CAVITATING FLOW PAST CYLINDERS AND OTHER OBSTACLES.

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1. INTRODUCTION.

The term 'cavitation' has come into general use to describe the phenomenon first noticed by Osborne Reynolds and described by him as 'cold boiling'. He demonstrated that water, in flowing through a constriction in a glass tube, could flow at such a speed that the pressure at the constriction fell to the vapour pressure of the water. When this occurred the water appeared misty and contained many bubbles of attenuated steam. Since that time the phenomenon has been found to occur in a very wide range of circumstances, in fact in most hydraulic plant where considerable changes of velocity occur. The mechanical significance of the phenomenon is twofold. The space occupied by the vapour may be so great that the flow ceases to follow the paths intended by the designer of the plant and, secondly, at the moment of collapse of the vapour cavities high shock stresses arise, sufficiently great to cause local destruction of the solid boundaries.

Soon after the publication of Reynolds' paper the difficulties experienced by marine engineers, due to cavitation on the blades of screw propellers, were

brought into prominence by Sir John Thornycroft and S.W. Barnaby in their description of the trials of the destroyer "Daring"\(^1\). The propulsive efficiency of this ship was abnormally low with the original propellers and it was not until the sixth set had been fitted, with about 50% greater blade area, that the contract speed was reached. The breakdown in thrust with the unsatisfactory propellers was accompanied by excessive slip and severe vibration. The trouble was attributed to the formation of cavities on the backs of the blades.

In addition to the lowering of efficiency that is caused by cavitation, there is also erosion. This was noticed with steel propellers even before reciprocating engines began to be replaced by turbines, but the use of manganese bronze as a material for the propellers temporarily overcame the difficulty. The propellers of the Mauretania and the Lusitania suffered bad erosion on their earliest voyages, and an investigation by O. Silberrad\(^2\) showed the erosion to be dependent on the structure of the metal, and a new alloy was evolved using aluminium, tin, silicon, manganese, iron and nickel as substitutes for copper and zinc, since zinc was eroded away leaving spongy copper. This new alloy

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2. Engineering, 93, 1912.
was called 'Turbadium'. Silberrad was of the opinion that the erosion was a mechanical effect, though sometimes accentuated or aggravated by corrosion.

In 1912, in a paper entitled 'Investigations into the causes of corrosion or erosion of propellers', Sir Charles Parsons and S.S. Cook\textsuperscript{1} considered the various possible causes and concluded that the nature of the surface, the state of initial stress, stresses under working conditions, and the impingement of water at high velocity on the surface cause little corrosion whereas the erosion is serious, being caused by the hammer action of the water on the propeller blades due to the cavities closing up on, or near, the surface of the blades. Results of experiments made with a diverging mouthpiece, with a syren, and with a cone, supported their conclusions. A mathematical investigation by Cook into the collapse of the water cavities was included in the paper. This latter also received the attention of Lord Rayleigh whose treatment is given in Appendix 1. These calculations show that the pressure of the water hammer is independent of the size of the cavity, depending only on the ratio of its contraction, so that the cavities causing the erosion may be large or small. Cavitation itself will only produce erosion when accompanied by conditions which cause the cavities to collapse in such a way and in such a position

\textsuperscript{1} Inst. Naval Arch. 61. 1919.
that the energy of collapse is concentrated on a small portion of the propeller surface. If the cavity, assumed to be spherical, is at the time of collapse reduced to one tenth of its original size, then a surface pressure of 24 tons per sq. inch could be attained. If the size is reduced to one hundredth then the surface pressure can be as much as 765 tons per sq.inch. Since 1894 a large number of theories of cavitation have been formulated but none give an adequate and reliable explanation of the observed phenomena.

It is convenient to consider cavitation first in relation to screw propellers, although most of the following applies also to turbine runners and centrifugal pump impellers. The thrust of a screw propeller is the resultant axial component of the integrated normal positive pressure on the face of the blade and the negative pressure on the back. These both increase as the speed of advance and the revolutions per minute are increased, and whereas the pressure may go on increasing, the limit of suction is the vapour pressure. When this is reached the fluid evaporates and no further increase in thrust results from further increase in speed and revolutions per minute, there will merely be an increase in the area over which the pressure is equal to the vapour pressure.
Cavitation on the back of the blades would be expected to cause a decrease of thrust but certain instances have been recorded in which an increase of thrust has been experienced as cavitation sets in. In a large number of cases cavitation has been known to cause a considerable loss of efficiency, and in the case of the 'Drake' class of armoured cruisers it was estimated that the waste of power at 23 knots, due to cavitation, would have been sufficient to propel the ship at 14 knots.

Rabbeno\textsuperscript{1} suggested that cavitation might increase efficiency by reducing friction. He recognised four distinct ranges of propeller action,

"(1) normal stage, no bubbles,
(2) low degree of emulsion and a tendency to slightly increased propulsive efficiency owing to the decreased friction,
(3) a stage of equivalence between the energy required to expand the fluid and the energy saved from screw friction,
(4) a stage of decreasing propulsive efficiency caused by the great expansion, work being absorbed by the production of foam accompanied by a reduction of thrust."

Records exist showing that in some instances

\textsuperscript{1} Trans. Inst. Naval Arch. 1929.
admission of air definitely increases the efficiency of a propeller. American designers of high power motor boats, with this object in view, frequently place the screw bracket in front of the propeller. In the case of the racing boat 'Miss America' this necessitated increasing the surface area of the screw, but Gar Wood claimed that he obtained a lubricating effect which reduced skin friction. Rabbeno carried out experiments to determine the proportion of air dissolved in sea water and suggested that the reduced pressure at the blades liberated the dissolved air and so lowered the density, and that this would account in large degree for the decreased thrust experienced. However, an experiment to be described later tends to contradict this and shows that most of the air in the wake of a cavitating propeller is unlikely to have come out of solution. In all probability it is caused by air being drawn in at vortices caused by the screw itself, by other screws, or by the propeller shaft fittings.

Turning now to cavitation effects in connection with water turbines, it is evident from a study of the literature on the subject that for some years designers did not seem to realise that the erosion of turbine blades and ships propellers had a common cause. In both cases it was the increase in speed that brought the phenomenon into prominence. The ship builders had gone over to steam turbines while the water turbine designers were
seeking to reduce the size and cost of electric generators by increasing the speed of rotation.

In a water turbine cavitation can be brought about not only as an effect of speed, but also by installing the machine at an excessive height above the tail water level. The result of either of these conditions, or a combination of the two, is to cause the pressure on the back of the runner blades to be reduced to the vapour pressure, either locally or over a considerable area, and cavities are formed exactly as with ships propellers. The consequences of cavitation are very similar; it reduces efficiency, destroys runner blades and causes heavy shocks in the suction pipe, these effects being particularly serious in the case of high speed turbines of the Kaplan and propeller type. The Propeller Subcommittee\(^1\) noticed that propellers working in cavities and vortices formed by other propellers ahead of them are especially prone to erosive action. In water turbines, eddies occurring at the guide vanes can cause cavitation and erosion of the runner.

The static suction head of a high speed turbine is normally made as small as possible by using a vertical shaft and keeping the runner near or even below tail

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water level. In the interests of economy, designers are forced to use higher specific speeds which imply higher relative velocities. Further, in the search for higher efficiencies draft tube losses must be reduced and this leads to greater dynamic suction heads. From the cavitation standpoint a long blade is desirable but to diminish friction high speed runners must have small blade surfaces. Thus the intensity of pressure on the blades, and therefore the danger of cavitation, is increased. The general practice in Kaplan turbines is to use blades of a length not much greater than the pitch but it would really be possible to use blades with more overlap although this would necessitate increasing the number of blades if the friction effects are to be kept low.

Dr. Betz, from experiments at the Aerodynamic Experimental Institution of Gottingen, shows that as the angle of incidence of the stream with respect to the blade is increased, so also does the suction increase. With a typical blade section, an increase of the angle of incidence from 3 to 15 degrees gives an increase in the greatest vacuum from a little less than the greatest pressure to about three times the greatest pressure.

Cavitation also depends on other factors such as the shape of the blade, and the temperature and the air content of the water, and this makes it difficult to
predict the cavitation limit. Many of the factors entering into the design of a machine cannot be estimated with great accuracy and it is evident that even with very careful design the actual conditions may deviate considerably from those assumed and an error in the blade angle may cause a much lower pressure than is allowed for and as a consequence any 'factor of safety' in the design may be more than cancelled out. For this reason some turbine manufacturers have installed special plant to study new shapes of runners and new materials. Scale models are of great assistance in determining the difference in the magnitude of the assumed and actual losses but they do not give much information in regard to actual cavitation because the controlling factor, the vapour pressure of water, is the same for the model and the prototype and cannot be made to comply with the requirements of similarity.

It has been found that the efficiency of certain types of blades increases somewhat when cavitation occurs, this agreeing with observations with ships propellers. It has been suggested by several investigators that this is due to the water breaking away at the first beginning of cavitation, but later returning to the blade surface, and that at the point where the water does not follow the blade the friction is reduced and therefore the
efficiency is increased. The extent to which this is true is discussed in a later section.

Much of the foregoing also applies to cavitation in centrifugal pumps, although here it is likely to occur at the inlet side of the machine and not at the outlet as in turbines. In recent years there has been a tendency to increase runner speeds for economical reasons and to decrease the number of stages in multi-stage pumps as far as possible. As a result, cavitation is now one of the limiting factors in pump design.

The research now to be described was directed towards determining the effect of cavitation on the pressure distribution around various obstacles and towards a better understanding of three factors. Firstly, the conditions which give rise to an increase or a decrease in efficiency when cavitation is occurring, a problem on which existing literature seems to be evenly divided, both effects having been noted by different investigators. Secondly, the relation between the pressure distribution under cavitating conditions and the position of the erosion caused by the stresses liberated on the collapse of the cavities; and thirdly, the most suitable shapes of blade sections for pumps and turbines.
2. THE CAVITATION TUNNEL.

The cavitation tunnel used in this research is of a type that does not appear to have been described previously. The design was developed by Dr. C. M. White. As can be seen from Fig. 1, the tunnel is similar in principle to the open jet type of wind tunnel, except that the jet is enclosed in a chamber in which the pressure can be adjusted over a large working range, giving pressures up to 50 lbs./sq. inch above atmospheric and down to about 26 inches of mercury below atmospheric pressure.

The experimental chamber of the tunnel is 3 feet above gallery level in the laboratory and the pump which circulates the water is situated at floor level, 20 feet below, so that there is always sufficient head on the pump to avoid serious cavitation in it when the pressure in the working chamber is very low.

Photographs of the working chamber and pump are given in Figs. 2 and 3 respectively. The pump is of the propeller type with a 10 inch diameter impeller and is rated at 5 cusecs at 17 feet head. The pump characteristics are shown in Fig. 4. It will be observed that, as is usual with this type of pump, the power taken increases as the head rises which gives some compensation
for the effect of the obstruction of flow caused by a model.

The vertical supply pipe from the pump to the working chamber is 10 inches diameter up to a point just below floor level where the transition to 12 inches square section commences. The bend immediately before the working chamber is a plain square bend with guide vanes set across the corner. This type of construction was adopted because it gives much more uniform flow after it than is given by a normal circular bend, and is also more compact. The guide vanes are brass castings of curved aerofoil section (fig. 5) and are mounted on brass side plates forming an adjustable assembly. They were originally set so that the trailing edges pointed downwards at an angle of 6 degrees as it was expected that the flow would not leave them absolutely tangentially but would still have an upward tendency. It was hoped that this correction would give horizontal flow immediately before the nozzle. However, as explained in the next section, this correction was found to be unnecessary and the guide vane assembly had to be reconstructed so that the vane tips were horizontal.
Immediately following this bend is the nozzle which is a copper spinning, 12 inches in diameter at its large end and approximately 6 inches in diameter at the discharge end, and of the form shown in Fig. 6. The jet obtained is very satisfactory.

The working chamber enclosing the jet is of square section with detachable and interchangeable cover plates on its four sides. The side pair normally have plate glass windows, while the top and bottom pair hold the model under test.

At the downstream end of the working chamber is a collector for the jet, and immediately following, a diffuser consisting of a taper section of small angle which serves to recover some 57% of the kinetic energy of the jet. The bend following the diffuser is of the Kaplan type and joins the 10 inch diameter pipe leading down to the inlet side of the pump. The bends before and after the pump are of the normal circular type, the one on the supply side of the pump having a curved central diaphragm to assist the flow.

The electrical equipment controlling the speed of the pump is mounted on the wall near the working chamber within reach of the operator. The normal speed range is from 500 r.p.m. to 1600 r.p.m.
corresponding to jet speeds from 10 feet/sec. to 35 feet/sec.

Owing to the system being a closed circuit some means of cooling the water had to be introduced. This consists of about 40 feet of \(\frac{1}{4}\) inch diameter galvanised pipe directly connected across the pump with the greater part of its length immersed in the water in the main sump of the laboratory. Under high speed working conditions the temperature rises from the normal temperature of the laboratory to about 25 degrees C. at which it remains practically constant. Whenever possible the pump is run at maximum speed for about 20-30 minutes before a test in order to reach the steady temperature more quickly. At low speeds the temperature rise is slow and the preliminary warming up is usually unnecessary.

As a result of the rise in temperature and consequent expansion of the water the pressure would slowly rise if there were no outlet. When running at pressures greater than atmospheric the actual magnitude of the pressure is usually unimportant. In any case water can be released occasionally, by opening one of the air release valves on the top of the casing. When a test is being made with reduced pressure in the working chamber the pressure
must be kept constant, and this is accomplished by one of two alternative methods. For pressures between atmospheric and about 15 feet of water negative, a drain cock on the working chamber is connected by a rubber tube to a container which can be raised and lowered between the gallery and the ground floor. This container is kept partly filled with water, being fitted with an overflow, and the end of the discharge pipe from the tunnel is kept submerged in it. The pressure in the working chamber is therefore governed by the height of the container relative to it. This device has been found to be very satisfactory over the range mentioned.

For pressures below 15 feet of water absolute, this device is not so satisfactory owing to air collecting in the connecting pipe. It was found to be practically impossible to maintain a constant pressure by means of vacuum pumps alone, since when the system is practically air free the pressure is considerably changed by the addition or removal of comparatively small quantities of water. An air vessel of about 2 cu. ft. capacity was therefore connected to a cock at the top of the working chamber. It was exhausted to the required pressure by means of a vacuum pump. The volume of this air vessel was
large in comparison with any small increases in volume of the water due to increase of temperature and allowed even quite low pressures to be maintained at a constant value.

The normal supply water contains a large proportion of air and when filling the system it is necessary to eliminate as much as possible of this air otherwise vibration is caused when running at low pressures. This is accomplished by connecting the system to a vacuum pump for several hours. As the pressure is lowered air is liberated throughout the system and collects at various points. Air is more easily released than dissolved, and the released air can be driven out by opening the supply valve, as very little will redissolve when the pressure is increased. When all the free air in the upper part of the system has been driven out, the pump is turned through a few revolutions and air that has failed to rise to the top is brought up by the water. This process is continued until the system is practically air free.

In order that readings might be taken with the pressure in the working chamber at a very low value, it was necessary for the manometers used for measuring
the various pressures to have their upper portions below the centre line of the working chamber, otherwise complete separation of the water column in the connecting tubes might occur. This entailed lowering a portion of the floor by about 2 feet. In order that separation, if it did occur in spite of this precaution, might be seen, a piece of glass tube was fitted in the highest part of the connecting tubes.

The vacuum in the working chamber is measured by means of a simple mercury gauge consisting of a glass tube, the upper end of which is connected to the working chamber and the lower end is immersed in a mercury container, the free surface of the mercury being at atmospheric pressure.

The tunnel and its accessories have proved to be very satisfactory in operation, except that discoloration of the water takes place very rapidly and makes observation difficult, it being impossible to obtain satisfactory photographic records. Various forms of filter have been tried without much success.
Fig. 1. Cavitation Tunnel.
Fig. 2. Cavitation tunnel, working chamber.
Fig. 3. Cavitation tunnel, pump and motor.
CHARACTERISTICS OF 10" PROPELLER PUMP.

FIG. 4.
Fig. 5. Guide Vane Assembly.
3. CALIBRATION OF THE JET.

Before any tests could be carried out in the tunnel it was necessary to determine the velocity distribution across the jet and to correct it if necessary, so that the velocity across the jet should be fairly uniform.

The velocity at the various points was determined by means of a pitot in the usual way. Three positions of the pitot tube were used for the vertical section, these being (a) \( \frac{1}{4} \) inch inside mouth of nozzle, (b) \( 2\frac{5}{8} \) inches from the nozzle (approximately the centre of the working chamber), and (c) \( 5\frac{3}{8} \) inches from the nozzle. From these three measurements it was possible to determine the speed of the jet. Position (b) was used in making the initial adjustments.

Considerable difficulty was experienced in obtaining a pitot tube which was strong enough to withstand the pressure of the jet and yet small enough to cause no disturbance of the flow. The first tube was made from \( 1/10 \)th inch diameter nickel tube, the end being bent at right angles for a length of half an inch to form the mouth. A brass rod extension of the tube passed into a guide at the lower side of the chamber and at the top of the chamber the tube passed
through a gland and was attached to a micrometer head by means of which its position could be adjusted (Fig. 7.).

This first tube had a very short life and 1/8 inch diameter nickel tube was very little better. A 1/10th inch tube inside a 1/8 inch tube, both of nickel and a 1/8 inch brass tube were both tried and abandoned. The final and most satisfactory tube was made of 5/32 inch diameter brass with a short length of 1/10th inch diameter nickel tube soldered on at right angles to form the mouth.

The vertical velocity distribution across the jet, at the centre of the working chamber, before any adjustments were made is shown in Fig. 8 for speeds of the pump of 500 r.p.m. and 1000 r.p.m. It will be seen that too much water was passing through the bottom half of the jet and the top half seemed a little unsteady. In order to determine whether the guide vanes were functioning they were removed and a further set of readings taken, giving the curves shown in Fig. 9. These curves show that without the guide vanes the flow was tending to concentrate in the top half of the jet. Obviously the guide vanes were fulfilling their purpose but they were turning the
flow through too great an angle. As mentioned in the description of the design, the guide vane tips were turned down through $6^\circ$ as it was expected that the flow would not follow the guide vanes very closely and would still have an upward tendency if the tips were horizontal.

Setting back the guide vane assembly at its top end gave an improvement as shown in Fig. 10, the drop in velocity beyond the halfway point becoming more gradual. Further tilting of the vane assembly was of no avail as the tilting effect was counterbalanced by the lowering of the top vane and consequent starving of the top part of the nozzle.

A slight further improvement (Fig. 11) was obtained by fitting before the guide vanes a baffle consisting of two sheets of $\frac{1}{4}$ inch mesh expanded metal, wired together at right angles. With this baffle fitted after the guide vanes the curve Fig. 12 was obtained. This is apparently a great improvement but owing to the nature of the expanded metal, spin of the jet was suspected and the use of the mesh after the jet was abandoned. This resulted, however, in an idea that the jet might be improved by using a better mesh filter in this position and a search for suitable material resulted in obtaining what was
almost the ideal, a portion of a motor car honeycomb radiator, 2 inches thick. The effect of this on the jet, with and without the guide vanes is shown in Figs. 13 and 14. It will be seen that with the guide vanes it has caused the uniform portion of the jet to spread still further upwards and that without the guide vanes the peak at the upper side is greatly diminished. Also of course, there is now very little chance of spin in the jet.

Attention was now returned to the guide vanes and new slide plates holding the vanes were fitted, the tips being moved upwards through 6° while still leaving the assembly across the corner of the upstream right angle bend. This, with the expanded metal before the vanes and the honeycomb after, was the arrangement used during this research.

The vertical velocity distribution at the centre of the chamber for this arrangement is shown in Fig.16 for five different speeds and in Fig. 19 the total energy distribution is plotted with a dimensionless ordinate of pitot tube differential head divided by nozzle differential head. Similar curves for the section just inside the nozzle and at the far end of the chamber are shown in Figs. 17, 18, 20, 21. For any one series the differential head on the nozzle
was kept constant by controlling the speed of the pump.

The horizontal velocity distribution was measured in a similar manner at three sections at the centre of the working chamber. These horizontal sections were taken at the centre line of the jet, 1\(\frac{1}{2}\) inches above, and 1\(\frac{1}{2}\) inches below the centre line. The total energy distribution at these points is shown in Figs. 22-24, the curves given being the average for several speeds. The ordinates are dimensionless, the measured values of total energy being divided by the nozzle head, making the curves comparable to Fig. 20 which gives the total energy distribution at a vertical section at the centre of the working chamber.

The pressure distribution across the jet was measured at the same sections as the total energy distribution by using a 'static' pitot tube. This is shown in Figs. 25-30. It will be seen that at the centre of the working chamber the pressure at the axis of the jet is higher than that of the working chamber by about 5% of the nozzle head. At the nozzle and at the downstream end of the working chamber the pressure at the axis of the jet is higher by about 10% of the nozzle head. Correction of the total energy curves by the corresponding amount at each point gives the kinetic energy curves of Figs. 31-36. In these the
total energy curve is shown dotted.

Although the horizontal distributions are not quite so uniform as the vertical ones, it was decided that since the present research would be confined to comparatively narrow sections in the vertical plane of the jet where the velocity distribution is very satisfactory, no useful purpose would be served at the present stage by attempting further correction. The distribution of kinetic energy at a cross section at right angles to the jet at the centre of the working chamber is shown in Fig. 37, the lines being 'contours' of local kinetic energy divided by nozzle head. It will be seen that there is a region of high velocity at the right side of the jet, (facing nozzle), which may be due to a slight error in the alignment of the nozzle. The major portion of the jet will be seen to have a kinetic energy of within 4% of the mean. This seems to be quite up to normal wind tunnel standard.

The spread of the jet can be determined from Figs. 31 and 33. Around any jet freely discharging into a body of the same fluid there is a mixing zone where a portion of the fluid of the jet is retarded owing to interchange of momentum with the surrounding fluid. As a result the high velocity core is
gradually decreased in area as the distance from the nozzle increases. In the present case the angle that the boundary of the surrounding mixing zone makes with the axis of the jet is about 5 degrees 20 minutes. Experiments by Tollmien\(^1\) have shown that the boundary of the mixing zone of a jet of air freely discharging into the atmosphere makes an angle of just under 5 degrees with the jet axis. Hence it seems that very little interference with the jet is caused by the sides of the working chamber.

Fig. 7. ARRANGEMENT OF PITOT TUBE.
TOTAL ENERGY DISTRIBUTION AT CENTRE
OF WORKING CHAMBER

VERTICAL SECTION

- 730 r.p.m.
- 960 r.p.m.
- 1150 r.p.m.
- 1450 r.p.m.
- 1600 r.p.m.

FIG. 19.
TOTAL ENERGY DISTRIBUTION AT NOZZLE
VERTICAL SECTION

- 730 r.p.m.
- 1450 r.p.m.
- 1600 r.p.m.

FIG. 20.
TOTAL ENERGY DISTRIBUTION AT DOWNSTREAM END OF WORKING CHAMBER

VERTICAL SECTION

@ 7:30 A.M.
@ 1450 A.M.
@ 1600 7 P.M.

FIG. 21.
TOTAL ENERGY DISTRIBUTION
AT CENTRE OF WORKING CHAMBER
HORIZONTAL SECTION

FIG. 22.
TOTAL ENERGY DISTRIBUTION AT CENTRE OF WORKING CHAMBER

HORIZONTAL SECTION 1\(\frac{1}{2}\)" ABOVE C.L.

FIG. 23.
TOTAL ENERGY DISTRIBUTION AT CENTRE
OF WORKING CHAMBER

HORIZONTAL SECTION 1/2" BELOW C.L.

FIG. 24.
PRESSURE DISTRIBUTION AT CENTRE OF WORKING CHAMBER

VERTICAL SECTION

FIG. 25.
PRESSURE DISTRIBUTION AT NOZZLE VERTICAL SECTION.

FIG. 26.
PRESSURE DISTRIBUTION AT DOWNSTREAM END OF WORKING CHAMBER.
VERTICAL SECTION.

FIG. 27.
PRESSURE DISTRIBUTION AT CENTRE OF WORKING CHAMBER HORIZONTAL SECTION

FIG. 28.
PRESSURE DISTRIBUTION AT CENTRE OF WORKING CHAMBER.
HORIZONTAL SECTION 1½" ABOVE C.L.

Fig. 29.
LOCAL PRESSURE
NOZZLE HEAD

+ 0.05

-0.05

-3 -2 -1 0 1 2 3

INCHES

PRESSURE DISTRIBUTION AT CENTRE
OF WORKING CHAMBER
HORIZONTAL SECTION 1\(\frac{1}{2}\)" BELOW C.L.

FIG. 30.
Kinetic Energy Distribution at Nozzle Vertical Section

Fig. 31.
KINETIC ENERGY DISTRIBUTION AT CENTRE OF WORKING CHAMBER VERTICAL SECTION

FIG. 32.
KINETIC ENERGY DISTRIBUTION AT DOWNSTREAM END OF WORKING CHAMBER, VERTICAL SECTION.

FIG. 33.
KINETIC ENERGY DISTRIBUTION AT CENTRE OF WORKING CHAMBER.
HORIZONTAL SECTION

FIG. 34
Kinetic Energy Distribution at Centre of Working Chamber

Horizontal Section 1/2" Above C.L.

Fig. 35
KINETIC ENERGY DISTRIBUTION AT CENTRE OF WORKING CHAMBER
HORIZONTAL SECTION 1/2" BELOW C.L.

FIG. 36.
KINETIC ENERGY AT CENTRE OF WORKING CHAMBER IN TERMS OF NOZZLE HEAD

FIG. 37
4. CAVITATING FLOW PAST A CIRCULAR CYLINDER.

A circular cylinder was chosen for the initial study of the influence of cavitation upon the general flow pattern about an obstacle. This was partly because of the ease with which the pressure distribution can be measured by using one tapping point and rotating the cylinder about its axis. There is also in existence a considerable amount of data relating to the pressure distribution at a circular cylinder at normal pressures for a large range of values of the Reynolds number. This has been summarised very excellently by Thom1.

A stainless steel tube \( \frac{3}{4} \) inch diameter was used, this being the maximum size that could be used in the 6 inch diameter jet. Also for reasonable speed of observation the pressure tapping hole had to be at least 1/16th inch diameter and this would have subtended too large an angle if the diameter of the cylinder had been less. The cylinder was fitted with two \( 4 \) inch diameter end shields, 4 inches apart.

The range of flow in the cavitation tunnel is

1. R. & M. No. 1194.
from about 10 ft./sec. up to 35 ft./sec. which, with a cylinder of \( \frac{3}{4} \) inch diameter corresponds to Reynolds numbers from 50,000 to 230,000 approximately. This includes the critical range in which the flow pattern changes considerably as may be inferred from Thom's curves of pressure at 80 degrees and 160 degrees from the upstream generator (Fig. 38). It will be seen that over this range the pressure at 80 degrees drops very rapidly. Owing to the fact that these curves are based on very scattered test points and are greatly dependent, in this region, on the initial disturbance in the jet, it is wise to regard them as qualitative rather than quantitative.

Accordingly, the first step in the present work was to map out such pressure distributions over the entire velocity range to be considered, the pressure being maintained at a value high enough to prevent cavitation. These pressure distributions are shown in Figs. 39-49 and some of them are combined in Fig. 50 to show the progressive stages. It will be seen that the maximum negative pressure first decreases and then increases with increase of Reynolds number, and at the same time the point where it occurs moves first towards the frontreaching 78 degrees and then further towards the rear of the cylinder, but
VARIATION of PRESSURE with R, at $\theta = 80^\circ$ & $\theta = 160^\circ$

REPRODUCED FROM R & M, 1194

+ - R & M. No 1176
o - " 1194 (Expt.)
$\Delta$ - " 1194 (Theory)
- - Fage.
$\Delta$ - " (Large Cylinder)
$\times$ - Taylor R & M. No 191
$\Box$ - Parkins

FIG. 38.
not beyond 83 degrees in the range of speed investigated. The point of breakaway also moves farther to the rear, its variation being from 90 to 110 degrees approximately. The pressure at 180 degrees at the low speeds is a little more negative than at about 120 degrees; but as the velocity is increased the difference reverses and at the maximum flow the 180 degrees pressure is less negative than that at the 120 degrees.

In order to compare these results with the curves given by Thom for the local pressures at the 80 and 160 degrees points, the two curves of Fig. 51 were drawn. These are very interesting, especially the one for 80 degrees which shows that the change in the flow pattern occurs at a Reynolds number of about 130,000. From $R = 50,000$ to $R = 110,000$, the points give a practically straight line, and there is another straight line from $R = 160,000$ to $R = 230,000$. In the transitional region from $R = 110,000$ to $R = 160,000$, the flow seems to be a little indefinite and is probably liable to fluctuate according to local disturbances. The point of intersection of the two straight lines is at $R = 130,000$. This change of flow is due to a change from laminar to turbulent flow in the boundary layer and is accompanied by a
rapid regression of the point of breakaway from 80 degrees when the boundary layer becomes turbulent. That the motion is actually turbulent is proved by the magnitude of the pressure gradient, which from 100 to 110 degrees greatly exceeds that which could be stable if the motion were laminar.

The effect of additional initial turbulence is clearly demonstrated in Figs. 52-54 which show pressure distributions obtained after increasing the turbulence of the jet by fixing a wire mesh screen, 10 meshes to the inch, across the nozzle. The corresponding curves with the normal jet are shown dotted. With the screen, change of flow begins at a Reynolds number well below 70,000 instead of 130,000. Pairs of curves, with and without the screen, can be found which are almost identical when the comparison is made at suitable Reynolds numbers.

The next step was to determine how these curves vary when the pressure in the working chamber was reduced until cavitation occurred. After a few runs it became apparent that the pressure at the downstream face of the cylinder was subject to an unexpectedly large variation, which at first appeared to be without systematic connection with the actual pressure. On setting the cylinder with its pressure tapping hole at 180 degrees from the upstream
generator and taking a series of readings at steadily diminishing pressures in the working chamber, the curves of Fig. 55 were obtained, each one corresponding to a different velocity. They show that the local pressure remains constant until a certain negative pressure is reached. Then the local pressure rapidly becomes more negative and the curve approaches the line corresponding to the atmospheric pressure less the sum of the vapour pressure and the local pressure. As the chamber pressure is still further reduced the local pressure falls steadily, the points following a line near and parallel to the absolute zero line. The final portion of the curve does not coincide with the absolute zero line because the pressure rises between 80-90 degrees, where cavitation occurs, and 180 degrees where these measurements were made. This rise of pressure decreases as the speed decreases.

Figs. 56-59 show for a number of flows the various stages in the alteration of the pressure distribution curve as the working chamber pressure is lowered. The only point where the local pressure falls to absolute zero is at the peak of the curve just before breakaway. Beyond that point the pressure never falls to absolute zero. The shape of these curves explains the rather peculiar variation of the pressure at the 180 degrees point described above. It is evident that as soon as
cavitation occurs at the point of maximum negative pressure the flow breaks away immediately, and it is this sudden change in the breakaway point that causes the steep rise in the curves for the pressure at the 180° point.

If the pressure in the working chamber is progressively lowered while still maintaining the same velocity, a fringe composed of a number of streaks is seen to form behind the cylinder. At the higher speeds the first signs of the streaks appear just before the first deviation of the 180° pressure curve from the horizontal occurs. The characteristic cracking noises are heard slightly earlier. Both the streaks and the cracking noises increase rapidly as the 180° curve shows decreasing local pressure and by the time the maximum negative local pressure is reached the streaks are very numerous and form a fringe distributed over the full length of the cylinder. Still further lowering of the working pressure causes a lengthening of the fringe without altering its form. At the lower speeds corresponding to the laminar boundary layer the fringe appears and the cracking noises are heard much earlier, and the fringe is fully developed some time before the 180° pressure curve shows the actual inception of cavitation. The reason for this will be discussed later in this section.
The lower curves in Figs. 57 and 58 show the first stages in the development of the cavitation phenomenon, the pressure over the rear portion of the cylinder being more negative than for normal conditions, and the upper curves of Figs. 56–58 show the stage where the pressure over the rear half of the cylinder is becoming less negative and the curve is approaching the zero line. Typical lift and drag diagrams for a half cylinder for the first condition compared with normal flow are shown in Figs. 60 and 61. When the pressure is 27.9 feet below atmospheric pressure and the cavitation is developing, the drag remains practically the same and the lift is increased by 22%.

The diagrams for negative pressure at the rear less than the normal, the working pressure being 28.9 feet below atmospheric pressure, are shown in Figs. 64 and 65 (R = 152,000 approximately) the decrease in drag being 9.1% and the decrease in lift 62.2%. The condition of maximum negative pressure at the rear of the cylinder is also shown. In this case the increase in drag is 51% and the increase of lift 53.2%. If the pressure in the working chamber could be reduced to absolute zero, a decrease of lift and drag of 113% and 38% respectively would be expected.

The measurements of lift and drag show that, with
the type of flow existing at the lower speeds, there is an increase of lift but practically no increase of drag when cavitation first begins but that with the second type of flow the commencement of cavitation corresponds to an increase in the drag as well as in the lift. In both cases, as cavitation becomes more extensive the lift is decreased at a faster rate than the drag.

In order to study the destructive action of cavitation in the various stages of reduced pressure, the stainless steel tube was replaced by a piece of lead piping which was stiffened by means of a \( \frac{\frac{1}{2}}{2} \) inch diameter rod through it, and then skimmed up in the lathe to a smooth surface. The end shields were retained so that the conditions would be similar to those with the stainless steel tube. The jet speed was adjusted to about 28 feet per second \( (R = 168,000) \) and the pressure in the working chamber was lowered to 8.9 feet of water below atmospheric pressure, at which pressure cavitation 'streaks' could just be detected. The cracking noises were first heard at 6.6 feet of water below atmosphere. After six hours running under these conditions the surface of the lead was examined carefully but no effect was visible. The chamber pressure was then lowered to 11.0 feet below
atmosphere and the pipe subjected to a further six hours running. There was still no visible effect. It is probable that these first noises and streaks were formed by inequalities in the surface of the lead, and this is supported by the fact that according to the 180° pressure curves, cavitation should not begin until a working pressure of about 12.0 feet of water below atmospheric pressure is reached. After another six hours at a pressure of 13.4 feet below atmosphere, some erosion was obtained. This is shown in Fig. 70. It began at a well defined line at 90° and extended over an area equivalent to approximately 20°, the depressions being most marked at the foremost part, and diminishing in intensity towards the rear.

The lead pipe was again replaced and run for a further six hours at 15.5 feet below atmosphere in the working chamber. Examination now showed that the eroded area commenced a little earlier and extended very much further round to the rear, the extent now being from 87½° to 160°. After a further six hours running, this time with a working pressure of 19 feet of water below atmospheric pressure, the commencement of the erosion had again advanced slightly, and the pitting generally had increased a little, as was to be expected. The condition of the
The slight advance of the beginning of the eroded region towards the upstream side is caused by the zero pressure and breakaway occurring progressively further towards the upstream side as the pressure is lowered. With zero absolute pressure in the working chamber, the breakaway would probably be in the region of 55°.

If these results are studied in connection with the pressure distribution diagrams for cavitating conditions, it becomes evident that the length of the eroded portion is related to the rate of increase of pressure after the maximum negative value has been reached. Evidently, as one would expect the rate of rise of pressure controls the maximum distance travelled by the cavities before collapse takes place. In the initial stages of cavitation, the pressure distribution curve rises fairly steeply, but as the working pressure is lowered, the rise of pressure becomes more gradual and the cavities can travel a greater distance before they collapse. This increase of cavitated area is not, however, directly proportional to the decrease of working pressure, since after running for six hours at 30.1 feet of water below atmospheric pressure, there was little
addition to the cavitated area. (Figs. 72 & 73).

A microscopic examination of the eroded surface showed it to consist of a mass of lead crystals of various sizes together with pits. The crystals were of normal appearance and not rounded off as would be expected if there were any chemical effects. The largest of the pits was about \( \frac{1}{2} \) mm. across and of similar depth. Photo-micrographs of typical eroded portions are shown in Fig. 74.

At low speeds as the pressure is reduced a visible fringe is formed behind the cylinder before the pressure is low enough to give actual cavitation. The appearance of the fringe is accompanied by noises which are similar to those of cavitation proper. A measurement shows that at a Reynolds number of 54,000, for example, the fringe appears when the pressure is only 30.5 feet water below atmosphere.

A possible explanation of this fringe is that it consists of air bubbles and not cavities. It does not seem to depend on the amount of air dissolved in the water since observations of the decrease of pressure necessary to cause the fringe to appear at various speeds with water that was comparatively air free and with water containing much air, gave very similar
results. To study further this fringe extra turbulence was given to the jet by inserting the wire screen of 10 meshes to the inch previously mentioned across the mouth of the nozzle. The effect of the screen was to delay the appearance of the fringe until the pressure was low enough for actual cavitation to commence. The vibration at fully developed cavitation was much greater with the screen in position than without. Assuming that the fringe behind the cylinder without the screen is due to the liberation of air it seems that the effect of the screen is to cause the air to be liberated before it reaches the cylinder and it is then distributed over the whole area of the jet and not concentrated in the wake behind the cylinder. A prolonged test on a lead tube, without the screen over the nozzle, showed no erosion, and this would seem to support the view that this fringe is caused by air bubbles. The bubbles of vapour formed as a result of the low pressure and which are really the true cavitation effect, are probably completely hidden by this fringe.

A similar phenomenon has also been noticed by Van Iterson\textsuperscript{1} during the stroboscopic examination of the behaviour of centrifugal pump impellers and has

\textsuperscript{1} Cavitation et tension superficielle. Proc. Roy. Academy Science, Amsterdam.
been called by him 'surface cavitation'. It is described as the appearance of small bubbles of air on the surface of the blade from which they are entrained, afterwards vanishing as if absorbed by the metal. It is stated that erosion of the blades occurred at the points where the bubbles disappeared, this erosion being caused by the liberation of the surface energy of the bubbles. These air bubbles seem to remain in existence much longer than the bubbles of water vapour which are formed by true cavitation. This, no doubt, is due to the fact that whereas both the formation and collapse of a vapour bubble may take place in an extremely small period of time the liberation and, to a greater extent, the reabsorption of air from the water requires a much greater time interval, and it is possible that the reason of the absence of erosion caused by this surface cavitation in the present tests is that there is not sufficient rise of pressure for the bubbles to be reabsorbed.
\[ \frac{2(\mu - \rho)}{\rho u^2} \]

FIG. 41.

\( R = 68.100 \)
FIG. 50.
FIG. 51.
FIG. 52

- $R = 70.350$ with screen in front of nozzle
- $R = 68.100$ normal
- $R = 177.600$
$R = 137,700$ with screen in front of nozzle.

$--- R = 137,700$ normal

$--- R = 227,300$
In "Iron!"
$\frac{2(p - p_0)}{\rho v^2}$

---

**R = 96,000 ft**

- NORMAL
- 27.9 ft BELOW ATM.
- 30.6 ft BELOW ATM.

**DRAG**

**FIG. 60**
\[ \frac{2(p-p_0)}{\rho u^2} \]

---

Slip streamline:

\[ R = 96,000 \text{ a.v.} \]

- NORMAL
- 27-9 ft BELOW ATM.
- 30-6 ft BELOW ATM.

LIFT

FIG. 61.
$R = 135,000 \text{ a.v.}$

- NORMAL

--- PRESSURE 24.74± BELOW ATM.

FIG. 63.
\[ \frac{2(p-p_0)}{\rho u^2} \]

\( R = 152,000 \text{ atm.} \)
- NORMAL
- 18.85 ft below atm.
- 20.7 ft
- 28.9 ft

**DRAG**

**FIG. 64**
$\frac{2(h - h_0)}{\rho u^2}$

**FIG. 65**

$LIFT$

R = 152,000 GV.

- NORMAL
- 18.75 ft below atm.
- 20.75 ft
- 28.9 ft
$R = 164,000 \text{ au}$

- NORMAL
- PRESSURE 17.041 BELOW ATM.
  
  $28.61 \text{ Hg}$

LIFT

FIG. 67
R = 197,000 cm

- NORMAL

- PRESSURE 2149 ft BELOW ATM.

DRAG

FIG. 68.
Fig. 70. Initial erosion of lead tube.
Fig. 71. Erosion of lead tube.
Fig. 72. Erosion of lead tube.
Fig. 73. Pitting of lead tube.
Fig. 74. Pitting of lead tube, enlarged.
5. CAVITATING FLOW PAST AEROFOIL SECTION.

The next stage of the investigation was to determine whether cavitating flow has a similar effect on the pressure distribution about other obstacles. The first section to be studied was a symmetrical aerofoil of the Joukowski type which is one of the simplest transformations from the circle. The overall dimensions of the section used were governed by the size of the jet and the practical difficulties of obtaining an adequate number of tapping points. A thickness of \( \frac{3}{4} \) inch and a length in the direction of flow of approximately 3 inches were considered to be the most suitable. The length at right angles to the direction of flow was 4 inches, this being the same as the length of the cylinder between the end shields. A section of approximately these dimensions can be obtained by transforming a \( 1\frac{1}{2} \) inch diameter circle using a transformation of the form

\[ Z = z + \frac{b^2}{z} \]

The necessary calculations for determining the shape of the section are given in Appendix II and the section is shown, twice full size, in Fig. 76. A template was made to the calculated dimensions and the section was machined from brass bar 3" x \( \frac{3}{8} \)" x 4" long, the final
finishing and polishing being done by hand. The pressure tappings were made by drilling a 1/10th inch diameter hole from one end of the section in such a position that it could be met by a 0.04 inch diameter hole forming the actual tapping point, this being drilled at right angles to the surface at the required point. The section was fitted with end plates and was carried by two 3/4 inch brass tubes passing through glands in the cover plates (Fig. 75). Nickel tubes 1/10th inch diameter were soldered into the endwise holes and these tubes passed through one of the carrying tubes, the remaining space being filled up solid with solder. The size of the carrying tube was really the factor controlling the number of tapping points since only 10 tubes 1/10th inch diameter could be passed through the 3/4 inch tube and tubes smaller than 1/10th inch would have been more liable to become choked.

Owing to the thinness of the section near the trailing edge the last tapping point had to be located at about 3/4 inch from the trailing edge so that the pressure distributions can only be measured up to this point.

The aerofoil section was mounted in the tunnel so that the tappings were brought out at the bottom of the working chamber to avoid trouble due to separation of the water column in the connecting tubes at the low pressures. The supporting tube was fitted with a
circle graduated in degrees to allow the angle of incidence of the section to be set. The tappings were joined by rubber tubes to a common tube connecting with one side of a differential manometer, the other side being connected to the working chamber. Screw-down pinch cocks were fitted to each of the rubber tubes as experience has shown that this is the only reliable method when working below atmospheric pressure. Most metal seated cocks tend to leak, and such leaks allow air to enter the tubes and cause inaccuracies in the readings.

The position of the tapping points along the section is shown in Fig. 76. They were placed as close together as possible where the pressure distribution was expected to show rapid variation.

The maximum negative pressure given by the section for zero angle of incidence is only about one half of the stagnation pressure, instead of from one to two times as in the case of the circular cylinder. Consequently the available range of jet speeds at which cavitation will occur is less than for the cylinder. In order that the successive stages of the alteration of the pressure distribution with reduced working pressure could be studied it was necessary that cavitation should begin with a pressure of about 25 feet of water below
atmospheric in the working chamber since the maximum value that can be maintained for any length of time is about 30 feet. Hence the speed, in order that cavitation should occur at this pressure, must be such as to give a stagnation pressure of at least $2 \times (34-25) = 18$ feet. The jet velocity necessary is therefore about 34 feet per second, which is near the upper end of the available speed range of the tunnel and all measurements of pressure distribution with cavitating flow past this section were taken at about this speed, giving a Reynolds number of 600,000 approximately. Actually, measurements of pressure distribution for pressure working over the whole speed range available show very little change with speed, the pressure distribution being practically constant for Reynolds numbers from 100,000 to 650,000.

In the case of the cylinder, from 100,000 to 650,000 as mentioned previously (p. 64), there was a change in the flow pattern and in the form of the pressure distribution diagram at a Reynolds number of about 135,000. No such change of flow pattern occurs in the case of the aerofoil over the whole of the speed range of the tunnel and the measurements suggest that the flow condition is of similar type to that obtaining for the cylinder when the Reynolds number is considerably greater than 135,000. The pressure distribution for zero
angle of incidence and pressure working is only slightly
different from that obtained by transforming velocity
distribution around a circular cylinder in a stream of
perfect fluid into that around the aerofoil, and
calculating the corresponding pressures. This theoretical
pressure distribution is shown dotted in Fig. 76.

The curves of Fig. 76 show that, for zero angle of
incidence, as the working pressure is reduced the pressure
distribution is changed in an exactly similar manner to
that in the case of the cylinder. Breakaway occurs at
the point of maximum negative pressure, and the pressure
over the portion behind this point becomes more negative.
As the chamber pressure is further reduced the distribu-
tion diagram retains its general shape but closes on to
the zero line in exactly the same way as with the
cylinder.

Fig. 77 shows the pressure distribution for normal
and cavitating flow for 2° angle of incidence. The
distribution for the two sides of the aerofoil was
obtained by setting it at 2° on one side of the zero
position and taking a set of readings, and then setting
it at 2° on the other side and taking a second set.
This method is not strictly accurate in that the jet is
not absolutely uniform but is the only alternative to
providing tapping points in both sides of the aerofoil.
The effect of diminished working pressure is similar to
that previously found for zero angle of incidence.
Fig. 78 shows that pressure distribution for normal and cavitating flow for 6° angle of incidence.

In Fig. 79 is shown the distribution for 12° angle of incidence and here a slight difference will be noticed for the early stages of the cavitating flow. The maximum negative pressure on the suction side for pressure working is, of course, greatly increased, being 1½ times the stagnation pressure. When cavitation begins the pressure distribution on the suction side does not change completely as in previous cases, but, in the initial stages at any rate, the pressure distribution curve follows to some extent the line of the curve for pressure working. The distribution for a negative pressure of 13.9 feet of water is shown, and it will be seen that the principal change is in the cutting off of the peak and a widening of the lower portion of the area enclosed by the curve. When the pressure is reduced to 22.6 feet negative the distribution is transitional between the 13.9 feet curve and the type to be expected from previous measurements, which latter is obtained with a negative pressure of 28.3 feet of water. The distribution on the pressure side of the aerofoil retains its shape in the early stages but as the working pressure is further reduced there is less pressure recovery at the rear end.

The curves of pressure distribution show that for small angles of incidence, when the flow still follows the suction side of the aerofoil, the effect of
cavitation is to cause a reduction in the lift and an increase in the drag. The former is indicated by the smaller area between the curves for the pressure and suction sides, and the latter by a decreased pressure over the rear of the aerofoil. For larger angles of incidence there is an increase of lift when cavitation first begins, as shown by the curve for 12 degrees and 13.9 feet vacuum (Fig. 79). No actual figures can be given for the change in lift or drag since the pressure distribution over the rear end is not known.

An attempt was made to determine whether the position of the erosion caused by the cavitation bears the same relation to the pressure distribution as in the case of the cylinder. The aerofoil was subjected to cavitating flow for about 30 hours, the pressure in the working chamber being about 28 feet of water below atmospheric. At the conclusion of this period no definite signs of pitting could be seen, but owing to damage by vibration to the studs of the bracket supporting the bottom end of the aerofoil the test had to be abandoned. Brass is apparently too hard to show pitting in a comparatively short time, a fact that has been noticed by previous investigators who have stated that the surface appears to become hardened by the effect of the hammer blows. Brass profiles have been used for long periods in cavitation apparatus of the venturi type
without showing appreciable damage.

A study of the pressure distribution curves shows that a blade with a section of this type would be unsuitable for use in a pump where the pressure is low at the forward edge of the blade. The rounded nose of the profile causes the minimum pressure on the suction side to occur very near to the nose and as a result cavitation very readily occurs.
Fig. 75. Aerofoil showing tapping points.
AEROFOIL SECTION
Zero angle of incidence

\[ \frac{2(b_p - b_v)}{\rho v^2} \]

Pressure

30.55 ft vac.
27.05 ft vac.

FIG. 76.
AERODRIFT SECTION

2° angle of incidence

\[
\frac{2(b-k_w)}{\rho u^2}
\]

Pressure

27.65 ft vac.

FIG. 77.
AEROFOIL SECTION
6° angle of incidence

\[
\frac{2(p-p_0)}{\rho u^2}
\]

Pressure

25.8 ft vac.

FIG. 78
6. CAVITATING FLOW PAST REVERSED AEROFOIL SECTION.

Some measurements of pressure distribution were made with reverse flow past the aerofoil section for the purpose of examining the general characteristics of the flow. This was accomplished simply by rotating the aerofoil section through 180° and making measurements as before. For zero angle of incidence the point of maximum negative pressure is now moved much nearer the trailing edge as is to be expected from a consideration of the previous results, and consequently any cavitation effects are confined to this region. It was not possible to measure the pressure distribution near the leading edge of the section because the first tapping point was some distance from the leading edge.

Fig. 80 shows the pressure distribution for normal pressure working and for a negative pressure of 26.7 feet of water with zero angle of incidence, and Fig. 81 shows similar curves for 12° angle of incidence. The large angle of incidence causes a high value of negative pressure on the suction side at the leading edge at normal working pressure and with a reduced working pressure this negative pressure must be reduced since the local pressure cannot drop below
absolute zero. It then remains at a constant value over most of the length of the blade and does not become less negative until near the trailing edge. Consequently it is possible that most of the cavities which are formed near the leading edge may not collapse until they reach a point relatively near to the trailing edge where the resulting erosion would be comparatively unimportant. For pressure working it will be seen that there is a point of fairly large negative pressure near the trailing edge on the pressure side, and that the pressure here is more negative on the pressure side than on the suction side. This, however, is completely modified at the lower working pressure. This reversed section is, of course, not suitable for use as a turbine or pump blade since breakaway occurs at the leading edge as soon as the angle of incidence varies appreciably from zero.
AEROFOIL SECTION
Reversed Flow
Zero angle of incidence

\[ \frac{2\rho - \rho \_a}{\rho u^2} \]

FIG. 80.
AEROFOIL SECTION
Reversed Flow
12° angle of incidence

FIG. 81.
7. CAVITATING FLOW PAST BLADE SECTION.

This section was designed on the basis of the results of the previous experiments as being a suitable shape from which a pump blade could be developed. It was hoped to move the point of maximum negative pressure as far as possible towards the rear edge of the blade, the profile being at the same time one that could be easily constructed and have sufficient strength throughout. A study of the pressure distribution curves for flow past a cylinder reveals that the maximum negative pressure at high Reynolds numbers occurs at about 90 degrees and the erosion test has shown that when cavitation occurs pitting will take place at about this point. Therefore it would appear that if a blade section gradually increases in thickness, the surface being convex outwards, to a maximum at a point some distance from the leading edge, then the danger of cavitation is greatly reduced except when the angle of incidence is greater than one half of the included angle at the nose of the blade. An increase in the angle of the nose would cause the maximum thickness and therefore the maximum negative pressure to occur earlier, and in pumps the pressure may not have risen
sufficiently to prevent cavitation by the time this point is reached.

The included angle at the nose of the present section is about 18 degrees so that if the angle of incidence is less than 9 degrees it is to be expected that the flow will still follow the blade on the suction side but that it will break away right at the leading edge if the angle of incidence exceeds this amount. For this first $1\frac{1}{2}$ inches, or one half of its length, the radius of curvature of the two faces is 9 inches. The next portion, from $1\frac{1}{2}$ inches to 2 inches from the nose, is of a uniform thickness of $\frac{1}{4}$ inch and the last $\frac{1}{2}$ inch has a radius of curvature of 5\$ inches giving a thickness of $\frac{1}{4}$ inch at the blunt trailing edge. This blunt end was adopted because it was expected that breakaway having occurred, the shape of the rear portion of the section was of little consequence. The construction and mounting of this section was identical with that described in connection with the aerofoil section; and the tapping points, nine in number, were also similar though differently spaced.

Figs. 83 to 87 show in what degree these predictions have been fulfilled. Considering first the flow with zero angle of incidence, the pressure distribution for which is given in Fig. 83, it will be seen that the pressure decreases fairly gradually
to a negative value of about one eighth of the stagnation pressure at a distance of $\frac{3}{4}$ inch from the leading edge and then remains fairly constant for the next 1$\frac{1}{2}$ inches. The pressure then becomes much more negative, the increase being about 100% at a distance of $\frac{5}{8}$ inch from the rear edge. This sudden decrease of pressure was unexpected although it has been known to occur with certain aerofoil sections and curved plates. The pressure distribution for zero angle of incidence under cavitating conditions could not be determined owing to it being impossible to reduce the working pressure sufficiently, but it is obvious that cavitation would first occur near the rear edge at the point of maximum negative pressure.

As the angle of incidence is increased the negative pressure near the leading edge increases, but the drop and recovery near the trailing edge is still in evidence. For an angle of incidence of 6 degrees (Fig. 84) the maximum negative pressure occurs at $\frac{5}{8}$ inch from the leading edge and is about one quarter of the stagnation pressure. From this point onwards there is a rise for 1$\frac{1}{2}$ inches until the drop near the trailing edge begins. Under cavitating conditions the pressure remains practically constant over the last 2$\frac{1}{4}$ inches. Erosion would probably occur over this portion.

For an angle of incidence of 8 degrees (Fig. 85)
the maximum negative pressure under normal conditions occurs at only ¼ inch from the leading edge, but its value is still quite low, being of the order of 0.3 of the stagnation pressure. The pressure distribution diagram for cavitating flow is similar to that for 6 degrees.

When the angle of incidence exceeds 9 degrees the flow, as was expected, breaks away at the leading edge and the maximum negative pressure occurs at this point. Its value steadily increases from about 0.3 for 8 degrees angle of incidence to about 1.6 times the stagnation pressure for 10 degrees (Fig. 86). The successive stages as the working pressure is reduced are very similar to those obtained with the aerofoil section. In the early stages there is, however, a definite increase of lift and a slight decrease of drag. For 12 degrees angle of incidence and pressure working the maximum negative pressure increases to about 1.75 times the stagnation pressure and consequently cavitation would set in with an even smaller decrease in chamber pressure than for 10 degrees. It will be noticed that there is a fairly good recovery of pressure at the rear end in all cases, the local pressure becoming slightly positive for pressure working. The magnitude of the local low pressure on the suction side near the trailing
edge slowly decreases as the angle of incidence increases but the drop on the pressure side is of the same order throughout.

In order to determine whether the blunt tail was affecting the pressure distribution and causing the decrease in pressure near the rear end, the trailing edge was rounded off by adding a suitably shaped tail piece. This was found to have no effect on the magnitude of the negative pressure just before it but it increased the pressure at the trailing edge to a positive value of about one tenth of the stagnation pressure as shown by the dotted line in Fig. 87, thereby decreasing the drag. Since the flow leaves the surface at the widest part of the section this result is a little surprising, but it is probably due to an augmented reverse flow at the after part of the section.
Fig. 82. Blade section showing tapping points.
BLADE SECTION

Zero angle of incidence

FIG. 83.
BLADE SECTION

6° angle of incidence

\[ \frac{2(p - p_0)}{\rho u^2} \]

Pressure

28.9 ft. vac.

FIG. 84
Blade Section
10° angle of incidence

\[ \frac{2(p - p_0)}{\rho u^2} \]

FIG. 86.
BLADE SECTION

12° angle of incidence

2(p - p_{	ext{atm}}) / \rho u^2

FIG. 87.
8. CAVITATING FLOW PAST REVERSED BLADE SECTION.

For this series of measurements the blade section was used in its second form, with the blunt end rounded off, and turned through 180 degrees so that the rounded edge was foremost. For zero angle of incidence the maximum negative pressure now occurs near the leading edge and there is no sudden fall of pressure near the trailing edge as in the previous case. There is a slight fall in pressure at about one inch from the trailing edge for all angles of incidence, but there is a good subsequent recovery except where the cavitation is in a very advanced stage. For all angles of incidence of less than 9 degrees the maximum negative pressure near the nose is higher than with the flow in the opposite direction, but slightly lower for the larger angles.

At two degrees angle of incidence (Fig. 89), the pressure distribution diagram for cavitating flow shows practically no lift, but it should be noted that a very low pressure, about 4 feet of water absolute, was necessary to cause this. At 6 degrees angle of incidence (Fig. 90) there is an increase of lift when cavitation begins, and this also occurs for 10 and 12 degrees. (Figs. 91 and 92).
The pressure distribution diagrams show a steadily decreasing pressure near the leading edge on the suction side as the angle of incidence is increased, and there is no sudden decrease as in the previous series at 9 degrees angle of incidence. The diagrams are basically similar to those obtained in the previous series and at low working pressures tend towards the same triangular shape. The suitability of this section for use in a pump or turbine will be discussed later.
BLADE SECTION
Reversed flow
Zero angle of incidence

\[ \frac{2(p - p_0)}{\rho v^2} \]

Pressure
30.3 ft. vac.

FIG. 88
BLADE SECTION
Reversed flow
2° angle of incidence

\[
\frac{2(p-p_0)}{\rho u^2}
\]

Pressure

FIG. 89
BLADE SECTION

Reversed flow

6° angle of incidence
BLADE SECTION
Reversed flow
12° angle of incidence

FIG. 92.
9. BLADES IN ECHELON.

The pressure distribution round an aerofoil or blade section forming one of a series is influenced by the presence of the neighbouring blades. Owing to the restricted size of the jet in the cavitation tunnel it is not possible to make measurements under this condition. The results of tests on a series of aerofoil sections in a wind tunnel at Göttingen serve, however, to give an indication of what is likely to happen in the case of a turbine or pump. The pressure distribution obtained for various blade spacings and settings are shown in Fig. 93. It will be seen that with a fairly large spacing there is very little interference. As the spacing is decreased the maximum negative pressure is increased but the point at which it occurs moves towards the trailing edge of the blade, while the negative pressure at the trailing edge for a small spacing is very high. The effect of a change of angle of incidence is much greater when the blades are closely spaced.

If these facts are considered in relation to pumps it will be seen that since the increase of the

EFFECT OF CHANGE IN BLADE SPACING
\[ \alpha = 4.2^\circ \]

EFFECT OF CHANGE IN BLADE SETTING
\[ T = 0.97t \]

Fig. 93.
maximum local negative pressure coincides with a movement of the point at which it occurs into a region of higher average pressure (due to the rise of pressure through the runner) it is possible that the beginning of cavitation will be at much the same suction head as it would be in the case of a single blade.

The effect is much more serious in the case of turbines. The local pressure at the trailing edge of the blade is the most important and it will be seen from the diagrams that whereas the single blade has a slightly positive pressure at this point, this pressure becomes negative for all the given arrangements of blades in echelon. It is most negative for a small spacing or a small angle $\delta$, either of which, in effect, increases the overlap of the blades.

The test of a single blade cannot give a true indication of the deficiency of pressure likely to be experienced at any point on a similar blade in a machine. It can show, however, whether any particular form of blade is basically suitable for use in a machine.
10. THE SOLUTION OF AIR IN WATER.

There has been considerable diversity of opinion as to the effect on cavitation phenomena of air in solution in the water. According to Henry's law, water adsorbs in all cases the same volume of a compressed gas as of gas at normal pressure, if the temperature remains constant. If the pressure acting on a quantity of water saturated with a gas is halved then half of the gas will come out of solution and bubbles of the gas are formed at the walls of the container and at any minute particles present in the water. The truth of this law cannot be questioned but the effect of the time factor and of nuclear particles is not at all clear. In an absolutely pure liquid no bubbles could be formed since a nucleus is necessary for the formation of a bubble. Super-saturation must therefore be possible for a gas in the same way as for a solid in solution. Sea water contains much air in solution and it has been suggested that the low local pressures causing cavitation will result in air being liberated from the water and the air will enter the cavities and act as a cushion when they collapse on moving to a region of higher pressure. The entrance of air in any quantity into the cavities
seems unlikely since the time during which any portion of the water is subjected to the low pressure is extremely small and secondly, even if the air is liberated locally in the form of bubbles, the speed at which these bubbles can move through the water to enter the cavities is so low in comparison with the velocity of flow that the cavities will have collapsed before any air has entered them. The presence of air in a cavity would not necessarily give a cushioning effect since at the high pressure developed the reabsorption of the air would probably be instantaneous, and the magnitude of the blow produced would be very little less than if no air were present. Air in larger quantities does definitely lessen the vibration and noise caused by cavitation and it is often admitted into the draught tube or suction pipe of a machine to this end.

In an endeavour to obtain some indication of the magnitude of the time factor an experiment was made in which water containing air in solution was suddenly subjected to a low pressure and observations made of the rate at which the air was liberated. The apparatus used (Fig. 94) was of a simple nature and was not intended to give exact quantitative results. It consisted of a bottle of about 1 litre capacity,
the bottom outlet of which was joined by a length of brass tube to the top of a burette. The bottom of the burette was connected by a rubber tube to a mercury container. The mercury container was raised to bring the mercury level in the burette near to the top and then the rest of the burette, communicating tube and bottle were filled with water and the top outlet of the bottle closed. The cock on the burette was then also closed leaving the water in the bottle at atmospheric pressure and the mercury container was then lowered about 2 feet. When the burette cock was opened the pressure of water in the bottle was suddenly reduced. If all the air in the water had come out of solution immediately the level of the mercury in the burette tube would have dropped suddenly and remained at the new level, because however small the bubbles they would occupy a total volume equal to the volume of air released. Actually the mercury level dropped suddenly through a relatively small amount and continued falling for some considerable time as shown by the curves of Figs. 94 & 95. Thus it would appear that the release of air from solution due to a diminished pressure is not instantaneous and under cavitating conditions there is but little likelihood of the dissolved air having an effect of any magnitude.
11. THE FORMATION OF BUBBLES.

The cavities formed at a pump or turbine blade form and collapse so rapidly that it is extremely difficult, if not impossible, to study them in detail even with the aid of stroboscopic devices. The author has examined cavitating flow by means of a stroboscope giving up to 6,000 illuminations per second without any definite result, the bubbles having no regular sequence. The formation of steam bubbles, however, can be studied much more easily since there is no necessity for flow past the heated surface at which they are being formed, and there is every reason to think they are similar to bubbles formed by cavitation, both being cavities containing water vapour.

According to Jakob\(^1\) steam bubbles formed at hot surfaces may be classified into three types, their form depending on the nature of the surface. These three types are shown in Fig. 96. Type (a) is found with a very rough surface covered with a thin layer of oil, so that the surface is unwetted and therefore the angle of contact of the water surface with the heating surface is small. The bubble spreads itself on the unwetted surface.

surface and the free edge of the bubble is drawn into a fine wedge between the fluid and the heating surface and is held fast against the action of buoyancy. Such bubbles get very large before they are detached and are immediately replaced by new ones. Type (b) occurs with a polished chromium plated surface. The angle of contact is about midway between that of type (a) and of type (c) which is formed at a clean rough surface which is entirely wetted. This latter type is quickly broken away from the surface while still small.

The sequence of events, as revealed by high speed cinematography, is shown in Fig. 97. The bubble, after breaking away from the surface, does not retain its spherical shape but assumes a one-sided lenticular form, gradually becoming flatter as it rises in the fluid. The frequency of formation of steam bubbles depends on the size of the bubbles at the moment of breaking off since the heating surface is cooled by the water when the bubble leaves.

It has been shown mathematically, as mentioned previously, that if a spherical cavity collapses suddenly an extremely high local pressure is generated. This pressure will be radiated in all directions and the intensity of the blow on a surface even a small distance away is not necessarily large. If, in
cavitating flow, the cavities formed are of shape (a) or (b) and if the flow could drag them along the surface into a region of higher pressure without detaching them, it would appear that there would be a possibility of a very severe blow at the surface, this blow being of much greater magnitude than could be obtained by spherical radiation from a point some little distance away from the surface.

If the bubble leaves the surface instead of being dragged along it, it is highly probable that it will, momentarily at any rate, assume the lenticular shape due to the momentum of the wall of the bubble at the point where it leaves the surface of the blade. If this lenticular bubble now moves into a region of higher pressure its shape will cause the impact due to its collapse to be focussed on to some point which may quite possibly be on or near the surface of the blade.
12. CAVITATION AND THE EFFICIENCY OF MACHINES.

Most bodies, when immersed in a stream of fluid, experience a resistance far greater than that due to friction alone, owing to the formation of layers of discontinuity and vortices in their neighbourhood. These cause the greater part of the resistance and prevent the fluid from closing up behind the body in the same way that it opened in front, the distribution of pressure being unsymmetrical.

To help to explain the effect of cavitation on the efficiency of a machine it is convenient to consider first the flow past a flat plate with its plane at right angles to the direction of flow. Kirchoff's mathematical treatment of this problem\(^1\), assuming that there are two surfaces of discontinuity extending away to infinity, such as AC and BD in Fig. 98(a), gives a value of resistance which is less than that found experimentally. The pressure is assumed constant throughout the space bounded by the plate and the surface of discontinuity, and thus the velocity in these surfaces is constant. If these conditions are observed the theory only gives solutions in which the velocity in the surfaces of

\(^1\) Zur Theorie freier Flüssigkeitsstrahlen, Crelles Journ. Vol. 70. (1869).
discontinuity is equal to the velocity of the undisturbed fluid at infinity. The distribution of pressure is such that at the centre of the upstream side of the plate the full dynamic pressure exists, and this falls away towards the edges, reaching a value equal to the pressure of the undisturbed fluid. On the rear side of the plate the pressure is constant and equal to that of the undisturbed fluid.

In reality the surfaces of discontinuity are extremely unstable and break up to form eddies of varying size. The dead water region does not extend to infinity but the flow closes in again at a short distance behind the plate enclosing eddying fluid, as shown diagrammatically in Fig. 98 (b). This is due to the pressure behind the plate being considerably less than the undisturbed pressure, and it is this suction effect that causes the resistance to be greater than that obtained from Kirchoff's calculations. These calculations, however, do agree with the experimental value of resistance obtained in the special case where, instead of an eddying mass of dead water at the rear of the plate, the space is filled with air so that the surfaces of discontinuity do not break down. It is to be expected that the same low value of resistance will be obtained when the space is filled with the vapour of the fluid as occurs when there is cavitation. The
presence of the vacuous space, by decreasing the eddying dead water region, is therefore a cause of lower resistance.

The application of the foregoing to a more general case is best illustrated by the experiments with the reversed blade section. The development of cavitation is represented in Figs. 99-103. Fig. 99 shows normal working conditions. The flow breaks away at the upper or suction side near the leading edge and there exists between the main flow of the fluid and the blade surface a region filled with eddies of constantly varying sizes and shapes. When cavitation occurs a vacuous region containing only water vapour is formed just behind the leading edge, as indicated by the hatched portion in Fig. 100, and the peak of the pressure distribution diagram begins to flatten off as shown in the lower portion of the figure. Over the rest of the blade the eddying "dead-water" persists and the flow, together with the pressure distribution, remains very similar to that for normal working. Figs. 101-102 show the successive stages in the growth of the vacuous region and the corresponding change in the pressure distribution as the working pressure is progressively lowered until the final condition, as represented in Fig. 103, is reached when the vacuous region extends to the rear
edge of the blade. The pressure distribution diagram is now practically horizontal throughout on the suction side and further reduction of the working pressure can only cause the suction side to approach nearer to the zero line. This final condition, when the vacuous region extends the whole length of the upper surface of the blade, is very similar to that for the flow about a section which is rounded at its front edge and parallel for the remainder of its length, the flow being in the direction of the longitudinal axis as shown in Fig.104. The effect of cavitation, therefore, may be likened to a change of form of the section under consideration, and the amount of increase of lift and consequently of efficiency in a machine depends upon how bad the section was originally, and upon the shape of the vacuous region when cavitation sets in. The vacuous region will restrict the area of the water passage, and this may offset any advantage otherwise gained. The pressure distribution curves provide quite definite evidence of the existence of these successive stages. For example, in Fig.92 the commencement of cavitation corresponding to Fig. 100 can be seen in the curve for 13.8 feet vacuum. The next stage is seen in Fig.91 for 19.4 feet vacuum, and the third stage in Fig.90 for 25.4 feet vacuum while the final stage is to be found in Figs. 91 and 92 for 27.7 and 25.1 feet vacuum respectively.
Exactly the same sequence of events occurs for the cylinder and the aerofoil, but for the former there is, as has been previously stated, a large corresponding increase in drag. It is therefore evident that the supposition made by earlier investigators, namely that the increase in efficiency is due to a decrease in friction, is incorrect and that the increase is really due to the change of effective shape. Should the blade be originally of good form and correctly set, it seems that cavitation could only act adversely.
13. LOCATION OF EROSION IN RELATION TO PRESSURE DISTRIBUTION.

The erosion tests with the cylinder have shown that when there is an appreciable rise of pressure beyond the point where cavitation occurs, the erosion is localised. When there is only a very slight rise in pressure the erosion occurs over practically the whole of that portion of the body downstream of the cavitated area. This may be made clear by reference to Fig. 58 (p.94). For the conditions corresponding to the curve for 18.45 feet below atmospheric pressure, erosion takes place over the area between 90 degrees and 110 degrees. It will be seen that there is a considerable rise in pressure beyond 90 degrees and that after 110 degrees the flow pattern is probably very similar to that for normal working. For the conditions corresponding to the curve for 20.75 feet below atmospheric pressure erosion occurs from 87 degrees to 160 degrees. The advance of the erosion from 90 to 87 degrees is due to the advance of the point of minimum pressure. When the working pressure is further decreased the eroded area will begin slightly earlier, up to about 70 degrees as a limit, and will extend over the whole of the after part of the cylinder.

It was hoped to confirm these results by running
the reversed blade section under conditions corresponding to the curve for 19.4 feet vacuum in Fig. 91 (p.144) for a long enough period to cause erosion. From the above it would be expected that the erosion would be localised in the region \( \frac{1}{2} \) inch to \( 1\frac{1}{2} \) inches from the leading edge. After seventy hours running under these conditions the brass still showed no definite marking and unfortunately the test had then to be abandoned owing to the failure of the inner water lubricated bearing of the pump, caused apparently by the runner being out of balance. A new runner was not delivered in time to complete this test.

It may therefore be of interest to examine some photographs of actual turbine blades\(^1\) which have suffered erosion due to cavitation, and to compare them with the results given above. Fig. 105 shows a portion of a Francis runner. Here there is a badly eroded area near the leading edge and a second farther to the rear. The first is no doubt due to a faulty blade angle and the second to a local deficiency of pressure such as has been met with in the case of the blade section tested. The space between the two pitted areas is free from erosion and there was evidently a considerable rise of pressure after the first and then a fall to cause the second.

In Fig.106 is shown a stainless iron blade that

\(^{1}\) Engineer. Oct. 17, 1930.
has been used in an experimental turbine which had provision for running under considerably reduced pressures. The only erosion is that shown at B which is quite local and is probably due to an irregularity in the leading edge. Fig.107 shows a similar blade, this time of chromium plated cast steel. This again has a local eroded area near the leading edge, and, in addition, a small one right at the trailing edge caused, as before, by a local pressure deficiency and having no real connection with the first area. The erosion shown in Fig.108 extends over the whole face of the blade and was probably due to a sufficiently low pressure in the experimental plant to give a pressure distribution diagram corresponding to that at 27.9 feet vacuum in Fig.87. (p.138).

The pressure given in the pressure distribution diagrams for the blade sections are for turbulent flow in the boundary layer, and therefore even though the diagram may show a region where the pressure is equal to the vapour pressure it would be possible for erosion to occur in this region, because the turbulent fluctuations of velocity and pressure could permit collapse of some of the cavities.
Fig. 105. Portion of Francis runner, showing erosion.

Fig. 106. Stainless iron Kaplan blade.
Fig. 107. Chromium plated cast steel Kaplan blade.

Fig. 108. Cast steel Kaplan blade.
14. TECHNICAL APPLICATIONS.

The application of the results of this research to the design of blade sections must be considered separately for the two types of machines, pumps and turbines. In pumps the average pressure tends to increase as the fluid passes the blades and so pressure deficiencies due to blade form are most dangerous when located near the leading edge, where the general pressure is already low. Near the leading edge of the blade the pressure distribution depends greatly upon the angle of incidence and this becomes a factor of importance in pump design.

Pumps, when delivering against a constant head, have necessarily to be driven at approximately constant speed. If the discharge then happens to be that for which the pump was designed the angle of incidence of the leading edge of the blades will be zero. Should the external load differ from that anticipated in the pump design, then the discharge will differ, and also the angle of incidence. A designer is unlikely to more than ± 30% wrong in his estimate of the total head and flow characteristics of the system against which the pump is to work, and this error, with a blade angle of 45 degrees, represents a variation of about ± 10 degrees in the angle of incidence. The
practical range for the angle of incidence thus lies between 0 and ± 10 degrees.

The influence of the angle of incidence upon the maximum negative pressure is shown in Fig. 109, for three different sections. It will be seen that, providing the angle of incidence is less than one half of the included angle at the nose, the blade section with the pointed end foremost gives much less negative pressure and therefore less risk of cavitation. As soon as this angle is exceeded the curves cross and the pointed blade section becomes inferior to the other two, both of which have rounded leading edges. An increase in the nose angle will delay this sudden change but the advantage is offset by the blunter form of the blade which causes the lowest pressure to occur nearer the leading edge of the blade and to be of greater magnitude. In most cases it would seem preferable to adopt a smaller angle than 10 degrees providing the periods during which the pump would be subjected to overload were comparatively short. Both the aerofoil and the reversed blade section show a steadily increasing value of negative pressure as the angle of incidence increases. At small angle of incidence the reversed blade section appears to be better than the aerofoil, but it must be remembered that the point where the maximum negative pressure occurs is
farther forward due to the smaller radius of the nose.

In a turbine there is a fall of pressure through the runner and cavitation will tend to occur near the trailing edge. It is therefore necessary to adopt a shape of blade that will give a good recovery of pressure along the after portion of its length. To fulfil this requirement the after part of the blade must be tapered off at as fine an angle as possible consistent with strength. The aerofoil section tested gives a very good recovery of pressure but difficulties of construction and liability to damage would prohibit its use in a turbine. The blade section reversed (Fig. 88.) gives a fairly good recovery of pressure and is definitely better than a rounded or bluff form of tail as given by the aerofoil reversed or the blade section used normally. With the bluff tail there is a region of low pressure very near to the trailing edge and therefore a definite risk of cavitation. Change of angle of incidence slightly modifies the magnitude of the negative pressure at the rear of the suction side of the blade section, causing it to decrease slowly as the angle increases, but the value on the pressure side remains constant. With a fall of pressure along a blade section of this type, as in a turbine, it seems highly probable that there will be a risk of cavitation on the pressure side of the blade.
To summarise - for a pump in which the overload is not likely to exceed 30% except for very short periods there is no doubt that a form similar to that of the blade section tested is most suitable. For large and continuous overloads, which of course should not normally occur, the aerofoil will give slightly better results. In the case of a turbine the best form is definitely one similar to the blade section reversed which is less liable to cavitation at small angles of incidence and is only a little worse than the aerofoil at angles exceeding 9 degrees. A bluff form of trailing edge should be avoided owing to the low pressure associated with it.
Fig. 109.
Abstract from Lord Rayleigh's paper "On the pressure developed in a liquid during the collapse of a spherical cavity" - Philosophical Magazine (6) 34; 94-98 (1917).

Besant (Hydrostatics and Hydrodynamics, 1859, par. 158) formulated the problem - 'An infinite mass of homogeneous incompressible fluid acted upon by no forces is at rest, and a spherical portion of the fluid is suddenly annihilated; it is required to find the instantaneous alteration of pressure at any point of the mass, and the time in which the cavity will be filled up, the pressure at infinite distance remaining constant'.

Since the fluid is incompressible the whole motion is determined by that of the inner boundary. If \( U \) be the velocity and \( R \) the radius of the boundary at time \( t \), and \( u \) be the simultaneous velocity at any distance \( r > R \) from the centre then

\[
\frac{u}{U} = \frac{R^2}{r^2} \tag{1}
\]

and if \( \rho \) = density the whole kinetic energy of the motion is

\[
\frac{1}{2} \rho \int_{R}^{\infty} u^2 \cdot 4 \pi r^2 \, dr = 2 \pi \rho U^2 R^3 \tag{2}
\]
Again if \( P \) = pressure at infinity and \( R_0 \) = initial value of \( R \), the work done is

\[
\frac{4}{3} \pi P \left( R_0^3 - R^3 \right)
\]

Equating (2) and (3) -

\[
U^2 = \frac{2P}{3 \gamma} \left( \frac{R_0^3}{R^3} - 1 \right)
\]

expressing the velocity of the boundary in terms of the radius. Also since \( U = \frac{dR}{dt} \)

\[
t = \left( \frac{3P}{2P} \right) \int_{R}^{R_0} \frac{3/2}{(R_0^3 - R^3)^{1/3}} \left[ \int_{\beta}^{1} \frac{\beta^{3/2}}{\left(1 - \beta^3\right)^{1/3}} \right] \frac{d\beta}{\beta} = \frac{3P}{2P} \int_{R}^{R_0} \frac{1}{(R_0^3 - R^3)^{1/3}} \left(1 - \beta^3\right)^{1/3} \frac{d\beta}{\beta}
\]

if \( \beta = \frac{R}{R_0} \). The time of collapse to a given fraction of the original radius is thus proportional to

\[
R_0 \int_{\beta_{\text{final}}}^{1} \frac{3/2}{(R_0^3 - R^3)^{1/3}} \left(1 - \beta^3\right)^{1/3} \frac{d\beta}{\beta}
\]

which might have been anticipated by a consideration of dimensions.

The time \( T \) of complete collapse is obtained by making \( \beta = 0 \) in (5).

Writing \( \beta^3 = z \), we have

\[
\int_{0}^{1} \frac{\beta^{3/2}}{(1 - \beta^3)^{1/3}} \frac{d\beta}{\beta} = \frac{1}{3} \int_{0}^{1} z^{-1/6} (1 - z)^{-1/3} \frac{dz}{z}
\]

which may be expressed by means of \( \Gamma \) functions. Thus --

\[
T = R_0 \int_{0}^{1} \left( \frac{z^5}{6^5} \right) \cdot \frac{\Gamma(\frac{5}{6}) \cdot \Gamma(\frac{1}{2})}{\Gamma(\frac{4}{3})}
\]

\[
= 0.91468 R_0 \sqrt{\left( \frac{P}{P} \right)}
\]
According to (4) $U$ increases without limit as $R$ diminishes. This indefinite increase may be obviated if we introduce — instead of an internal pressure zero or constant, one which increases with sufficient rapidity.

We may suppose such a pressure due to a permanent gas obedient to Boyle's law. Then, if the initial pressure be $Q$, the work of compression is $4\pi Q R_o^3 \log \left( \frac{R_o}{R} \right)$ which is to be subtracted from (3).

Hence,

$$U^2 = \frac{2\rho}{3\rho} \left( \frac{R^3}{R_o^3} - 1 \right) - \frac{2Q}{\rho} \frac{R_o^3}{R^3} \log \frac{R_o}{R}$$

and $U = 0$ when

$$P \left( 1 - z \right) + Q \log z = 0.$$  \hspace{1cm} (8)

$z$ denoting as before the ratio of the volumes.

Whatever be the (+ve) value of $Q$, $U$ again comes to zero before complete collapse, and if $Q > P$ the first movement of the boundary is outwards. The boundary oscillates between two points of which one is the initial.

The following values of $\frac{P}{Q}$ are calculated from (8) —

<table>
<thead>
<tr>
<th>$z$</th>
<th>$\frac{P}{Q}$</th>
<th>$z$</th>
<th>$\frac{P}{Q}$</th>
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</thead>
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<td>$\frac{1}{100}$</td>
<td>4.6517</td>
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<td>$\frac{1}{2}$</td>
<td>1.3863</td>
<td>1000</td>
<td>0.0069</td>
</tr>
</tbody>
</table>
Reverting to the case where the pressure inside the cavity is zero, or at any rate constant, we may proceed to calculate the pressure at any internal point. The general equation of pressure is --

\[
\frac{1}{\rho} \frac{dp}{dr} = - \frac{D}{Dt} \frac{u}{u} = - \frac{du}{dt} - u \frac{du}{dr} \tag{9}
\]

u being a function of r and t, reckoned +ve in the direction of increasing r. As in (1)

\[
u = \frac{ur^2}{r^2}
\]

and

\[
\frac{du}{dt} = \frac{1}{r^2} \frac{d}{dt} \left(UR^2\right).
\]

Also

\[
\frac{d(U^2R)}{dt} = 2R \frac{dR}{dt} U + R^2 \frac{dU}{dt}
\]

\[
= 2RU^2 + R^2 \frac{dU}{dt}
\]

and by (4)

\[
\frac{dU}{dt} = - \frac{P}{\rho} \frac{R_o^3}{R^4}
\]

so that

\[
\frac{d(U^2R)}{dt} = 2RU^2 - \frac{P}{\rho} \frac{R_o^3}{R^2}
\]

Thus by suitably determining the constant of integration we get

\[
\frac{P}{P} - 1 = \frac{R}{3r} \left\{ \frac{R_o^3}{R^3} - 4 \right\} - \frac{R^4}{3r^4} \left\{ \frac{R_o^3}{R^3} - 1 \right\} \tag{10}
\]

At the first moment after release, when R = R_o we have

\[
\frac{P}{P} = P \left(1 - \frac{R}{r}\right) \tag{11}
\]
When $R = r$, i.e. on the boundary, $p = 0$, whatever $R$ may be, in accordance with the assumptions already made.

Initially the maximum $p$ is at infinity, but as the contraction proceeds, this ceases to be true. If we introduce $z$ as before to represent $\frac{R_0^3}{R^3}$, (10) may be written:

$$\frac{p}{P} - 1 = \frac{R}{3r} (2 - 4) - \frac{E^4}{3r^4} (z - 1)$$  \hspace{1cm} (12)

and

$$\frac{dp}{P} = \frac{R}{3r^2} \left\{ \frac{(4z - 4)R^3}{r^3} - (z - 4) \right\}$$  \hspace{1cm} (13)

The maximum value of $p$ occurs when

$$\frac{r^3}{R^3} = \frac{4z - 4}{z - 4}$$  \hspace{1cm} (14)

and then

$$\frac{p}{P} = 1 + \frac{(z-4)R}{4r} = 1 + \frac{(z - 4)^{4/3}}{4^{4/3}(z-1)^{1/3}}$$  \hspace{1cm} (15)

So long as $z$, which always exceeds 1, is less than 4, the greatest value of $p$, viz., $P$ occurs at infinity; but when $z$ is greater than 4 the maximum $p$ occurs at a finite distance given by (14) and is greater than $P$.

As the cavity fills up, $z$ becomes great and (15) approximates to

$$\frac{p}{P} = \frac{z}{4^{4/3}} = \frac{R_0^3}{4^{4/3}R^3}$$  \hspace{1cm} (16)

corresponding to

$$r = 4^{1/3} R = 1.587 R$$  \hspace{1cm} (17)
It appears from (16) that before complete collapse the pressure near the boundary becomes very great, e.g. if \( R = \frac{1}{20} R_0 \), \( p = 1260P \).

This pressure at a relatively moderate distance outside the boundary. At the boundary itself the pressure is zero so long as the motion is free. Mr. Cook considers the pressure here developed when the fluid strikes an absolutely rigid sphere of radius \( R \). If the supposition of incompressibility is still maintained, an infinite pressure momentarily results; but if at this stage we admit compressibility, the instantaneous pressure \( P' \) is finite and is given by the equation

\[
\frac{P'^2}{2\beta'} = \frac{1}{3} \beta U^2 = \frac{P}{3} \left( \frac{R_0^3}{R^3} - 1 \right) \tag{18}
\]

\( \beta' \) being the coefficient of compressibility.

\( P, P' \) and \( \beta' \) may all be expressed in atmospheres. For water \( \beta' = 20,000 \), \( P = 1 \), \( R = \frac{1}{20} R_0 \).

Cook then finds ——

\( P' = 10,300 \) atmospheres = 68 tons/sq.in.

and it would seem that this conclusion is not greatly affected by the neglect of compressibility before impact.

It would seem that for a satisfactory theory compressibility would have to be taken into account at an earlier stage.
APPENDIX II.

Transformation of a circle into a symmetrical aerofoil.

Using the transformation

\[ Z = z + \frac{b^2}{z} \]

the aerofoil form may be developed to a close approximation as follows.

Put \( a = b(1 + e) \)

where \( a \) is the radius of the original circle, and

\( b \) is the radius of the auxiliary circle, and let \( e \) be small. Then for the point \( A \) (Fig. 110)

\[ z = b(1 + 2e) \]

and

\[ Z = b(1 + 2e) + \frac{b^2}{b(1+2e)} \]

Hence \( OA_1 = b(1 + 2e + 1 - 2e + 4e^2 \ldots \) \)

or \( OA_1 = 2b(1 + 2e^2) \) approx.

Also \( OB_1 = 2b \)

and \( A_1B_1 = 4b(1 + e^2) \)

For most purposes \( A_1B_1 = 4b \)

In \( \triangle OPQ \),

Putting \( a \) in terms of \( b \) and retaining only the terms in the first power of \( e \),

\[ r = b \{ 1 + e(1 + \cos \theta) \} \]
from which it follows that

\[ x = (r + \frac{b^2}{r}) \cos \theta = 2b \cos \theta \]

\[ y = (r - \frac{b^2}{r}) \sin \theta = 2b e (1+\cos \theta)\sin \theta \]

from which the form may be constructed. The table below gives the calculated values of \( x \) and \( y \).

\[
\begin{array}{cccccccc}
\theta & \cos \theta & 1.30 & 1 + \cos \theta & \sin \theta & (1 + \cos \theta) \cdot 0.195(1 + \cos \theta) \\
0 & 1.000 & 1.300 & 2.000 & 0 & 0 & 0 \\
10 & 0.985 & 1.280 & 1.985 & 0.174 & 0.345 & 0.067 \\
20 & 0.940 & 1.222 & 1.940 & 0.342 & 0.664 & 0.129 \\
30 & 0.866 & 1.125 & 1.866 & 0.500 & 0.933 & 0.182 \\
40 & 0.766 & 0.996 & 1.766 & 0.643 & 1.135 & 0.221 \\
50 & 0.643 & 0.836 & 1.643 & 0.766 & 1.258 & 0.245 \\
60 & 0.500 & 0.650 & 1.500 & 0.866 & 1.298 & 0.253 \\
70 & 0.342 & 0.445 & 1.342 & 0.940 & 1.262 & 0.246 \\
80 & 0.174 & 0.226 & 1.174 & 0.985 & 1.156 & 0.225 \\
90 & 0 & 0 & 1.000 & 1.000 & 1.000 & 0.195 \\
100 & -0.174 & -0.226 & 0.826 & 0.985 & 0.813 & 0.159 \\
110 & -0.342 & -0.445 & 0.658 & 0.940 & 0.618 & 0.120 \\
120 & -0.500 & -0.650 & 0.500 & 0.866 & 0.433 & 0.084 \\
130 & -0.643 & -0.836 & 0.357 & 0.766 & 0.273 & 0.053 \\
140 & -0.766 & -0.996 & 0.234 & 0.643 & 0.150 & 0.029 \\
150 & -0.866 & -1.125 & 0.134 & 0.500 & 0.067 & 0.013 \\
160 & -0.940 & -1.222 & 0.060 & 0.342 & 0.021 & 0.004 \\
170 & -0.985 & -1.280 & 0.015 & 0.174 & 0.003 & 0.001 \\
180 & -1.000 & -1.500 & 0 & 0 & 0 & 0 \\
\end{array}
\]
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