller, J. Kirejczyk, *Evaluation of cavitation investigations results for a model reversible machine of Deriaz type (for the Czorsztyn water power plant)*. External report of the IF-FM, P. A. Sci., Gdańsk 1980 (in Polish).
- [26] A. J. Stepanoff, *Pumps and blowers*. Wiley 1965.
- [27] P. Ulith, *Kavitationsgrenzen bei Francisturbinen*. Voith Information, 3, 1981.

Diagnostyka kawitacji w maszynach hydraulicznych

Streszczenie

Praca należy do kategorii opracowań przeglądowych i zawiera informacje o obecnym stanie wiedzy w dziedzinie diagnostyki kawitacji w maszynach hydraulicznych.

We wstępie przedstawiono uwagi ogólne na temat kawitacji w maszynach hydraulicznych oraz wyjaśniono cele i metody diagnostyki kawitacji. Z kolei przedstawiono kilka przykładów własnych wyników badań kawitacyjnych oraz omówiono związki między różnymi oznakami kawitacji w turbinach wodnych i pompach. Osobny rozdział poświęcono sprawie przewidywania kawitacji i erozji kawitacyjnej. Wypowiedziano się również na temat wyboru parametrów pracy maszyn hydraulicznych ze względu na kawitację. Na zakończenie wskazano na przyszłościowe cele diagnostyki kawitacji w maszynach hydraulicznych.

Диагностика кавитации в гидравлических машинах

Резюме

Статья принадлежит к категории просмотровых отчетов и содержит информации о состоянии знания в области диагностики кавитации в гидравлических машинах.

Во вступлении представлены общие примечания на тему кавитации в гидравлических машинах и объясняются цели и методы диагностики кавитации. Затем представлено несколько примеров собственных результатов кавитационных исследований и обсуждаются связи между различными признаками кавитации в гидротурбинах и насосах. Отдельный раздел посвящен делу предвидения кавитации и кавитационной эрозии. Высказаны также мнения на тему выбора рабочих параметров гидравлических машин с учета на кавитацию. В заключении указывается на перспективные цели диагностики кавитации в гидравлических машинах.