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PUMP VIBRATION CAUSED BY CAVITATION

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The problem of pump vibration caused by various stages of cavitation is examined. The influence of the air content of liquid on the intensity and frequency of vibration is established.

The relation of the collapsing cavitation to a heavy increase of vibration in the average audio-frequency range is indicated. Measures for the prevention of vibration over the entire audio-frequency range are described.

One of the most important problems of hydraulic-machine design is the prevention of vibration. In the literature primary attention is given to vibration caused by unbalanced rotary masses. This vibration has a maximum amplitude at the rotation frequency $f_1 = \frac{n}{60}$ cps (where n is the rpm) and can be significantly reduced by more careful installation and production of the rotary parts of the machinery. However, experience with various hydraulic machines (hydro turbine, centrifugal, fan and screw pumps) indicates that in several working regimes additional-frequency vibration develops, the amplitude of which greatly exceeds the vibration amplitude at the rotation frequency [1]. Such vibration not only causes fatigue effects in various parts and units of the machinery, but sometimes leads to serious accidents. The cause of these intensive vibrations is assumed to be cavitation. Intensive studies [1], [2], and [3] have been devoted to the fight against cavitation and its destructive activity, but none of them has paid sufficient attention to the relation of the development of vibration to the degree of cavitation and the properties of the fluids pumped. The lack of such information prevents the development of effective measures for the elimination, or even the partial reduction of this vibration. The author investigated a series of centrifugal pumps. The qualitative picture of the occurrence of vibration from cavitation (within certain limitations) can be extended to all hydraulic machinery.

Centrifugal pumps of various design, sizes and speeds were investigated. Special testing equipment allowed us to conduct complex tests of pumps with open and closed cycles. An air spray device fitted to the intake pipe made it possible to alter the proportions of the dissolved and diffused air in the water. Cavitation was measured by constructing cavitation characteristics and completed by visual observation and from marks left by cavities on the vanes of the impellers. The vibration was measured on the spiral cam and lug of the pump by vibration pickups possessing a linear characteristic which extends over the entire audio-frequency range. Vibrations up to 8000 cps were analyzed by an acoustic device with a three per cent bandwidth pass. The general level of vibration was measured in two broad frequency bands: 25 to 1000 cps and 25 to 20,000 cps. Together with spectral analysis and visual observations, this permits us to determine the influence of cavitation on the vibration spectrum and to develop means for combatting this vibration.

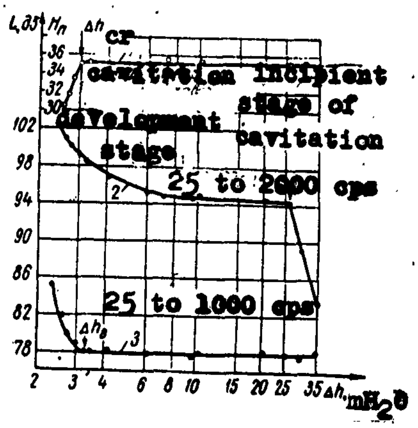


Fig. 1. The general level of vibration pump No. 1 (RK-2) n = 2700 rpm vs. the magnitude of the cavitational reserve during closed cycle tests.

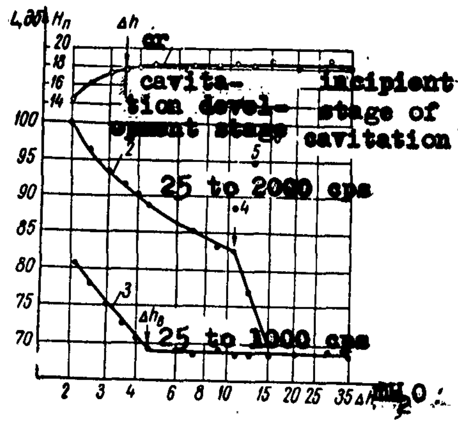


Fig. 2. The general level of vibration pump No. 2 (RK-3) n = 1400 rpm vs. the magnitude of cavitational reserve during closed cycle tests.

In Figs. 1 and 2 the relationship of the general level of vibration to the magnitude of cavitational reserve (Δh) for two centrifugal pumps which differ in weight and number of rotations is plotted. Analysis of these curves together with visual observations indicates that there are two phases of cavitation: air and steam cavitation.

Air cavitation is connected with the separation of air in eddies and eddy zones which are formed in the flow channels as a result of the flow break away from the surface of the vanes, and the presence of surface cracks as the flow descends from the vanes and the presence of an axial eddy in the inter-blade channels, etc. During the operation of the pump the extent of eddy formation can be so great that even at excessive suction pressure the pressure in the eddies can fall below the air saturation pressure of water. As a result of this air bubbles which move with the flow will form in the center of the eddies. Under the influence of changing internal and external forces they radiate very strong vibrations over a wide frequency band, which cause the vibration of the casing of the pump.

In Fig. 1 this intensive development of vibration in 25 to 20,000 cps frequency range begins at $\Delta h > 35$ m of a water column. This corresponds to an excess intake pressure of more than 2.5 kg/cm².

In Fig. 2 the onset of vibration begins at $\Delta h \gtrsim 17$ m of water. It must be noted that on the leading edges of the vanes where cavitation is usually thought to occur, there are no visible cavitation bubbles and cavities.

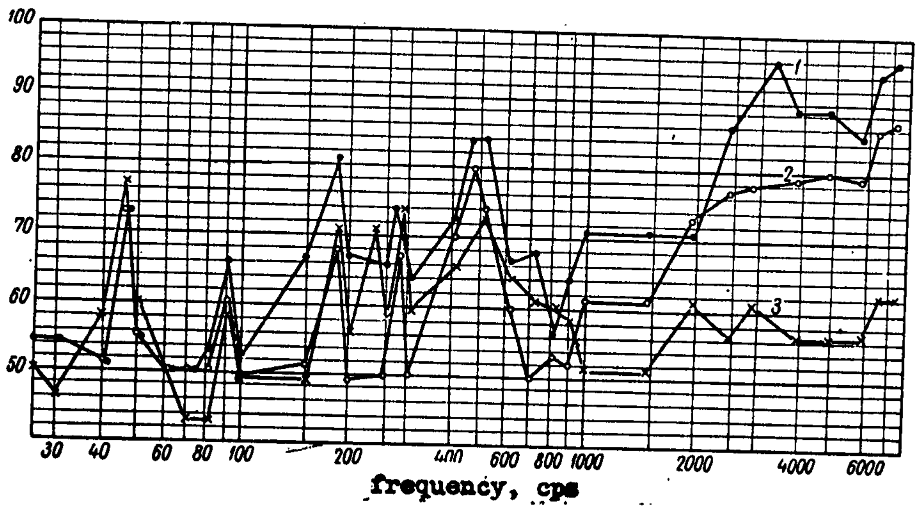


Fig. 3. Vibration spectrum of pump No. 1, n = 2700 rpm at various cavitation reserves: 1) Δh = 2.35 m H_2O; 2) Δh = = 10 to 25 m H_2O; 3) Δh = 36 m H_2O.

In Figs. 3 and 4 the vibration spectra measured on the lugs of experimental pumps are plotted. Curves 3 and 2 approximately correspond to the points of inflection of curves 2 in Figs. 1 and 2, that is, to the beginning and the end of the intensive increase of the level of vibrations caused by air cavitations in the 25 to 20,000 cps range. It is apparent from these graphs that air cavitation greatly increases the general level, by 10-15 db, and the high-frequency components of vibration by 10-20 db. Vibrations in the intermediate audio-frequency range which are a danger to the machine remain at a constant general level (curve 3, Figs. 1 and 2) and at approximately the same intensity of separate spectral components (curves 2 and 3, Figs. 1 and 2). The results of these tests allow us to draw two more interesting conclusions:

1. The different values of Δh at which the intensive vibration increase begins is determined by the intensity of the eddy formation which, in its turn, mainly depends on the number of rpm. Actually pump No. 1 (see Fig. 1) and pump No. 2 (see Fig. 2) have approximately the same average flow-channel speeds and cross-sectional speed-distribution curves, but their rpm speeds are different: pump No. 1 n = 2700 rpm, pump No. 2 n = 1400 rpm. As a result of this difference the intensive vibration increase begins in pump No. 1 at $\Delta h > 35$ m H_2O, whereas in pump No. 2 it begins at $\Delta h \approx 17$ m H_2O.

2. The low intensity of the spectral vibration components in the frequency range above 1000 cps in pump No. 2 at $\Delta h > 20$ m H_2O* (curve 3, Fig. 4) and in pump No. 1 $\Delta h > 35$ m H_2O* (curve 3, Fig. 3) indicates that the vibrations actually caused by eddies, turbulent flow pulsations, friction and other hydrodynamic forces are of no importance as independent sources of vibration in pumps and other hydraulic machines. Eddies and turbulent pulsations can help increase the vibrations only if air bubbles are formed in them, or in the fluid. As they move through the channels the bubbles fall into the eddies under the influence of high local pressure gradients, they begin to vibrate intensively causing the pump casing to vibrate. This was checked by injecting tiny air bubbles into the inlet

* A further increase of Δh does not usually cause changes in the vibration spectrum.

(suction) pipe of pump No. 2. Then the water was degassed to eliminate the influence of the dissolved and, consequently, liberated air in the eddies. At $\Delta h > 10$ (Fig. 2, experimental points 4, 5, 6) when there is no visible cavitation on the leading edges of the vanes, the injection of a small quantity of air causes a sharp increase (by 5-30 db) of high frequency vibrations.

Generally, no damage was found in flow channels with air cavitation. Vapor cavitation is connected with the formation of air-steam bubbles and cavities in the immediate neighborhood of the impeller's leading edges which takes place as a result of a decrease of local pressure to the value of the vaporization pressure of the liquid at the given temperature. Visual observations indicate that the three stages of vapor cavitation in the pumps can be clearly distinguished: the incipient stage, the developed stage and the collapsing stage.

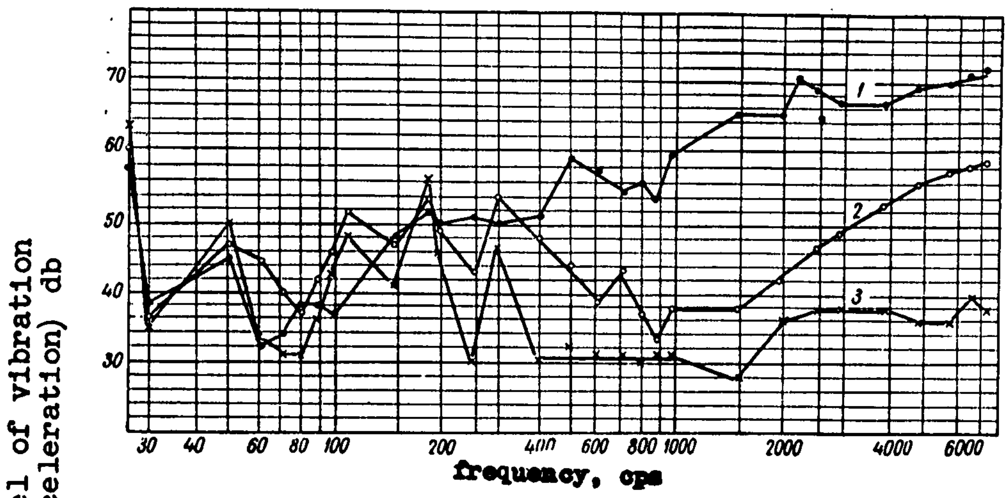


Fig. 4. Vibration spectrum of pump No. 2 n = 1400 rpm at different cavitation reserves: 1) $\Delta h = 23$ m H_2O; 2) $\Delta h = 10$ m H_2O; 3) $\Delta h = 20\text{-}30$ m H_2O.

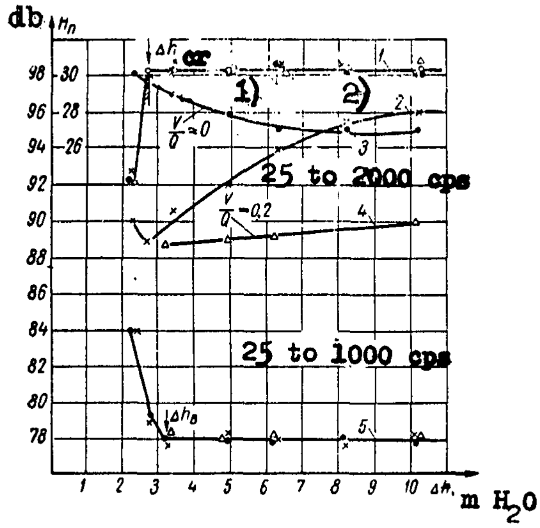
The incipient (visible) vapor cavitation stage depends mainly on the shape of the leading edges of the vanes and the angle of attack. This stage of cavitation considerably precedes the stages which bring about the alteration of the cavitational curves 1, (see

Figs. 1 and 2). The cavitation area in this instance consists of tiny bubbles composed primarily of air and steam, which are combined into a small cloudy band at the leading edges of the vanes. It is evident from Figs. 1 and 2 that the initial cavitation stage does not cause a sharp increase of the general level of vibration in the audio-frequency range. The spectral components measured before and after the initial cavitation stage also remain unchanged. In this stage the cavitations radiate ultrasonic vibrations, which can not be measured by the apparatus used in these investigations.

The developed steam cavitation stage begins with the appearance of visible cavitation and is assumed to continue until the beginning of the cavitation curve change (see Figs. 1 and 2). This stage is characterized by the steady expansion of the cavitation zone over the entering edges. The general level of vibration in the 25-20,000 cps frequency range also steadily increases without undergoing sharp change. The bubbles which are composed mainly of air and steam in the beginning of this stage, gradually turn into pure steam and the border of the cavitation area becomes more clearly defined. This is evidence of the instantaneous collapsing of the small bubbles [4]. During the formation of the pure steam cavitation the amplitude of vibration depends to a great extent on the degree to which the water has been saturated with air. The different content of air in water is the result of the different relationship of the vapor and air phases inside the bubbles themselves. This accounts for the different contraction rates of these bubbles. The less saturated the water is with air, the greater the closing rate of the bubbles and the more intense the vibration of the pump. Investigations indicate that the general vibration level of a pump operating with air saturated water is always 3 to 10 db lower than a pump using deaerated water. Visual observations of cavitation on the impeller vanes carried out with different ratios of air to water prove that the development of cavitation, that is, the area covered by cavitation with the same value of cavitation reserve Δh, remains constant and does not depend on the air content of water. It is only the structure of the cavitational region itself which changes. When the water is saturated with air the cavitation area is bright white and its edges are washed out, while further along the stream, separate streamlets with non-collapsed bubbles can be seen. If the water is deaerated the cavitation area is transparent and has a sharply defined border as a result of which the bubbles seem to be collapsing instantaneously. It follows that the intensity of the vibration depends not only on the dimensions of the cavitation zone, but also on the relationship of the air and vapor phases inside the cavitation bubbles themselves.

If the pump pumps air saturated water and has a long inlet (suction) pipe, air begins to be liberated when the suction pressure is lower than the atmospheric pressure (that is at $\Delta h > 10$ m H_2O). At constant high suction pressure this process is stabilized and the pump actually pumps a two-phase air-water mixture. This phenomenon can be observed when the sluice valves, cleansing jets, nonreciprocating valves, etc., are adjusted for suction. The presence of dispersed air bubbles in the stream of liquid, as well as the dissolved air in the water leads to a significant reduction of the high frequency vibrations of the pump.

The results of these tests are presented in Fig. 5. Curve 1 represents the averaged curve. Curve 3 represents the general level of vibrations during tests of pump No. 1 with deaerated water in a closed system at a different value of Δh created by a reduction of the system's reserve pressure.



1) developed stage of cavitation

2) incipient stage of cavitation

Fig. 5. Change of the general level of vibration of pump No. 1 (RK-2) n = 2500 rpm with different proportions of air and different means for creating rarefaction.

Curve 2 represents the general level of vibration during tests of the pump with saturated water when the flow is throttled by a valve placed in the inlet (suction) pipe. Curve 4 represents the general level of vibration when air is artificially introduced into deaerated water. It is apparent from this graph that cavitation curve No. 1 is not related to the content of air in water or the method used for creating rarefaction in the inlet. With the throttling of the suction flow vibrations change according to a law (curve 2) which is completely different from the law (curve 3) governing vibration in a closed system. This reduction is related to the appearance in a stream of a large number of air bubbles liberated in the sluice valve. By favoring the high frequency vibrations the air bubbles can somewhat lower the pump's suction capacity. The greater this deterioration is, the lower the suction rate in the intake pipe and the greater the distance between the pump and the sluice valve. In one of the pumps the liberation of bubbles in the sluice valve led to the increase of the critical cavitation reserve (Δh_{cr}) by 0.5 to 0.8 m H_2O. Change in the structure of the flow entering the pump can also be produced by artificial air injection. The results of these tests are given in Fig. 5 (curve 4) and Fig. 6 (curves 3, 4, and 5). These results indicate that artificial air injection is a very effective method of combatting high frequency vibrations produced by steam cavitation. The vibrations were reduced by 5 to 15 db, and even more, in several instances up to 20 to 30 db. At the same time it is necessary to determine the maximum air output V which can be approximately expressed in the formula $V/Q \leq 0.002$ m^3/hr (where Q is the output of the pump m^3/hr). An increase of the ratio of air to water produces an earlier disruption in the cavitation curves and does not help to reduce vibration. Thus the injection of air into the main line completely eliminates the vibrations produced by steam cavitation without impairing the cavitational properties of the pumps.

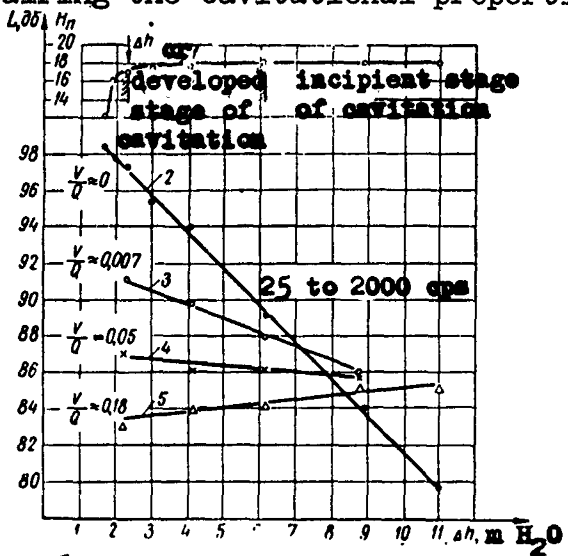


Fig. 6. Change of the general level of vibration of pump No. 2 (RK-4) n = 1400 rpm with different proportions of air to fluid.

At this cavitation stage damage to the flow channels at some distance from the cavitation zone can be observed. This damage decreases with the increase of the ratio of air to water. The developed cavitation stage, like the incipient stage preceding it, presents no threat to the machine as a whole but it causes intensive damage to the flow channels and promotes pump vibration, which in a number of cases it is desirable to limit.

The disruption stage of cavitation is accompanied by a complete breakdown of pump operations. The cavitation cavities cover a larger part of the impeller vanes and periodically break away from the vanes so that the movement of the fluid in the pump assumes a pulsating character. The entire equipment is subjected to heavy shocks. The head, the output and the efficiency of the pump fall. The presence of such a cavitation stage is indicated by a sharp drop of the cavitation curves 1 (see Figs. 1, 2, and 5). The analysis of curves 3 in Figs. 1 and 2 and of curve 5 in Fig. 5 indicates that the general vibration level in the 25-1000 cps frequency range, within broad limits, does not depend on the cavitation reserve Δh, the proportions of fluid and the method of producing rarefaction in the suction stage. For a given design of pump and rpm it depends mainly on the amplitude of the mechanical sources of vibration. In this frequency range cavitation is a source of vibration only under disruptive operating conditions or those close to them.

Investigations indicated that a marked increase of vibrations in the 25 to 1000 cps frequency range is definitely related to the disruption cavitation stage Δh as determined by the cavitation curve 1 in Figs. 1, 2 and 5. The intensive vibration increase always begins sooner than cavitation curves 1 commence to drop.

TABLE

Coefficients of Reserve φ

	pump No. 1			pump No. 2		
	rpm	RK-1	RK-2	rpm	RK-3	RK-4
$0.8Q_{specific}$	2500	1.26	1.15	1200	1.32	1.22
$Q_{specific}$		1.20	1.15		1.19	1.44
$1.2Q_{specific}$		1.32	1.10		1.38	1.58
$0.8Q_{specific}$		1.27	1.17		1.27	1.39
$Q_{specific}$	2700	1.25	1.10	1400	1.28	1.36
$1.2Q_{specific}$		1.21	1.17		1.23	1.35

In the table the relationship of the critical cavitation reserve Δh_{cr} to the cavitation reserve Δh_{v} corresponding to the onset of the increase of vibrations in the 25 to 1000 cps frequency range is presented. It is evident from the table that the maximum permissible cavitation reserve which guarantees the pump production of the required head characteristics, output, efficiency and the elimination of vibrations caused by disruption cavitation must be selected with a 30-40% reserve of the tabulated h_{cr} or that determined by the formula at the given high-speed cavitation coefficient C:

$$C = 5.62 \frac{n\sqrt{Q}}{\Delta h_{cr}^{3/4}}$$

where n is the rpm

Q is the output of the pump

Δh_{cr} [m H_2O] is the critical cavitation reserve

Vibrations causing the disruption stage of cavitation are a great danger not only to the mechanism as a whole* and to its separate parts by causing damage to the bearings, abuttments and fixing bolts [1], but also to structures and substructures at a considerable distance from the equipment.

There are two reliable methods for combatting such vibrations: first, by proper adjustment of the operating conditions to the suction height and second, the improvement of the caviation properties of the pumps if the suction height is given and it is not possible to change it. Such an improvement of pump cavitation properties can be brought about mainly by increasing the width of the impeller inlet by using vanes of double curvature and sickle-shaped profile.

Concerted application of the above recommendations significantly improves the cavitation properties of the pumps and thus broadens the range of the working suction heights.

This picture of the development of vibration caused by cavitation can be observed in all hydraulic machines. However, each one will have its peculiarities. For example, in hydroturbine and propeller pumps cavitation begins at first in the terminal eddy which breaks away from the vanes. In the beginning air cavitation occurs and it turns into steam cavitation as the load on the vanes (increase of rpm, delivery, etc.) increases. The developed stage of cavitations in this instance is related to the spread of cavitation over the vanes, and at the moment of disruption one can also observe the cavities periodically breaking away from the vanes. As a result of this the whole experiment is subjected to heavy shocks.

* The vibration amplitude can reach 0.5-1.0 mm and more.

Conclusions

1. Air and steam cavitation are a source of intensive vibration in centrifugal pumps.

2. Air cavitation is related to liberation of the air in the eddies which break off in the flow channels of the pump and are a source of high-frequency vibration.

3. Air cavitation consists of three stages: incipient, developed and disruptive.

4. The incipient steam stage of cavitation does not disturb the operation of the pump and is a source of ultrasonic vibrations.

5. The developed stage of cavitation is a source of flow channel damage and intensive high-frequency vibrations. The damage as well as the vibration can be completely eliminated by changing the ratio of air to water.

6. The disruptive stage of cavitation is a source of intensive pump vibration of pumps in the average audio-frequency range.

The amplitude of these vibrations can reach a high value, threatening the operation of the machine. A method of combatting these vibrations is the correct selection of suction heights and the improvement of the suctional properties of pumps.

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