A STUDY OF CAVITATION IN WATER
AS PRODUCED WITH LOW FREQUENCY SOUND ENERGY

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PREFACE

In recent years, many applications of ultrasonics (sound at frequencies above the audible level) have been developed and utilized in industry. In particular, ultrasonic cleaning has become well known and widely used in the solution of perplexing metal cleaning problems.

Basically, the mechanism of ultrasonic cleaning is achieved whenever ultrasonic energy is propagated into a liquid cleaning agent with sufficient intensity to cause the agent to cavitate. It is the cavitation, not the ultrasonic energy, that provides the cleaning effect. Therefore, identical cavitation, no matter how it is produced, should achieve the same end.

This raises the question of the possibility of a more economical, yet equally effective, method of producing the desired cavitation, particularly for large scale cleaning applications. One possibility would be the use of low frequency sound energy in place of the ultrasonic energy. As a result, this study was undertaken to determine whether or not effective cavitation could be produced with low frequency sound energy and, if so, some of the associated advantages and disadvantages.

Appreciation is expressed to Dr. R. L. Lowery for the suggestion and supervision of this study; to John Leland for his devoted assistance in the design and operation of the experimental equipment; and to the staff of the Mechanical Engineering Laboratory for their technical assistance.

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CHAPTER I

INTRODUCTION

Effective metal cleaning is a very important factor in many industrial processes, some of which may be initial parts manufacture, assembly, periodic cleaning, or maintenance and repair. One particular metal cleaning requirement, which falls into the last category, is generated when jet engines undergo overhaul. During the overhaul, many parts, particularly compressor blades, are thoroughly inspected for minute cracks. Flaw detection techniques currently used for this purpose require that the blades be perfectly clean. Since each engine may contain more than a thousand blades, a large metal cleaning requirement is generated.

Ultrasonic Metal Cleaning

An evaluation of different cleaning methods, conducted by the Non-Destructive Testing Project of the School of Mechanical Engineering at Oklahoma State University, has shown that ultrasonic cleaning is a very effective method for the rapid cleaning of compressor blades in preparation for flaw detection. Probably the biggest drawback to using ultrasonic cleaning for such a large scale cleaning operation is the relatively high cost involved. The bulk of the cost of an ultrasonic unit is associated with the conversion of 60 cycles per second line power to frequencies in the ultrasonic range, a cost which would be reduced materially if the same cleaning effect could be achieved with frequencies in the 60 cycles per second range. In such a system 60-cycle line power could be applied directly to the transducer with no intermediate amplifying device required.

Basically, the ultrasonic cleaning mechanism is achieved when a liquid cleaning agent is subjected to ultrasonic energy of sufficient intensity to cause the cleaning agent to cavitate. "Ultrasonic" is defined as pertaining to sound at frequencies above the audible level. The highest audible frequency level varies among individuals, but may be taken as about 20,000 cycles per second in defining the threshold of the ultrasonic range.

In ultrasonic cleaning, the actual cleaning results from the high pressures which are generated when cavitation takes place in the liquid cleaning agent. The collapsing bubbles are very small and the instantaneous high pressures are restricted to a small locality which permits the cleaning of small and delicate items, such as watch parts, without damage. Generally speaking, the higher the frequency of the ultrasonic energy, the smaller the cavitation bubbles. The largest cavitating bubbles encountered in the ultrasonic range are about 0.3 mm in diameter. (1).[1]

Cavitation Cleaning

Returning to the mechanism of ultrasonic cleaning, it may be seen that "cavitation cleaning" would be a more descriptive name for this

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[1] Numbers in parentheses refer to sources listed in the selected bibliography.

phenomenon. The ultrasonic threshold need not be a cutoff in achieving the same end result although it is apparent that the physical discomfort associated with intense audible sound energy might be an important factor. The use of the term "cavitation cleaning" removes the ultrasonic restriction and permits extension of the phenomenon into the audible frequency range. Accordingly, this term will be used to describe this effect without regard to the frequency of the sound energy source. The term "low frequency sound" will be used to indicate sound within the audible frequency range.

The Mechanism of Cavitation

A look at the mechanism of cavitation is appropriate at this point since it is the effect of this phenomenon that makes cavitation cleaning possible. The British engineer and author, Alan E. Crawford (1), wrote in 1955 that "of all the phenomena associated with the passage of an intense sound wave through a liquid, the production of cavitation is probably the widest known but at the same time the least understood." A review of the more recent literature on this subject indicates that Crawford's statement is still true. Considerable research and study has been directed toward this area, yet many questions remain unanswered, particularly from the quantitative standpoint. There is also disagreement among the authorities on certain qualitative points.

It is generally accepted that liquids ordinarily contain minute nuclei comprising cavities of undissolved gas, even if the liquid has been thoroughly degassed. When sound energy is propagated through the liquid the alternating positive and negative pressures compress and expand these minute cavities. The expanding and contracting cavities coalesce, forming larger cavities which reach a size where they can no longer sustain the negative pressure of the sound wave. When this happens, the cavity collapses with the result recognized as cavitation.

The pressure rises associated with the cavitation are mainly dependent upon the vapor content of the cavity and the maximum negative pressure which the liquid is capable of sustaining.

The latter consideration is a quantitative aspect about which there is little agreement among authorities. The problem here closely parallels that of tensile strength of solids except that the measurement of the tensile strength of solids is relatively simple. (2). Theoretical tensile strengths of water vary from 500 to 10,000 atmospheres, depending on which theoretical approach is used in making the calculations. (3). Measured values of the static breaking point of liquids have been found, like those for solids, to lie well below the theoretical values and to be erratic. One of the highest recorded measured values is 277 atmospheres. (4). The accuracy of any such measurements may well be suspect when the problems of measurement are considered. It should be noted that the theoretical and measured values cited are those for static conditions and that it may be reasonably assumed that lower values would be associated with the dynamic loading of sound energy.

The vapor content of the cavity has a pronounced effect upon the pressure rise resulting from the collapse of a cavity. This may be understood in the light of the cushioning effect of the cavity contents. A complete void would result in no cushioning and, consequently, extremely high pressures. Even though vapor is always present in the bubble, it is reported that pressures of 200 to 500 atmospheres have been measured at a distance of 0.5 cm from a cavitation center through the use of minute probes. (5). The damping effect of the surrounding liquid attenuates the associated shock wave over a short distance so that the high pressures are restricted to the immediate vicinity of the collapsing bubble. Those bubbles which collapse against a surface impart the full impact of the pressure pulse to the surface so that foreign matter adhering to the surface is dislodged. Exposure of the surface to cavitation for a prolonged period results in actual erosion of the surface, a detrimental effect long associated with ships propellors and high power, underwater sound transmission.

The Cavitation Threshold

Probably the most important parameter associated with sonically induced cavitation is the cavitation threshold, the sound energy intensity level required for cavitation. Also of importance is the intensity of cavitation which defines the amount of energy released as each bubble cavitates. It is commonly accepted that the cavitation threshold and intensity of cavitation are related in a complicated fashion. The most desirable relationship is one in which the cavitation threshold is low and the intensity of cavitation is high. Factors affecting the cavitation threshold are more readily defined than those affecting the intensity because the presence of cavitation is easily discernible whereas the intensity is not.

Temperature, static pressure, dissolved gaseous content, undissolved impurities, viscosity and surface tension of the cavitating liquid all affect the cavitation threshold. A high vapor pressure, temperature, or gaseous content results in a lower threshold while a high value of each of the other three parameters has the opposite effect. The effect of varying any of the parameters within a given liquid follows the same pattern.

A commonly accepted general relationship between frequency and the cavitation threshold is shown in Fig. 1.

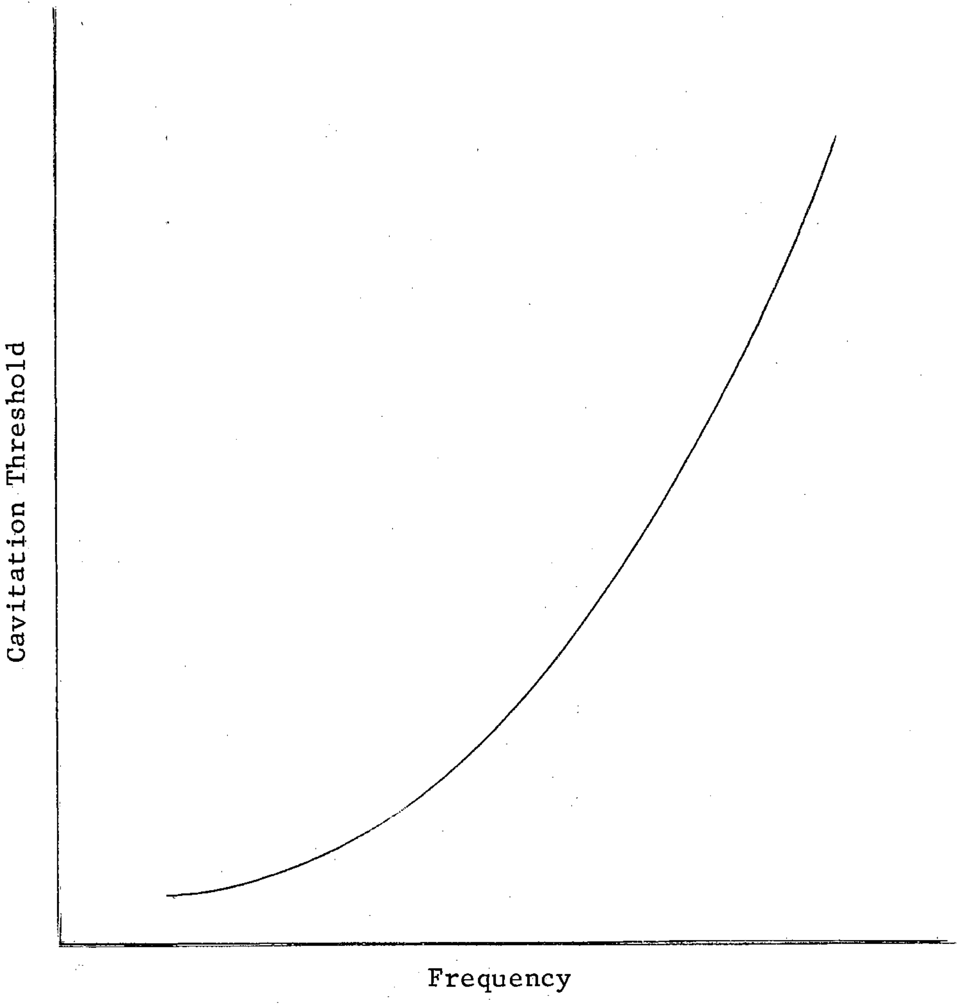

Figure 1. Cavitation Threshold - Frequency Relationship

Experimental results for degassed water show that the curve closely approximates a horizontal line for frequencies below 1000 cycles per second with an intensity of about one watt per square centimeter required to produce cavitation. (5). This is about one-third the intensity required at the ultrasonic threshold. The threshold rises very rapidly with frequency increase at higher frequency values so that with a frequency of one megacycle per second the threshold is about 500 times that at 1000 cycles per second. Assuming that the trend of the curve is valid at low frequencies, the cavitation threshold at 60 cycles per second would be about one third of that at the ultrasonic frequency threshold for water. This implies that cavitation would be sustained with one-third the power dissipation required at the ultrasonic threshold.

The Scope of Study

This study covers the production of sonically induced low frequency cavitation and compares it with cavitation produced by ultrasonic means.

The production of the cavitation is achieved by the propagation of low frequency sound of cavitation threshold intensity into the liquid. A plate or piston vibrating at the desired frequency becomes a sound source, acting as a transducer by changing the driving energy into sound (acoustical) energy. Although the driving of the transducer at the desired frequency is simple, the production of low frequency sound in a liquid medium is a difficult problem which is covered in the next chapter.

The comparison is made from a qualitative standpoint owing to the complexities of accurate quantitative measurements.

CHAPTER II

THEORETICAL ANALYSIS

As related in the introduction, the onset of sonically induced cavitation takes place when sound energy of the intensity of the cavitation threshold is propagated into a liquid medium. Since the cavitation threshold of water is fairly well defined as about one watt per square centimeter at frequencies below 1000 cycles per second, the production of low frequency cavitation in water requires the loading of a transducer face to this intensity level. The load placed on the transducer face by the water is determined by the impedance of the water as seen by the transducer, a complex impedance composed of real and reactive components. It is the real component, or radiation resistance, which determines the intensity of sound generated at the transducer face and propagated into the liquid.

Radiation Resistance

The total radiation resistance of the liquid, as seen by the transducer face, is a mechanical resistance defined in terms of force-time per length. Since the cavitation threshold has been defined in terms of intensity, or power per unit area, it is convenient to define a specific radiation resistance as mechanical resistance per unit area with dimensions of force-time per volume. Designating the specific resistance as r, the sound intensity I is,

$$I = rv^2 \tag{1}$$

where v is the rms velocity of the transducer face. In the case of a sinusoidally driven transducer the intensity becomes,

$$I = \tfrac{1}{2}rV^2 \tag{2}$$

where V is the peak velocity of the transducer face.

Since the velocity of the transducer face is determined by the frequency and displacement, both of which may be controlled readily, the parameter of primary importance at this point is the specific radiation resistance. In determining this value, the case where the transducer radiates into a semi-infinite medium, commonly called a free field, will be considered first because in this case there is no reflected, or returning wave, to complicate the radiation. Although this case does not hold when the radiation is into a closed container, considerable insight may be gained into the relationship of frequency and radiation.

Two types of sound waves may be transmitted by the transducer; plane or spherical. In the case of plane wave transmission, the radiation resistance for a given medium is a function of time and a single space coordinate. The result is the well known characteristic impedance, ρc, the product of the density and the velocity of sound in the medium. As long as plane wave transmission exists, the radiation resistance is a function of only these two properties of the medium. A representative value of the characteristic impedance of water is $1.4 \times 10^5 \frac{\text{dyne-sec}}{\text{cm}^3}$.

It was shown by Lord Rayleigh (6) that plane wave transmission ceases to exist whenever the wavelength of the radiated sound is greater than the circumference of a circular transducer face. Since a representative value of the wavelength of 60 cycles per second sound in water is about 75 feet, the transducer radius would have to be greater than 12 feet for plane wave transmission. A transducer of this size would be out of the question so 60 cycles per second transmission into water is of the spherical wave form.

The determination of the radiation resistance for the spherical wave form is much more complicated, but this too was computed by Lord Rayleigh (6) for the case under consideration. As modified by Crandall (7) its value is

$$r = \tfrac{1}{2}\rho c\,(ka)^2 \tag{3}$$

where ρc is the characteristic impedance,

a is the radius of the transducer, and

k is the wave number.

The wave number, k, exercises the same function in the space domain as the circular frequency, ω, exercises in the time domain. Its value is given by

$$k = \frac{2\pi}{\lambda} = \frac{\omega}{c} = \frac{2\pi f}{c} \tag{4}$$

where λ is the wavelength of sound in the medium,

ω is the circular frequency in radians per second,

c is the velocity of sound in the medium, and

f is the frequency in cycles per second.

From Eq. (3) it may be seen that the spherical wave radiation resistance is a function of not only the characteristic impedance, ρc, but also of the wavelength of the sound and the transducer area. Since, at a given frequency, the wavelength in water under representative conditions is defined, the only parameter that can be altered is the radius of the transducer face.

Assuming a transducer radius of 10 cm (about 4 in.) the value of ka for 60 cycles per second transmission into water becomes

$$ka = \frac{2\pi f}{c}\, a = \frac{2\pi(60)}{1.4\times10^5}\, 10 = 2.7 \times 10^{-2} \qquad (5)$$

The resistance then is

$$r = \tfrac{1}{2}\rho c(ka)^2 = 3.65 \times 10^{-4}\rho c = 51 \ \frac{\text{dyne-sec}}{\text{cm}^3} \qquad (6)$$

The ratio of plane wave resistance to that obtained above is

$$\frac{\rho c}{3.65\times10^{-4}\rho c} = 2750$$

In comparison with plane wave transmission this indicates very poor coupling between the transducer face and the water, requiring a relatively high transducer face displacement to achieve a given sound radiation intensity.

If the transducer is a sinusoidally driven piston, the amplitude required for an intensity of one watt per square centimeter at its face may be computed readily. The peak velocity, V, is given by

$$V = 2\pi Af \qquad (7)$$

where A is the amplitude and f is the frequency. Substituting this value for V in Eq. (2) and rearranging, the required amplitude is

$$A = \frac{1}{\pi f}\sqrt{\frac{I}{2r}} = 1.66 \text{ cm} = 0.65 \text{ in.} \qquad (8)$$

A similar computation for a frequency of 20,000 cycles per second, where the transmission is plane wave and the piston sees the characteristic impedance of the water, shows the required amplitude to be only 9.55×10^{-5} cm. This value is independent of the piston area at this frequency.

The amplitude computed in Eq. (8) will give the desired intensity at the transducer face only. As an individual wave is propagated into the water, the wave front area is increased because of its hemispherical shape. At a radial distance, R, from the transducer face, the wave front area is $2\pi R^2$. Since the total energy radiated from the transducer face is the product of the face area and the intensity at the face, I_F, the intensity at a radial distance, R, is

$$I_R = I_F \frac{a^2}{2R^2} \qquad (9)$$

At a distance of $R = 2a$, the intensity is decreased by a factor of 8, while at $R = 4a$ the factor is 32. In plane wave transmission the energy is essentially focused and contained within a small area with small change in intensity. The two concepts are illustrated in Fig. 2.

The foregoing has been based on propagation through a "lossless" medium, one in which there is no sound energy absorption by the medium. This cannot be exactly correct, but in the case of no cavitation, the loss is relatively small. Once cavitation is initiated, the energy is rapidly dissipated so that the intensity level drops rapidly with distance of propagation.

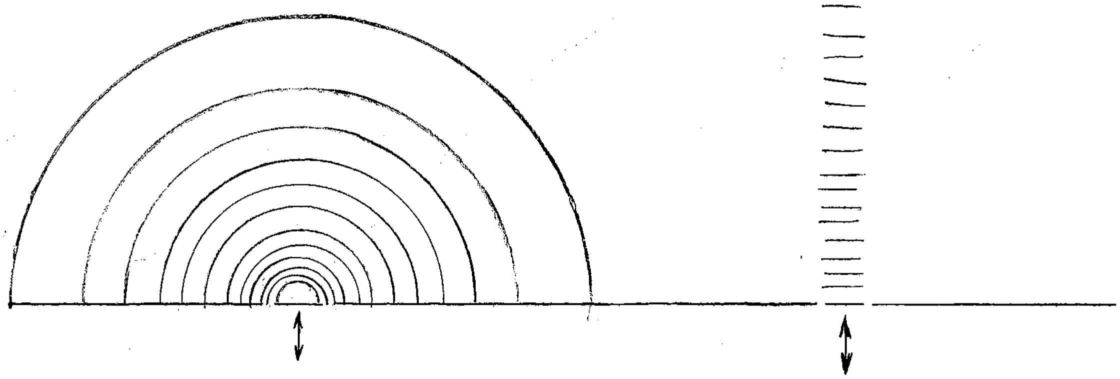

Figure 2. Plane and Spherical Wave Comparison

The combination of spherical wave shape and energy dissipation results in a requirement of an intensity several times greater than the cavitation threshold at the transducer face if the cavitation threshold level is to be attained at an appreciable distance from the transducer.

Mass Reactance

The imaginary or reactive component of the impedance seen by the transducer face results in an apparent increase of mass of the transducer. Its value for spherical wave transmission was computed by Rayleigh for the same conditions as those used in determining the radiation resistance and is

$$x = \frac{8}{3\pi} \rho c (ka) \tag{10}$$

For the conditions under consideration, its value is

$$x = 2.3 \times 10^{-2} \rho c = 3200 \frac{\text{dyne-sec}}{\text{cm}^3} \tag{11}$$

This is about 64 times the real component. If this value is multiplied by the transducer area and divided by the driving circular frequency, the apparent increase of mass of the transducer face is obtained. In this case it is 5.9 lbs.

A comparison of the two impedance components shows that the transducer load is almost purely reactive. Resorting to an electrical analogy, the power factor of the load given by

$$\text{P.F.} = \frac{r}{\sqrt{r^2+x^2}} \tag{12}$$

is only 0.016. The matching of an electrical drive to this load for maximum power transmission would present a difficult problem. A more effective method would be mechanical matching using a linear spring to tune the transducer, with its added apparent mass, to resonance.

Transmission Into An Enclosed Medium

The preceding theory has been based on transmission into a semi-infinite medium or free field, where the wave form is not complicated by reflection or refraction. In the case of transmission into a closed container, both of these phenomena are present to such an extent that the impedance cannot be determined analytically.

Consider a tank with a transducer in the bottom, as shown in Fig. 3, with spherical waves transmitted into the liquid. As the waves

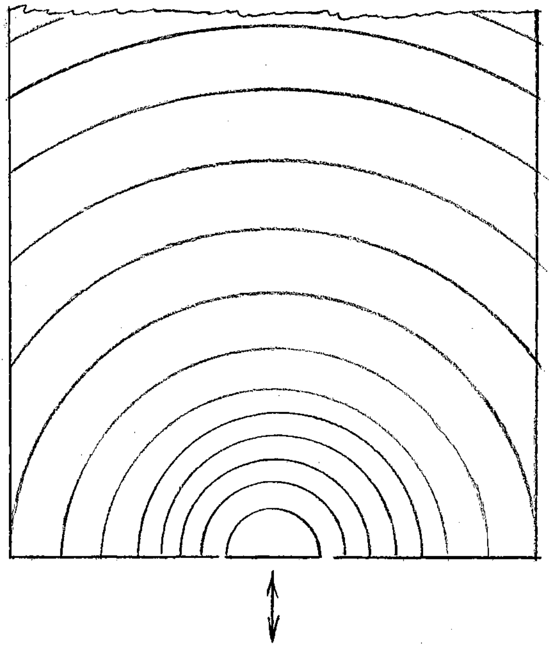

Figure 3. Spherical Wave Propagation

strike the vertical walls there is a combination of reflection, refraction, and transmission owing to the mismatch of impedance at the interface of the liquid and wall, and the angle of incidence of the wave. The impedance mismatch is determined by the ratio of complex impedances of the media forming the interface. For a water-steel interface, about 13% of the energy is transferred through the interface into the steel. (7). The energy which enters the wall encounters an even greater impedance mismatch at the wall-air interface with essentially no transmission into the air. Since the walls cannot be entirely rigid and have little sound absorption capacity, their internal impedance is almost purely reactive and they act as secondary sound sources. (8). The situation is further complicated by the fact that the walls, as solids, are capable of supporting shear waves whereas the liquid is not. The sound waves striking the liquid-air interface at the open top are almost entirely reflected.

The result of the reflection from the surfaces is a complete scattering of sound energy throughout the liquid. The reflected wave is 180° out of phase with the incident wave so that at a particular point and time it may coincide with a radiated wave and cancel its effect, whereas at another point, two or more inphase waves may coincide and become additive. This results in a totally unpredictable sound field and experiment must be resorted to in order to determine the radiation of the transducer into a given enclosed medium.

Theoretical Conclusions

Certain conclusions may be drawn from the foregoing as to the problems to be encountered in an experimental attempt to produce cavitation in water with 60 cycles per second sound energy, as follows:

1. The coupling between a transducer of reasonable size, say 8-inch diameter, and the water will be very poor. It may be expected to be somewhat better for transmission into a closed container than into a free field as computed by Eq. (6), but the value will still be very low. Some method for more nearly approximating plane wave transmission would increase the coupling.
2. The poor coupling between the transducer and the water would result in a low sound intensity for a given amplitude of vibration of the transducer face. The amplitude is the only

parameter which may be varied to increase the intensity when the transducer face area is fixed.

3. Spherical wave transmissions will result in scattering of the sound energy throughout the container with a resulting lower intensity level. Plane wave transmission allows focusing of the energy into a small volume with high intensity.
4. The reactive load on the transducer presents a problem in tuning a given system to resonance whenever the transducer is driven with a limited power source. This problem may be partially alleviated through the use of a spring system to match the mass reactance.

CHAPTER III

PRELIMINARY EXPERIMENTATION

Certain preliminary experiments were conducted with an electronically driven transducer to determine guide lines for the design of the mechanically driven system employed in the production and study of the low frequency cavitation. A schematic of the preliminary system is shown in Fig. 4. Components within the dashed box of the schematic are component parts of the M. B. Electronic Exciter Assembly used to drive the transducer located in the bottom of the cylindrical tank.

Procedure

The variable Frequency Oscillator permitted the selection of the desired driving frequency of the transducer while the amplitude was controlled, within the limitation of the maximum available power, by the amplifier output control. The driving coil, signal generator coil, and center of the transducer were rigidly interconnected so that the motion of one was essentially the motion of the others. Consequently, the displacement, velocity, or acceleration displayed on the vibration meter and oscilloscope was essentially that of the center of the transducer. The dual beam oscilloscope permitted the simultaneous display of the pressure pickup signal and the vibration meter output with a common time base. This, in turn, provided a monitor of the respective

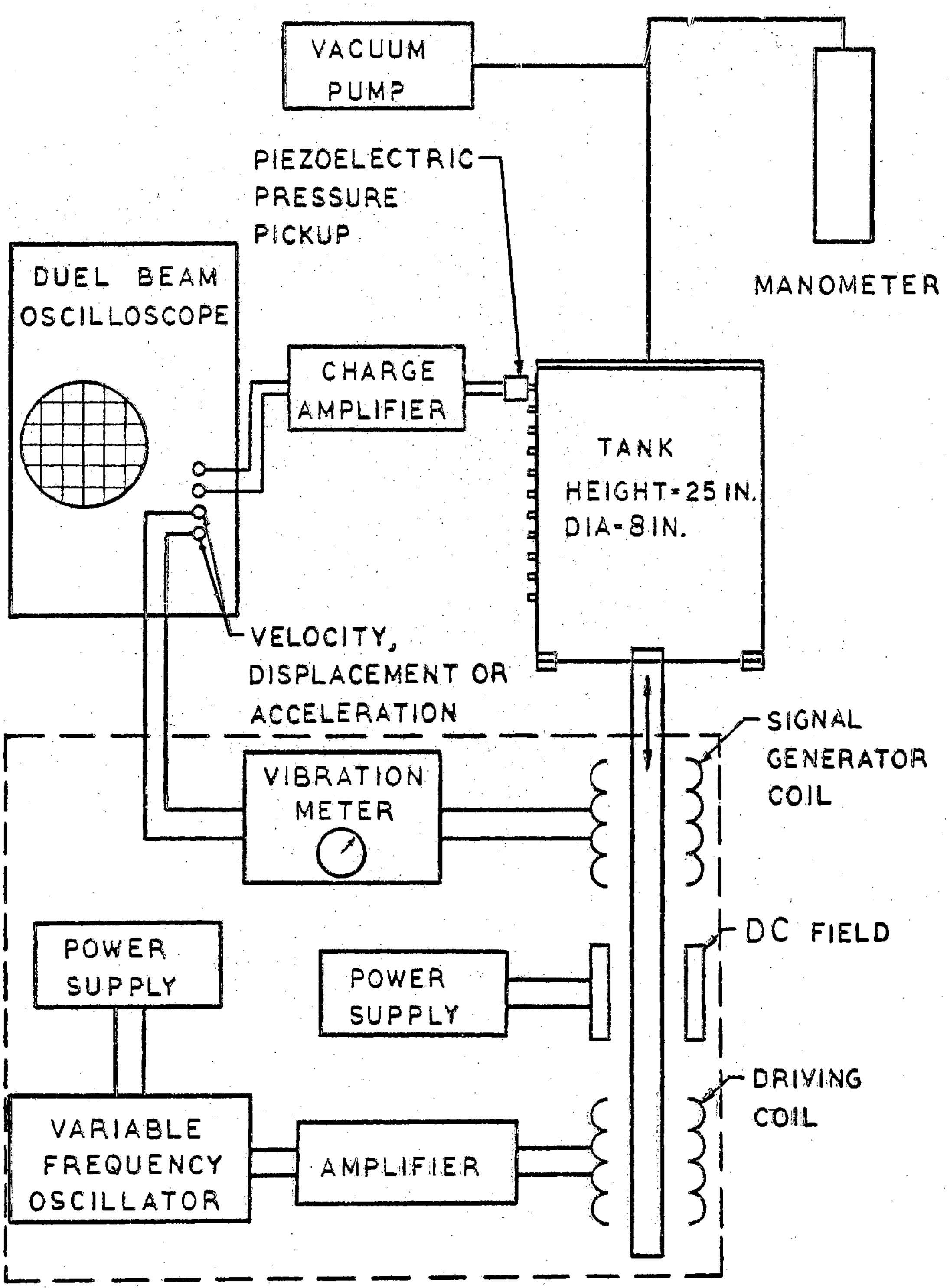

Figure 4. Schematic of the Preliminary Experimentation System

wave forms and the phase relationship. Ten equally spaced pressure taps in a vertical line on the side of the tank permitted pressure pickup at different heights. The vacuum pump and manometer permitted the maintenance of any desired static pressure in the tank between the vapor pressure of the water and atmospheric pressure.

Several transducer configurations were tried during the preliminary experimentation. This was necessitated by the mass reactance of the water and the limited power output of the driver coil. Sufficient displacement of the transducer could only be obtained when it was driven at the natural frequency of the system. Without mechanical tuning of the system with springs the natural frequency was less than 20 cycles per second. Several spring arrangements were tried in an attempt to tune the system to a higher natural frequency.

A flat plate transducer tuned with cantilever springs gave the higher natural frequency but different modes of vibration of the plate were easily excited so that its motion was not that of a flat plate. The most satisfactory arrangement was a 0.2-inch thick stainless steel clamped circular plate of 6.5-inch diameter. This arrangement provided an almost linear spring constant up to a displacement equal to about 1/3 of the plate thickness and vibration in the lowest symmetrical mode at the resonant frequency of the system. The average velocity over the area of such a plate vibrating in its lowest symmetrical mode is only 30.6% of the velocity at the center so that in comparison with a flat piston the amplitude was relatively small. (7). This configuration permitted sinusoidal driving of the transducer whereas a complex wave form was observed for the motion of the other configurations.

Results

The resonant frequency of the system was found to be dependent upon the height of the water in the tank, the static pressure of the water, and, to a small extent, the amplitude at which the diaphragm was driven. Any pronounced change in the resonant frequency would be principally the result of change in reactive impedance since a change in the real impedance component would have little effect upon the resonant point. Two reactive elements were subject to change; the mass reactance of the water and the spring reactance of the diaphragm with amplitude change. The first would lower the resonant frequency while the latter would raise it.

It was found that the change of spring constant of the diaphragm with change of amplitude was small and, in comparison with the mass reactance, could be neglected. Consequently, the varying resonant frequency could be interpreted as a variance of mass reactance of the water. Three cases are of particular importance and are discussed.

1. With a set water height and atmospheric pressure on the water, the natural frequency was observed to decrease as the amplitude of driving the diaphragm was increased. This indicated that an increase of amplitude resulted in more of the water being raised and lowered in phase with the diaphragm and thus increasing the apparent mass of the diaphragm. This was borne out through observation of the action of the water.

2. With a set amplitude of 0.03 to about 0.05 inches, the maximum

which could be obtained, and the water at atmospheric pressure, a change in water height had a very pronounced effect upon the resonant frequency. This change was almost linear, being a decrease of about 2 cycles per second for each inch of increased water height. At amplitudes below 0.03 inches, little change in frequency occurred as the height approached 25 inches. This indicates that for a given amplitude there must exist a water height above which the mass reactance will not increase.

3. Most important of all, with a constant amplitude the frequency change effect with change in water height was decreased as the static pressure on the water was decreased. When the static pressure of the water was reduced to about 2 Hg-inches above its vapor pressure, a change in water height had no effect upon the resonant frequency, the resonant frequency being only slightly lower than that when no water was in the tank. This indicated that separation took place at the diaphragm with the motion of the water lagging that of the diaphragm by a 90° phase angle, thus placing it in phase with the diaphragm velocity. Consequently, the diaphragm saw a real impedance instead of the reactive impedance of the water mass with a resultant higher coupling between the water and diaphragm. This relationship was verified by the phase relationship of the driver velocity and pressure as displayed on the oscilloscope. It was under these conditions that cavitation was most easily obtained. With the presence of cavitation, random high pressure fluctuations prevented any phase relationship comparisons.

Since significant driving amplitudes could only be obtained when the diaphragm was driven at system resonance, it was impossible to study at a given frequency, the effect of any parameter which changed the system resonance. It was also found that the rigidity of the tank mounting arrangement was insufficient to prevent excitation of cross-modes of vibration and that tension on mounting bolts had a marked effect upon the system's fundamental frequency. Because of these variables, repeatability of data was very poor.

The results of the preliminary experimentation indicated that a proper study required a very rigid, highly powered system with a piston transducer so that variation of any of the parameters within the tank would not affect the motion of the transducer face. Further, the effect of lowering the static pressure of the water held promise of much higher coupling between the piston and water than the theory of Chapter II indicated.

CHAPTER IV

EXPERIMENTAL EQUIPMENT AND PROCEDURE

After the completion of the preliminary experimentation, a mechanically driven system was designed and constructed. Particular emphasis was placed on strength and rigidity of the tank and structural members so that extraneous modes of vibration would not be easily excited during the experimentation. Photographs of the resulting system are shown in Figs. 5 and 6. The system schematic, including the associated instrumentation, is shown in Fig. 7.

System Design

In the design of the tank, the desired rigidity and strength of the walls was provided for through the use of ½-inch thick steel material, angle iron ribbing and welded seams. Two 6 × 12 × 1 inch plexiglass windows were placed in the walls to permit interior lighting and observation. The windows were placed in adjacent walls so that the directions of observation, lighting and sound propagation would be mutually perpendicular. Additional bracing material was welded around the windows to aid in maintaining the desired rigidity. A bolted-on and sealed ½-inch thick steel top was used so that the tank would support static pressures below atmospheric. A plexiglass porthole in the center of the top provided access to the tank interior as well as an additional lighting and observation position.

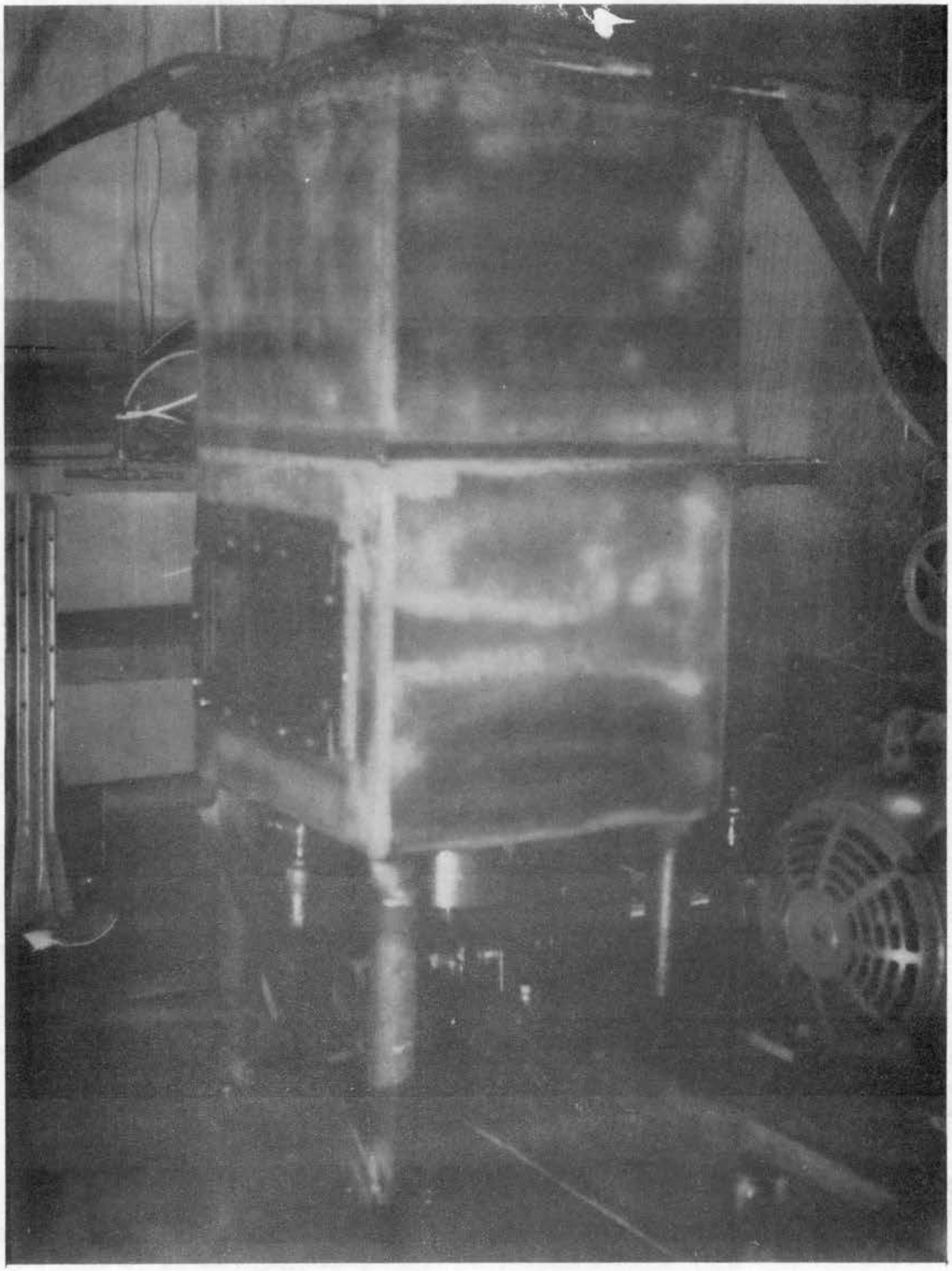

Figure 5: Full View of The Experimental System.

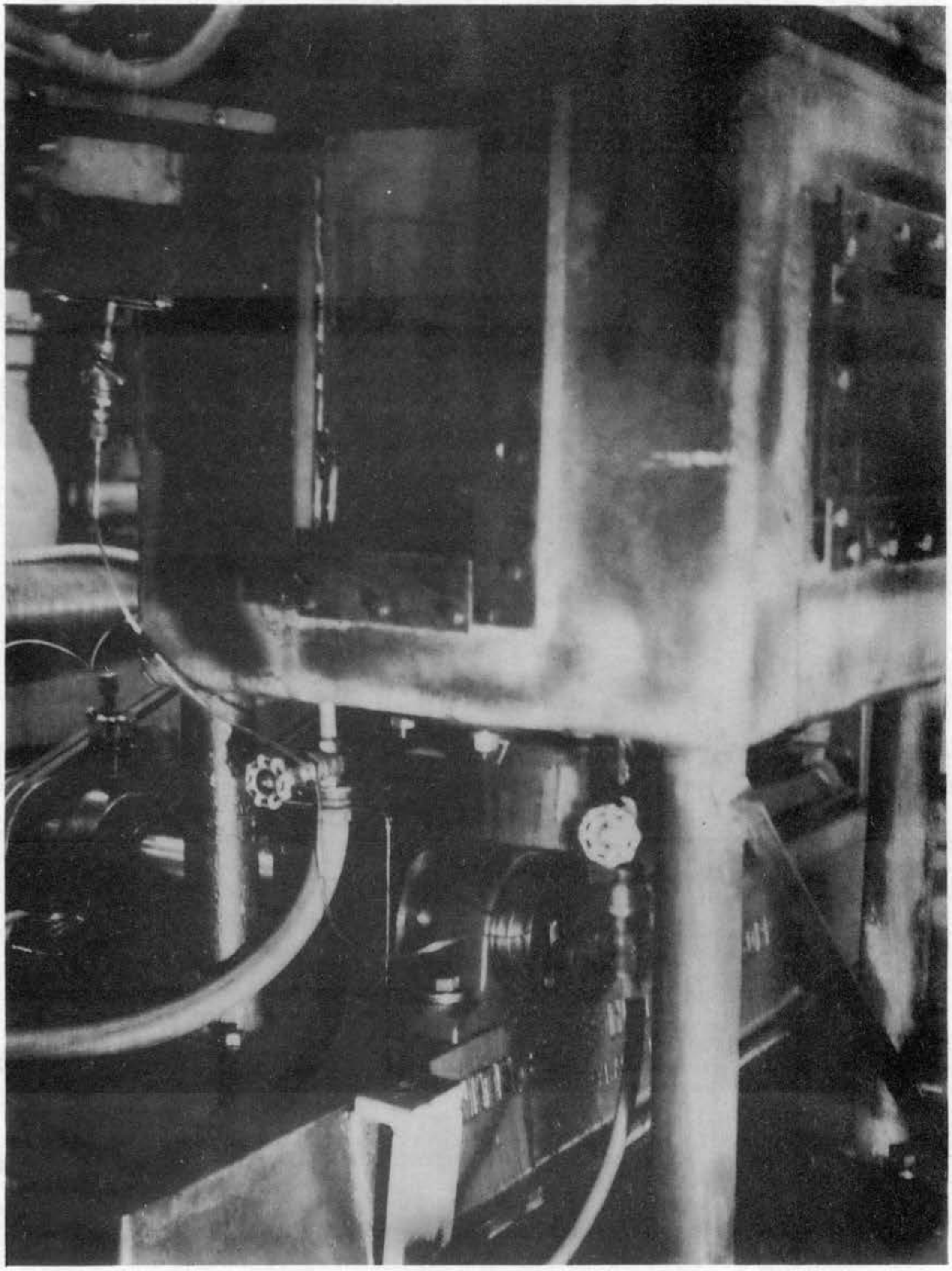

Figure 6: **Driving** Mechanism of the Experimental **System.**

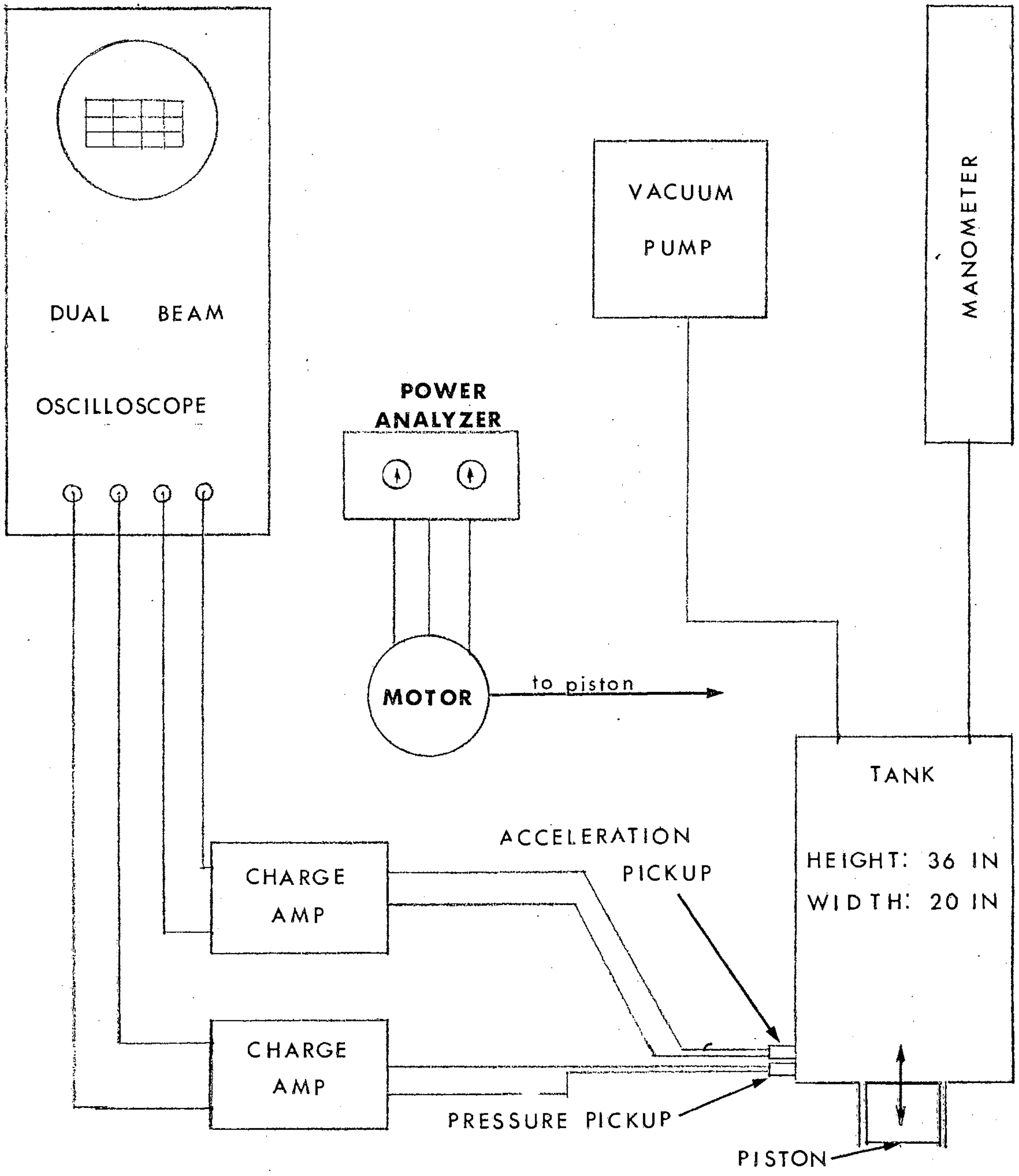

Figure 7. Experimental System and Related Instrumentation Schematic

An 8-inch diameter aluminum piston was used as the sound transducer. Its construction was sufficiently rigid to insure that motion of the flat face would be essentially plane under all loading conditions. The piston was driven by a connecting rod with adjustable amplitude being provided by a cam and sleeve arrangement on the driveshaft. An adjustable weight was used for the balancing of the mechanism as the amplitude was varied. The ratio of connecting rod length to piston amplitude was sufficient to insure sinusoidal motion of the piston face. Leakage between the piston and cylinder wall was prevented by use of two O-rings in conjunction with four back-up rings.

The system was driven by a ten-horsepower, 3-phase, 3515 r.p.m. electric motor through two V-belts. Adjustment of the driving speed was achieved by using V-belt sheaves of the proper radii ratios. The entire system was securely bolted to the concrete floor.

Procedure

Referring to the schematic of Fig. 7, the power input to the electric motor was monitored through the use of the power analyzer which indicated the power dissipated through the motor and the power factor of the load. The piston amplitude was set manually and measured before each run.

A piezoelectric accelerometer pick-up was mounted beside a piezoelectric pressure pick-up with the respective signals fed through the charge amplifiers to the dual beam oscilloscope. This arrangement permitted the simultaneous display of the two signals so that the portion of the pressure signal due to acceleration of the tank wall was indicated. The pressure pick-up had a small acceleration sensitivity, indicating one psi of pressure for each 25 G's of acceleration.

As in the preliminary experimentation, the static pressure on the water in the tank was controlled and measured through the use of a vacuum pump and manometer.

Although not shown in the schematic, a Strobotac was used to measure accurately the rotational speed of the driveshaft. This in turn gave the frequency of oscillation of the piston face. A Strobolux, driven by the Strobotac, was used during certain phases of the experimentation to study the relative motion of the piston face to that of the tank contents.

CHAPTER V

EXPERIMENTAL RESULTS

Cavitation was produced and studied at frequencies of 32 and 69 cycles per second with the mechanically driven system. These frequencies were the result of the motor speed and V-belt sheave sizes on hand. Initial runs made at 32 cycles per second showed that an exact frequency was not critical to the validity of the results. Consequently, it was decided that the installation of special sheaving to produce the desired 60 cycles per second frequency was not warranted.

Experimentation was conducted initially at 32 cycles per second to ascertain that the system was capable of withstanding the higher speed required for the higher frequency. Initially there was some doubt about the capacity of the bearings to withstand the 4140 r.p.m. required to produce the 69 cycles per second frequency. It was found that continuous operation was satisfactory at 32 cycles per second and that runs of 5 minutes duration with short cooling periods could be made at the higher frequency.

The runs at 32 cycles per second proved very productive and interesting. Because of this, and the advantage of continuous running, much of the study was conducted at the lower frequency with the shorter runs at the higher frequency reserved for data comparison.

One limitation of the system became apparent during the very first runs at 32 cycles per second. Even with the strong and rigid construction of the system, violent excitation of the wall and structural crossmodes of vibration resulted during heavy cavitation whenever the piston amplitude was greater than about 0.18 inch. The vibration was so severe that the window and tank interfaces would separate sufficiently to allow water to flow from the tank. Each window was held in place with twenty ¼-inch bolts and the interface sealed with epoxy resin. After shutdown it was impossible to find any break in the seal or looseness in the bolts, thus indicating that the window and interface had been vibrating 180° out of phase with such amplitude that the bolts were elongated within the elastic range sufficiently to open a gap between the window and wall through which the water could flow. Consequently, the majority of the 32 cycles per second runs were made with smaller piston amplitude. An amplitude of 0.17 inch was found to be the most satisfactory for producing severe cavitation without excessive vibration of the tank or structural members.

It was necessary to find some fairly accurate means of determining the degree of cavitation occurring within the tank so that the effect of varying the operating conditions could be studied. Four means were available for this purpose; visual observation, outside sound level, pressure measurement, and power dissipation measurement. All were found to be directly related to the cavitation level but the first two provided only qualitative information. The pressure measurement, as will be shown later, was very difficult to make. Consequently, indications of the power analyzer, with visual and aural verification, proved to be the most effective quantitative indication of the degree of cavitation present.

Since the power analyzer measured all power dissipated by the electric motor, it did not provide a means of distinguishing between the power radiated from the piston face as sound energy and that dissipated in overcoming friction of the system. It may be safely assumed that a resistive loading of the piston face would result in a somewhat larger friction load from the bearings, but the exact relationship between the two would be impossible to determine. The friction power without a resistive load on the piston was measured easily and it, along with certain assumptions which will be discussed as used, produced some quantitative results.

The Effect of Static Pressure Variation

The one parameter, other than piston amplitude, which had a pronounced effect upon the production of cavitation was the static pressure to which the water in the tank was subjected. This effect was almost equally pronounced at both running frequencies. Typical plots of the power dissipated at the two experimental frequencies by the motor as a function of the static pressure on the water are shown in Figure 8. Both were plotted from values obtained with partially degassed water at a temperature of 71°F. The two curves will be discussed separately.

The 32 cps curve:

The data from which this curve was plotted was taken with a piston amplitude of 0.17 inch while the pressure was being gradually decreased from atmospheric. The effect upon the curve from data taken as the

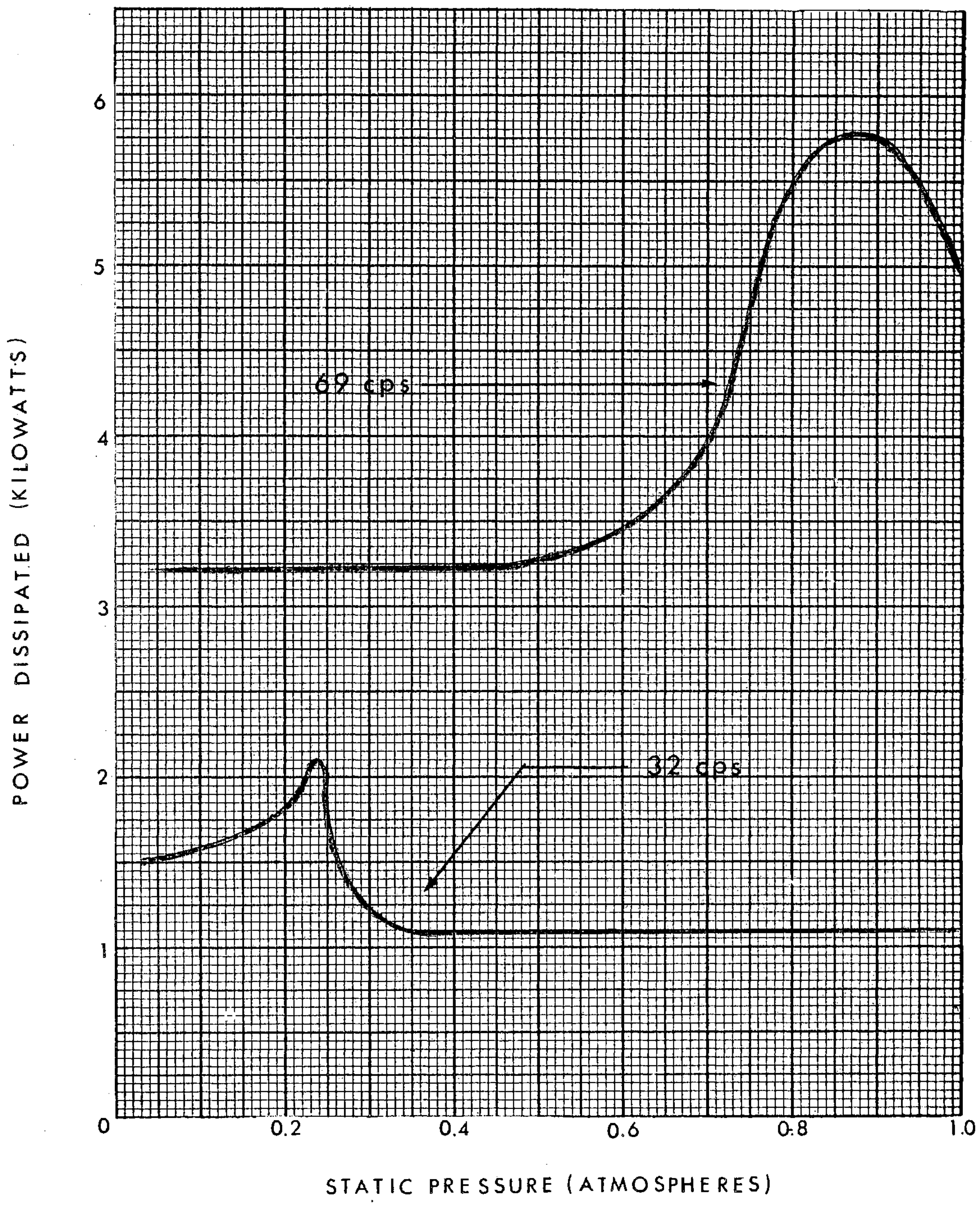

FIGURE 8. Effect of Static Pressure on Power Dissipation.

static pressure was being increased is discussed later.

Referring to the curve, very little cavitation was present when the static pressure in the tank was in excess of 0.5 atmospheres. Lowering of the pressure below this value resulted in an abrupt increase of cavitation followed by a more gradual decrease. As verified visually, the cavitation response was very closely approximated by the power curve. The 1100 watts shown for pressures in excess of 0.5 atmospheres was the same, within the readability of the wattmeter, as that required to overcome the friction of the system when operating at the same frequency and amplitude with no water in the tank. This indicated that there was practically no radiation load on the piston when operating in this static pressure range.

As stated previously, a radiation load on the piston would tend to increase the friction load by an unknown amount. Making an assumption that the ratio of friction load increase to radiation load would not be greater than 1:2 at this frequency provides some radiation load values for consideration. The 32 cps curve would then indicate a peak radiation load on the piston of about 670 watts. Dividing this by the piston area of 330 cm gives an energy intensity of about 2 watts per sq. cm at the piston face. Computation of the radiation resistance required for this intensity at 32 cycles per second gives 72,000 dyne-sec/cm^3 which is one-half the characteristic impedance of the water. This value is 5000 times the radiation resistance which the piston would be expected to see when radiating into a semi-infinite medium at this frequency as computed from Eq. (6).

The reactive load on the motor, as computed from the power and power factor readings, randomly varied from 2560 to 2690 vars for the runs from which Fig. 8 was plotted. The readability of the power factor meter was ± 1% which could result in a ± 4% error in the computed reactive power for the range of power factors being read. This could account for the entire difference in the reactive power values. The mass reactance of the water would be small in comparison to that of the reciprocating parts of the system. This, in conjunction with the readability of the power factor meter, prevented the detection of any change of the mass reactance of the water as a result of static pressure change.

Several factors were found to influence the curve. The use of fresh tap water had the effect of shifting the curve to the right with peak cavitation occurring at a static pressure of about 0.42 atmospheres. Tap water also decreased the power readings from those shown by about 200 watts for all static pressures at and below 0.42 atmospheres. As the tap water was subjected to continued cavitation and evacuation, it bacame slowly degassed with the result that the curve was shifted to the left. At the same time the power readings were increased in the lower static pressure area.

Whether the static pressure was being increased or decreased also influenced the curve. Readings taken as the pressure was being increased had the effect of shifting the curve to the right. Consequently, a plot of the power values for a complete static pressure cycle showed a small hysteresis loop. The rate at which the static pressure was changed had little effect upon the curve.

Piston amplitude changes had a pronounced effect upon the curve. Increasing the amplitude shifted the curve to the right and upward, increasing the friction power as well as the radiation power. Cavitation ceased to occur when the amplitude was decreased to 0.07 inches, the smallest amplitude which could be set on the cam.

The 60 cps curve:

Accurate and repeatable power values with varying static pressure were difficult to obtain during the short runs at 69 cycles per second. However, sufficient data was obtained to determine the general location and shape of a curve. Data taken from runs made with a piston amplitude of 0.11 inch, the largest amplitude used at this frequency, is plotted as the 69 cps curve. The number of readings and the duration of the runs were not sufficient to provide a curve as accurate as the 32 cps curve. Even so, it is sufficient to show the general effect of varying static pressure at this frequency and in effect is the 32 cps curve displaced to the right and upward.

As nearly as could be determined, the effect on the curve of varying piston amplitude was the same as that found at 32 cycles per second. The power value in the low static pressure range was approximately that measured with no water in the tank indicating that little radiation loading of the piston was present in this pressure range. However, some cavitation was present at low static pressures. Although the power spread was more than twice that at 32 cycles per second, the most intense cavitation appeared from sound and visual indications to be no more severe than that observed at the lower frequency.

Very rapid heating of the piston rod bearing was always present during the runs at this frequency. This probably resulted in a widely varying friction load during the runs thus invalidating the power readings. For this reason, little significance is attached to these power readings as being indicative of the radiation load on the piston.

The Resulting Low Frequency Cavitation

Representative photographs of the cavitation produced at a frequency of 32 cycles per second are shown in Figs. 9 through 14. With the exception of smaller bubbles, the cavitation at 69 cycles per second was identical to that shown. In each photograph, the inside of the tank was illuminated by a floodlight shining through the window located on the left. The motion of the piston was vertical to the photographs. The black pieces of material shown floating in the water were flakes of paint which had been chipped from the walls by the cavitation. The interior of the tank had been painted black to provide a better photographic background but the paint did not last for the duration of the photography.

The size of the window through which the photographs were taken was 6 X 12 inches. Identical lighting was used for each photograph with an exposure time of 1/100 second.

A brief description of each photograph follows:

Figure 9: This photograph shows a large cavitation center located in the right side of the tank. It was taken just before the onset of severe cavitation as the static pressure on the water was being lowered. The camera was focused about midway into the tank giving the impression

Figure 9: Severe Cavitation About to Commence.

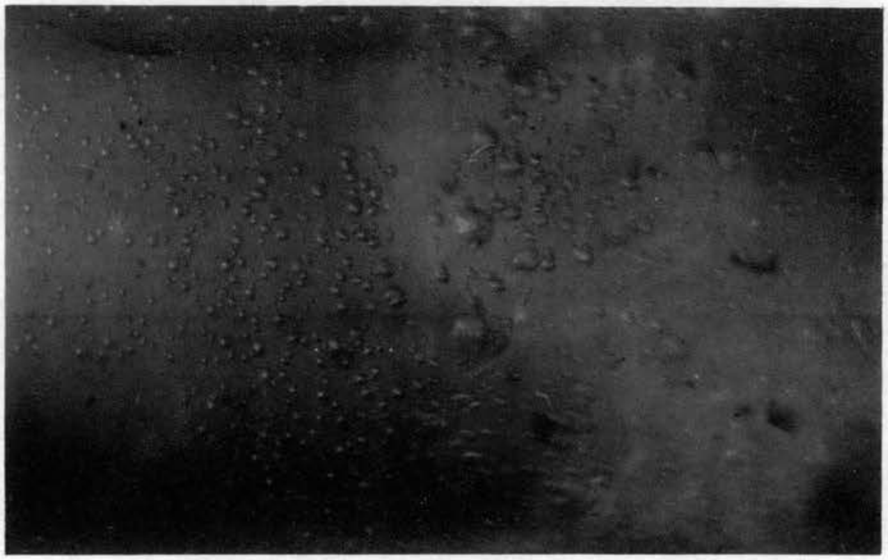

Figure 10: Severe Cavitation.

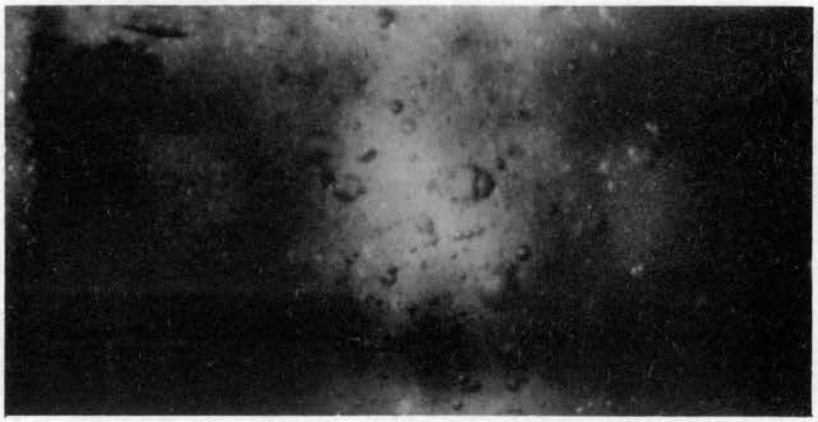

Figure 11: Light Refraction Effects.

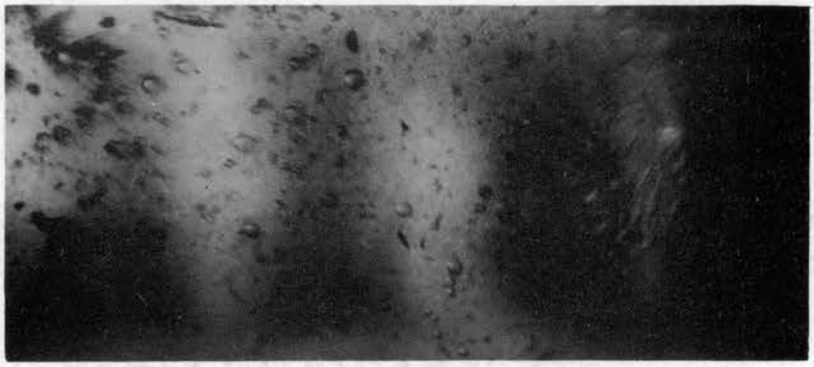

Figure 12: Sharp Wavefront Depiction.

Figure 13: Intense Cavitation on the Window Face.

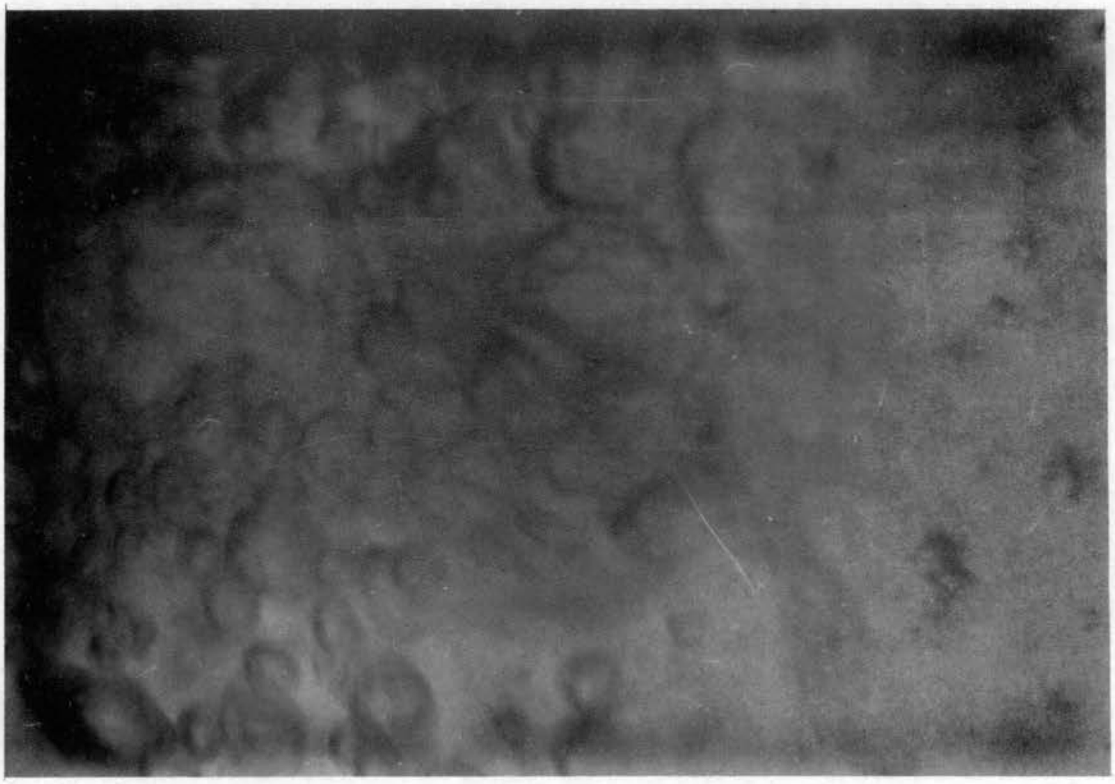

Figure 14: Initial Sound Energy Burst Effect

of relatively poor lighting.

Figure 10: This photograph shows intense cavitation taking place in the tank. The large cluster has broken up into smaller bubbles with other bubbles radiating outward from its center. The speed of this motion is so great that the camera did not stop it thus indicating tails on the bubbles as they moved outward.

Figure 11: This photograph illustrates how a large cavitation center located near the window through which the light was coming blocked and refracted the interior illumination. Much light refraction was noted during the cavitation observation. This had no set pattern but seemed to change according to the migration of the large cavitation clusters which were always present to some degree.

Figure 12: Here is shown a relatively quiet area in the lower right corner of the window. The large cavitation center in the upper right shows a very sharply defined front, something which was not apparent to the naked eye. Bubbles migrating rapidly toward a forming cluster are shown in the upper right corner.

Figure 13: This photograph was taken with the camera focused on the inner face of the window. The cavitating bubbles were against the plexiglass face and represent the most severe cavitation photographed.

Figure 14: The camera was pointed downward into the tank and focused about four inches above the piston face for this photograph. It shows the initial burst of sound energy from the piston as the system was started from rest with the static pressure at that value which produced the most severe cavitation on the previous run. This phenomenon only occurred

with fresh tap and partially degassed water. It is probable that minute air bubbles collected on the piston face during shutdown and that they were liberated with the initial burst of sound energy.

Some general observations of the cavitation are appropriate at this point.

The large cavitation centers shown in the photographs were more prevalent during the less severe cavitation. As the cavitation became more severe, the clusters became fewer and smaller with an accompanying increase in the number of the smaller individual bubbles such as those shown on the plexiglass face in Fig. 13. Most of these would grow to a size of about ¼-inch diameter before collapse. No doubt the intensity of the sound field at the particular location of the bubble influenced the maximum size before collapse.

The audible sound accompanying the severe cavitation, as heard in the vicinity of the tank, might be likened to that resulting from lighting a long string of Chinese firecrackers. It is probable that the sound of only those bubbles collapsing at or near the walls was heard on the outside of the tank. Very little outside sound could be detected from the action of the large cavitation clusters.

The large cavitation centers, or clusters, did not remain stationary for long but migrated about the tank in a random manner. During the short periods when they were stationary, they were always located in a sharp corner of the windows or tank. Sharp objects were suspended in the middle of the tank to see if the cavitation clusters would migrate to them. They did not, so it was indicated that the geometrical shape of the corners had no bearing on this phenomenon.

A Strobolux, driven by the Strobotac at the piston frequency, was used to light the tank interior in an attempt to "freeze" the cavitation field motion for better study. This proved unsuccessful for this purpose but did show water circulation, vertical buoyancy motion of the bubbles and the fact that much of the motion of the cavitation field was not at the driving frequency of the piston. Varying the frequency of the lighting provided no additional information thus indicating that the bubble motion was random.

Any air leaks into the tank had the effect of destroying the cavitation in their vicinity. In order to study this effect further, a method was devised to permit the introduction of controlled amounts of air onto the piston face during runs. With severe cavitation in progress the introduction of small amounts of air on the piston face caused the cavitation to cease with a corresponding power dissipation drop thus indicating that the coupling between the piston face and the water had been broken. Shutting off the air resulted in the cavitation gradually returning to its previous severity within a few seconds.

Dynamic Pressure Measurement

Much difficulty was encountered in measuring the dynamic pressures associated with the sound field and cavitation. The diaphragm of the piezoelectric pressure pick-up ruptured after being exposed to the cavitation for a few minutes. A new pick-up was secured but the same failure occurred again. The diaphragm failures were significant from the standpoint that the pick-ups had been designed by the manufacturer to withstand pressures of 7500 psi. Whether the failures resulted from pressures in excess of this amount or from the negative pressures of the sound field could not be determined. However, it was determined that the diaphragms did not fail from cavitation erosion. A pick-up with a stronger diaphragm was obtained from the manufacturer which performed satisfactorily for the duration of the remaining runs.

Although the pressure pick-up had an acceleration sensitivity of only 1 psi per 25 G's of acceleration, high, randomly varying accelerations accompanying the runs did add somewhat to the indications from the pressure pick-up. The pressure and acceleration pick-ups were mounted together close to a low corner of the tank with the expectation that only small G-forces would be present in the area and that the measured acceleration effect could be used to correct pressure readings. However, an attempt to subtract the acceleration pick-up values from those of the pressure pick-up proved futile. This was due to the complexity of the frequency spectrum of both indications, a factor illustrated in the pressure oscilloscope photographs shown in Figs. 15 through 18.

Each of the photographs was made during 69 cycles per second runs with a piston amplitude of 0.07 inches. In the first three, one centimeter of vertical deflection indicates 2 psi of pressure and one time period of the piston is indicated by 7.25 cm. The respective values in the last photograph are 10 psi and 2.9 cm. Each photograph is discussed separately.

Figure 15: This photograph was taken with little cavitation present in the water. It illustrates the complexity of the waveform with several

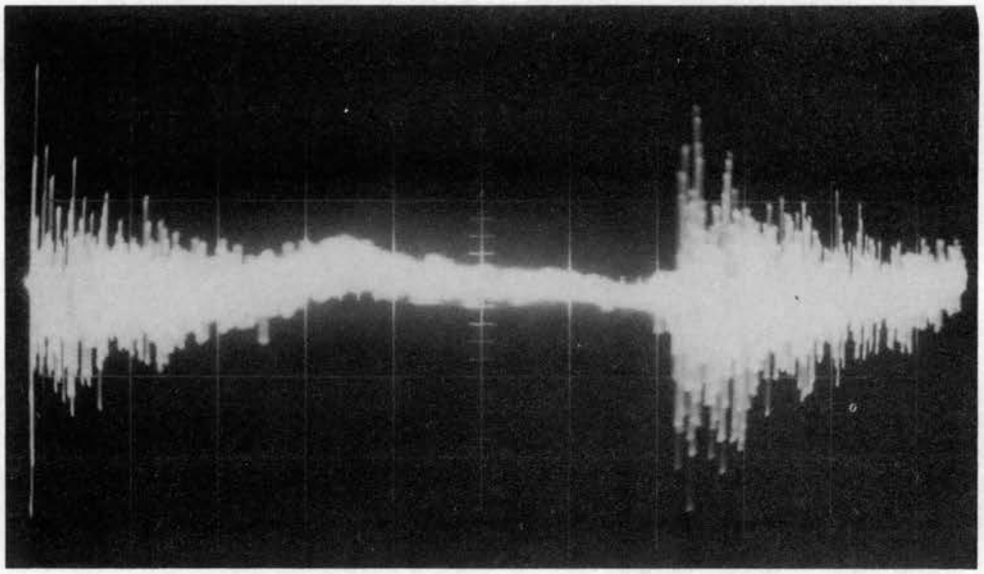

Figure 15: Complexity of Pressure Waveform.

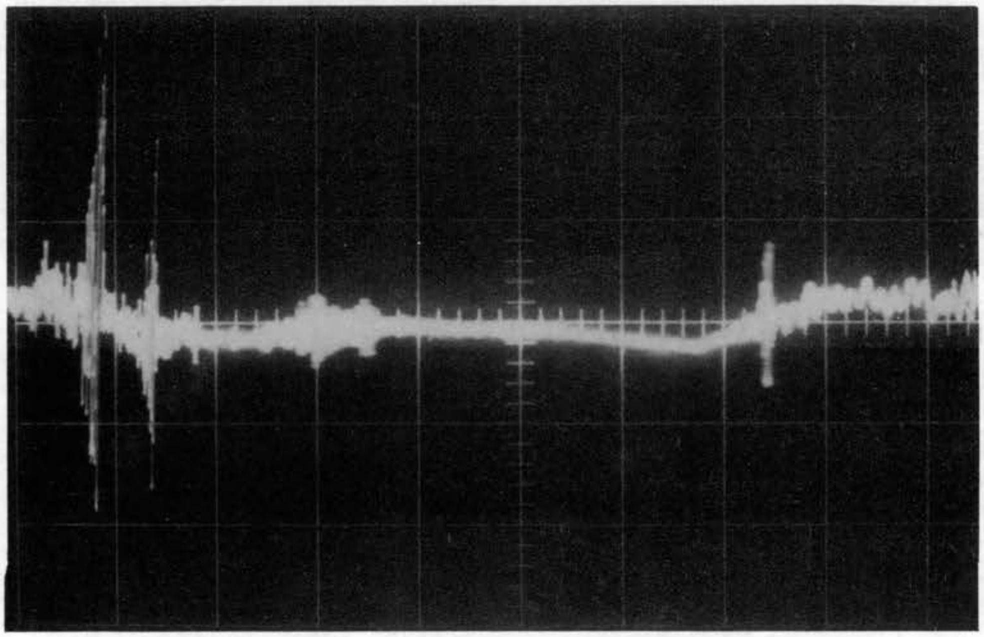

Figure 16: Random Pressure Fluctuations.

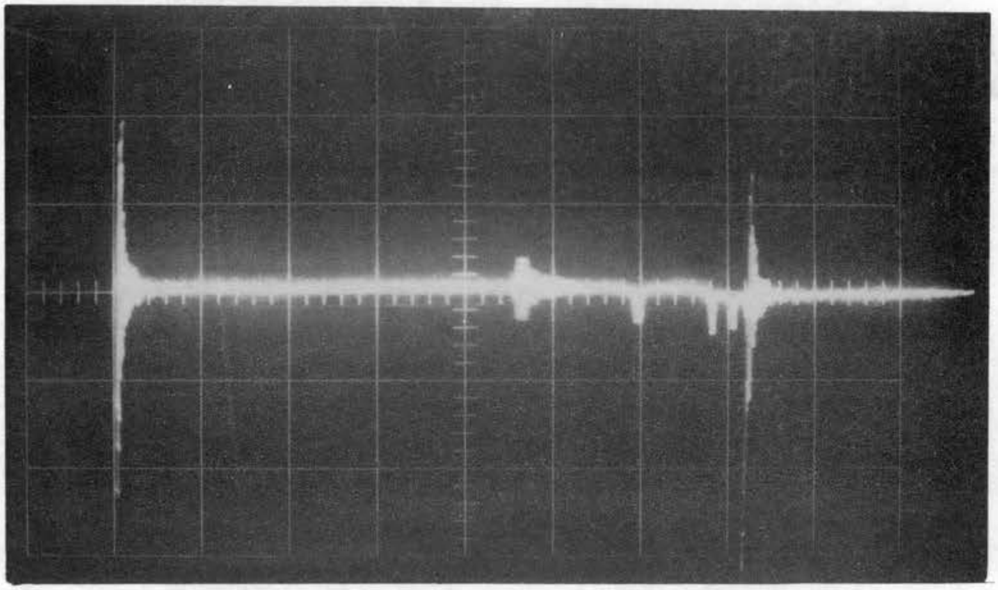

Figure 17: Intervening Negative Pressure Indications.

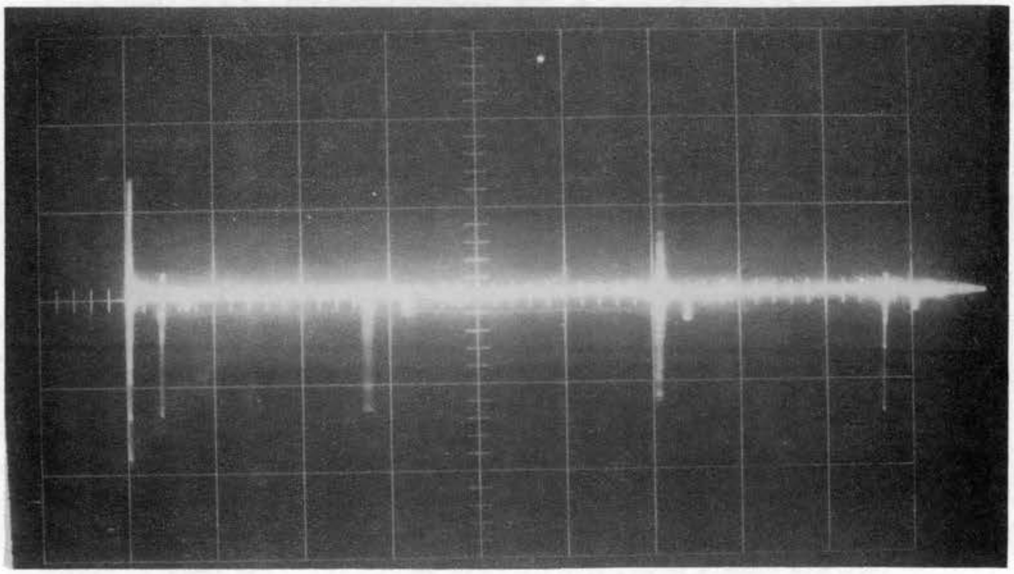

Figure 18: Higher Pressures.

harmonics shown in addition to the fundamental frequency of the piston. The fluctuations following the triggering pulse are almost entirely damped out within one cycle of the piston which is shown by 7.25 cm along the time base.

Figure 16: A greater amount of cavitation was present when this photograph was taken. It shows three sharp fluctuations taking place within one time period of the piston, no two of which are separated by one period. Consequently, with respect to the piston motion, they are random indications which could have resulted from the collapse of bubbles some distance from the pressure pick-up.

Figure 17: This photograph is noteworthy from the standpoint of the two sharp fluctuations located exactly one piston period apart and the three smaller, entirely negative pressures in between. Purely negative pressure indications of much higher value were not uncommon during the study.

Figure 18: This photograph was triggered on a negative pressure of 19 psi with other pressure indications following at random intervals. Three almost entirely negative 13 psi pressures are indicated.

Although not photographed, positive and negative pressures as high as 230 psi were observed during the heavy cavitation at 32 cycles per second. Such pressure indications were random and sometimes separated by a few seconds thus indicating that they were only detected when a cavitating bubble collapsed on or near the pressure pick-up face. Many of the pressure pulses associated with cavitation would be of the Dirac impulse type so it is doubtful that the response of the oscilloscope would be sufficient to display all of these. Therefore, it is probable that many larger pressure pulses than those observed were present but not displayed on the oscilloscope.

Although it was impossible to correct the pressure readings for acceleration effect, the low sensitivity of the pressure pick-up to acceleration was such that this effect would have added little to the true pressure indications. The factor having more bearing on the validity of the pressure indications was that of the oscilloscope being unable to display some of the larger and sharper fluctuations. This undoubtedly prevented the detection of the exceptionally high pressures associated with the cavitation under study.

CHAPTER VI

DISCUSSION OF THE RESULTS AND CONCLUSIONS

Discussion

Although severe cavitation was readily attained with the experimental system at the frequencies of 32 and 69 cycles per second, very little reliable and repeatable quantitative data could be obtained. This was due primarily to the equipment vibration which accompanied the experimental runs. Even though this possibility had been considered in the design of the experimental system, the required rigidity could not be built into it.

The measurement of the power dissipated by the driving motor did provide excellent qualitative data on the cavitation, particularly in regards to the effect from varying the static pressure on the water. It also showed with reasonable accuracy that the radiation resistance load on the piston transducer could be greatly increased by the selection of an appropriate static pressure. The shifting trends of the curves of Fig. 8 indicated that a somewhat larger transducer amplitude or a higher running frequency would provide maximum transducer radiation loading at atmospheric rather than some lower static pressure. A system with less reciprocating mass and greater rigidity would be required to determine whether or not this is true.

The results did not indicate what bearing, if any, the shape of the tank had upon the radiation loading of the piston. It is probable that the high radiation loading obtained under certain operating conditions was solely the result of a mass phase shift brought about by separation at the water-piston interface such as that discussed in Chapter III. It is doubtful that the onset of cavitation aided this phenomenon. To the contrary, it is probable that cavitation actually decreased the radiation resistance since research in the ultrasonic range has shown that radiation resistance on a transducer is decreased by a factor of about 0.3 whenever cavitation is present. (9).

The combination of power and power factor readings showed that during heavy resistance loading the piston did not see a highly reactive impedance as was predicted from theoretical considerations and that mass reactance was fairly constant and independent of the resistive component.

It is probable that the large cavitation clusters observed were produced from vibration of the walls and windows with the severest vibration taking place at the sharp corners. The vibrating areas would then be acting as secondary transducers, vibrating at one or more of the high natural frequencies of the walls. The apparent migration was probably a slow shifting of the points of severest vibration.

Only the small individual bubbles can be considered to be the result of the sound field from the piston. Since the radiation load on the piston was high and cluster activity low when these bubbles were present in quantity, it may be concluded that the secondary sources contributed little to their production.

The surrounding noise associated with the runs was of sufficient intensity to prevent the carrying on of a normal conversation. The intensity was somewhat greater at 69 cycles per second than at 32 cycles per second. In neither case was the sound of sufficient intensity to reach the threshold of feeling. This cannot be said for the preliminary experimentation with the electronically driven system where the intensity was so great as to be objectionable. It appeared that much of the noise was a result of equipment vibration rather than transmission through the tank walls.

A comparison of the produced cavitation with ultrasonic cavitation shows that the cavitating bubbles were much larger and that more energy was associated with each collapse. On the other hand, bubbles within the sound field were relatively sparse so that little can be said about the relative values of energy released. Any comparison of relative cavitation cleaning merits would require controlled cleaning tests.

It cannot be said whether or not the produced low frequency cavitation has any practical possibilities. However, the relatively large size of the cavitating bubbles would provide excellent possibilities for a more detailed study of the mechanism of cavitation in general. Such a study might very well answer many of the questions still unanswered in this field. High speed motion photography would greatly aid such a study by providing a pictorial history of the entire life of the cavitating bubbles. If such studies were undertaken, the possibility of generating the necessary sound by use of a bubble transducer such as the one developed by Sims (10) would be well worthwhile. Such a transducer might solve many of the vibration problems which plagued this study. Preliminary steps were taken to design such a transducer to be used at 32 cycles per second, but it was found that sufficiently strong rubber material for the power transmission required was not available. A special rubber transducer face is currently being fabricated to be used in such a design.

Conclusions

1. Cavitation can be produced through the use of low frequency sound energy but it is very difficult to do so without lowering the static pressure on the water.

2. Satisfactory low frequency sound energy production can be attained with a reciprocating piston provided the appropriate static pressure is maintained.

3. The system used for this study vibrated too much to permit the collection of accurate and repeatable quantitative data.

4. The results of this study did not indicate the cleaning ability of the low frequency cavitation produced.

5. The size of the low frequency cavitation bubbles would be ideal for a study of the mechanism of cavitation.

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