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SUGGESTED DESIGN CHANGES FOR A CENTRIFUGAL PUMP IMPELLER HANDLING DREDGED MUD

A report for

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by

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A C K N O W L E D G E M E N T S

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INTRODUCTION

There is an ever increasing use of centrifugal pumps in applications which require the pumping of liquids containing solids in suspension, and liquids other than water. A dredge pump is an example of the first application and an oil pump of the second. With the increasing use of centrifugal pumps for applications of this type, it becomes important to be able to predict pump characteristics and create efficient designs.

It is the object of this report to review one phase of the available research work carried out on the subject of pumping solids suspended in liquids. Design changes will be suggested in an attempt to improve the performance of the impeller used in the pump on the United States Corps of Engineers' hopper dredge, the "Essayons". The changes in impeller design will hold only for this particular pump when handling a mudlike material, predominately silt, suspended in water. A research project, sponsored by the Corps of Engineers, is currently under way in Fritz Engineering Laboratory at Lehigh University. This project consists of the detailed study of a one-eighth scale model of the centrifugal dredge pump referred to above. Some information gained from this research will be presented in this report. This project will undoubtedly furnish much valuable information which will contribute to the knowledge about the design and characteristics of centrifugal dredge pumps.

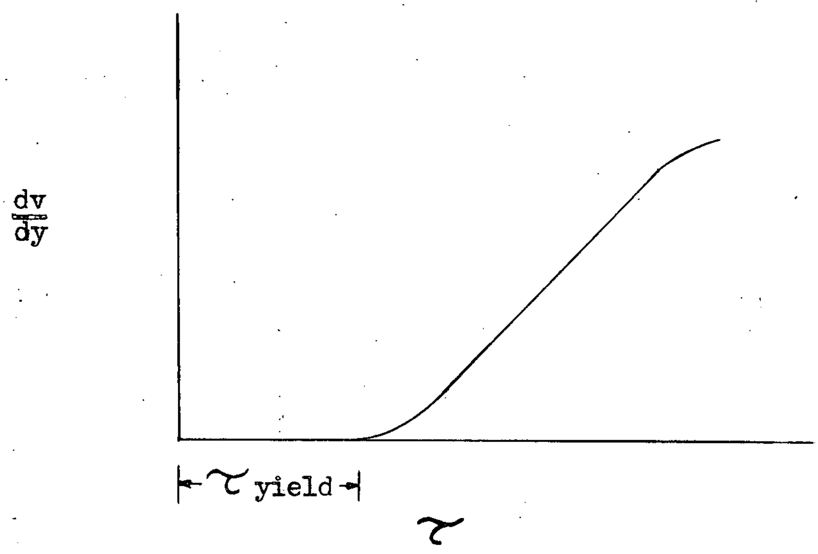
GENERAL CONSIDERATIONS

In the design of a centrifugal dredge pump it is necessary to consider more factors than those customarily considered in the design of a centrifugal pump for water. The nature of the dredging operation is such that sufficient clearances must be provided through the pump so that occasional gravel, rocks, and debris may be passed through the pump without jamming (1)*. This requirement means that there is a practical limit to the number of vanes which the impeller may contain and also that clearances in parts of the pump must be made in excess of those which highest performance would dictate. An example of the above is the clearance at the cut-water between the impeller and the volute. The increased wear and abrasion due to particles in suspension require the use of special materials. Provision must be made for easy access to parts of the pump for maintenance and renewal of components.

The type of material pumped has been shown to have a great effect on the performance of a centrifugal pump. The impeller of the centrifugal dredge pump under study handles a mudlike fluid common to dredging operations on the East coast of the United States. This mud, composed largely of silt with some organic matter, possesses some very interesting properties (2). Beside its specific gravity which is greater than that of water depending on the weight of solids in suspension, the mixture has a highly variable viscosity. The viscosity of the material is dependent on the concentration, temperature, past history and the rate of shearing stress after an initial yield value has been reached. Figure 1 shows the variation of viscosity with rate of shearing stress for a certain concentration. This behavior is characteristic of a so-called Bingham body (3). The material exhibits

* Numbers in parenthesis refer to References on page 19.

$\frac{dv}{dy}$ vs τ FOR A GIVEN CONCENTRATION AND TEMPERATURE



V - velocity

τyield - shear stress at yield point

y - distance measured perpendicularly to the direction of motion

FIGURE I

both plastic and viscous properties. It is like an ideal plastic in that it flows when a given yield shearing stress, τ_y, is reached. It is unlike a plastic in that once flow starts, it is retarded by an increased resistance to shear. If the shearing stress, τ, is such that it is less than τ_y, the material will deform elastically but will not move. Once $\tau=\tau_y$, the material will move but it will not flow until τ is greater than τ_y. The diagram in Figure 1 holds for one concentration only, a similar diagram with different values being required for each concentration.

The effects of viscosity are important considerations in any pump analysis or design (4). The distribution of velocity of flow through any passage is dependent on viscosity; any increase in viscosity will produce a corresponding reduction in the effectiveness of the area available. Loss of head through the passage also increases with increased viscosity. The power absorbed by the pump is increased with increase in viscosity, because of the greater resistance of the fluid to rotation of the impeller.

There is considerably more difficulty in working with a material which has a viscosity dependent on a number of variables than there is in working with a fluid such as water which possesses a viscosity which is dependent on temperature only. Thus the type of application and the material to be handled impose difficulties on design and analysis which must be kept in mind.

PREVIOUS RESEARCH

Research in the field of pumping solids in suspension is meager. For the reasons stated previously, it is more difficult to design a centrifugal dredge pump than it is to design a centrifugal water pump. Even this latter field relies heavily on empirical formulae and plots of various parameters derived from experiment and successful past performance. The tendency now seems to pursue the theory (fully realizing the assumptions made) as far as possible and then to introduce corrections from research or past experience to come up with the final design. Strictly mathematical formulae have been developed (5, 6, 7) and are good stepping stones to design, but these theories are as yet limited to ideal fluids (i.e. nonviscous) and a pump or compressor with straight radial vanes.

Previous investigations have established some basic concepts about the pumping of solid-liquid mixtures. It has been definitely established that there is a difference in pump characteristics for clear water alone as opposed to the pumping of a sand-water mixture. (8)

Gregory (9) in a paper on the pumping of a clay slurry through a 4-inch pipeline drew the following conclusions in reference to the pump characteristics:

(a) the head developed at a given capacity decreased as the concentration of the solid material in suspension increased.

(b) the required power input at a given capacity increased as the concentration of the solid material in suspension increased.

(c) the efficiency at a given capacity decreased as the concentration of solid material in suspension increased.

The work of Fairbank (10) on the pumping of sand in water resulted in prediction methods for sand-water mixtures which gave results within about 5 per cent of experimental values in the normal operating range. A more involved refinement gave results within about 3 per cent for the same range. Fairbank's conclusions concerning the pumping of sand-water mixtures are as follows:

(a) at a given capacity the head developed by a centrifugal pump handling material in suspension, is in general less than that developed for water alone.

(b) the drop in the constant speed head capacity characteristics varies not only as the concentration but also as the particle size of the material in suspension.

(c) the fall velocity of the suspended material is the most important property in predicting the effect of the material on the pump performance.

(d) the effect on the pump characteristics of very fine particles in suspension, such as colloids, is of a different nature than that of a true suspension.

(e) the power input to a centrifugal pump varies directly with the apparent specific gravity of the suspension being pumped.

(f) the capacity for maximum efficiency of a centrifugal pump remains constant for all concentrations and sizes of suspended materials.

(g) the ordinary affinity relationships of centrifugal pumps are valid within small ranges of speed when pumping material in suspension.

Preliminary examination of data from the dredge pump tests carried out at Lehigh University confirm a number of the conclusions found above; namely these are:

(a) the required power input at a given capacity and speed increase almost linearly as the concentration of solid material in suspension increased. This held well for capacities from 20% to 120% of normal capacity (the capacity at maximum efficiency).

(b) the head developed in feet of fluid being pumped at a given capacity and speed decreased as the concentration of the solid material in suspension increased.

(c) the efficiency at a given capacity and speed generally decreased as the concentration of solid material in suspension increased. The efficiency did not decrease from that obtained with water until a certain concentration was reached and then the efficiency dropped off.

The above conclusions are valid only for the particular pump and fluid used in the test.

CONSIDERATIONS LEADING TO CHANGES IN IMPELLER DESIGN

General

With the use of the above experimental work on the pumping of water-solid mixtures and known centrifugal water pump information, a qualitative analysis will be made on the dredge pump impeller with the idea of increasing the efficiency of the pump when handling the mud mentioned earlier. Theoretical considerations are based largely on the work of Stepanoff (11) and Shepherd (12).

The impeller of a centrifugal pump is only one part of the machine and a complete study entails an investigation of the impeller's relationship to all of the other parts of the pump. Therefore, an investigation is in order into the manner in which the flow approaches the pump, how it enters the impeller, how it moves through the impeller, how it leaves the impeller, and finally how it travels out of the pump.

Approach conditions

Ideal conditions for approach exist when a sufficient length of straight pipe without valves or other disturbances precedes the entrance to the pump. Well upstream a normal velocity distribution can be expected, but as the distance to the pump decreases a phenomenon known as prerotation may develop.

Prerotation is the condition that exists in the approach pipe to a centrifugal pump when the flow is moving both axially toward the pump and rotating about the longitudinal axis of the pipe. Peck (13) has carried out investigations on the subject of prerotation and has obtained the following information from Pitot tube traverses at the suction flange of a pump handling water. At shut-off head conditions, a forced vortex was created in the suction pipe. The total head and pressure head curves were obtained and the differences gave the velocity head from which the curves of axial flow and circumferential flow components were plotted. Curves of this type are shown in Fig. II.

CONDITIONS IN SUCTION PIPE (FROM REFERENCE 13)

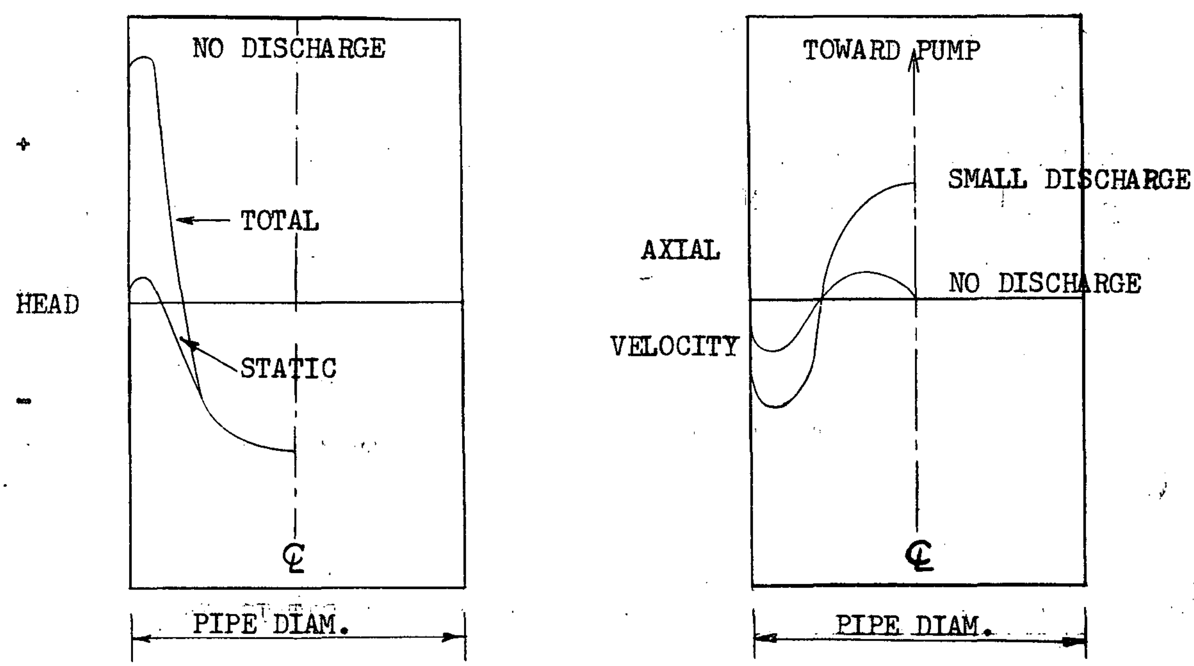


FIGURE II

This showed an axial flow away from the pump through a narrow annular ring extending from the pipe wall. For small discharges the forced vortex was confined to a somewhat larger annular ring than previously, and while there was still a small component of flow away from the pump, axial flow toward the pump occurred in the center portion of the pipe. The extent of the prerotation, with respect to distance from the pump, was found to decrease as the flow increased. The swirl gradually disappeared owing to friction between the pipe wall and the water.

Stepanoff (11) explains prerotation as follows. Referring to Fig.III, at section 1 sufficiently distant from the pump, pressure p_1 is uniform across the section and a normal pipe velocity distribution prevails. At section 2, near the pump the pressure p_2, as measured at the pipe wall is

PRESSURE DISTRIBUTIONS ALONG APPROACH TO PUMP

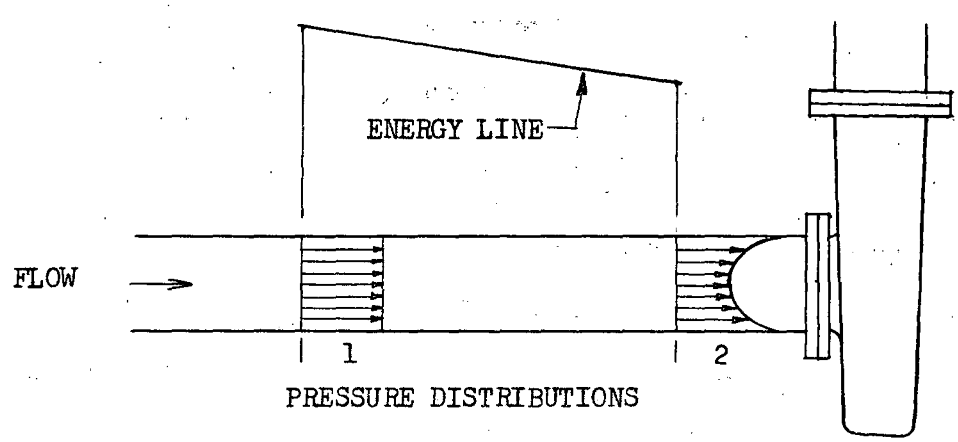


FIGURE III

higher than at section 1. This has been found to be true experimentally when prerotation is present (13). Since the energy gradient must decrease from section 1 to section 2 if there is to be flow, the higher pressure at 2, at the pipe wall, can come about only at the expense of the energy level of the center portion of fluid. A paraboloid of pressure distribution is developed at section 2 with the pressure at the periphery higher and the pressure at the center lower than the pressure existing at section 1. Pipe wall pressure taps at a location such as section 2, will indicate too high a mean pressure and thus often introduce error into experiments. The absolute velocities at the periphery of section 2 are higher than those in the middle as a result of the addition of a tangential component due to the rotation of the stream.

Prerotation is caused by the tendency of the fluid to follow a path of least resistance on its way to enter the impeller channels. Prerotation is most evident at flows less than the design capacity and practically disappear at that point. The angle at which the fluid enters the impeller channel is influenced by prerotation. Since the geometry of a given impeller is fixed, the entrance angle for the impeller vanes is calculated for the design capacity with no prerotation considered. It is customary to increase the design flow by a small percentage when calculating the entrance angle as this allows for the unavoidable leakage losses. It is evident that at flows other than design, prerotation will come into being and the effect will be detrimental to pump performance. When a pump must be operated at some flow other than the flow at which maximum efficiency occurs, there is one solution for overcoming the undesirable effect of prerotation. This remedy consists of placing guide vanes in the approach to the pump at such an angle as to make the fluid conform to the entrance angle of the impeller. Care must be taken to see that the guide vanes are so constructed as not to cause separation, cavitation, or high losses. The nature of dredge pump application makes this method impractical.

Inducing section

The purpose of the inducing section of the pump is to accept the incoming fluid from the suction pipe in the correct manner and to turn it so that its relative velocity is along the axis of the impeller channels (12). This means that the flow must be turned through ninety degrees for the case of the radial impeller. This turning of the fluid causes losses and disordered flow. The transition must be made as gradually as possible with no irregularities so that the flow is delivered to the impeller channels as ideally as possible. Low suction velocities and generously proportioned suction sections are recommended for pumps handling high viscosity liquids (4). The factor of manufacturing economy enters here to set a practical length which may be used for this turning transition. According to N A C A investigations (14), the inducing section has a marked effect on performance. An inducing section of comparatively large axial length gave very good results since it provided a much gentler curve for the transition from tangential to axial flow.

Flow at impeller entrance

It is important to limit the relative velocity at the inlet of the pump, since a high relative velocity could lead to cavitation. Assuming a uniform axial component of velocity at the eye, or inducing section, the critical region is located at the eye tip. This is where the impeller speed is highest for the inducing section, and hence the relative entrance velocity is also great here. Therefore, flow conditions should be checked at this point. The combination of variables, determining relative velocity at this point may be arranged to result in a minumum value. A high relative velocity can result from either of the two following situations.

1). a large eye area, resulting in a small axial entrance velocity, combined with a high impeller speed.

2). a small eye area, resulting in a high axial entrance velocity, combined with a low impeller speed.

Using the following notation, a formula for minimum relative entrance velocity may be developed;

V_{r1} = relative velocity at entrance in feet per second

$U_1 = \frac{\pi N d_1}{60}$ = tangential velocity at the eye tip in feet per second

N = impeller speed in RPM

$V_1 = \frac{4Q}{\pi d_1^2}$ = radial velocity in feet per second of through flow at entrance assuming no prerotation

d_1 = diameter of eye in feet

Q = flow in cubic feet per second

combining U and V ; $V_{r1} = \sqrt{V_1^2 + U_1^2}$ or $V_{r1} = \left[\left(\frac{4Q}{\pi d_1^2}\right)^2 + \left(\frac{\pi d_1 N}{60}\right)^2\right]^{1/2}$

The effect of any one variable on V_{r1} may be seen by holding all other variables constant. For instance with Q and N fixed the variation in V_{r1} with d_1 can be found. The minimum value of d_1 can be obtained by differentiation or trial and error. Fig. IV shows the variation for this case and it will be noted that for a certain value of d_1 there is a minimum value of V_{r1} which will tend to prevent cavitation.

Flow in the Impeller Channel

Due to viscosity, turbulence, and separation the velocities in an actual centrifugal pump impeller are seldom uniform over a given section. The turns in the impeller channel approach and the impeller profile add to the velocity distortion. In radial flow and mixed flow impellers, the fluid must make nearly a full 90° turn before it is acted upon by the vanes.

V_{r1} vs d_1 WITH Q AND N FIXED

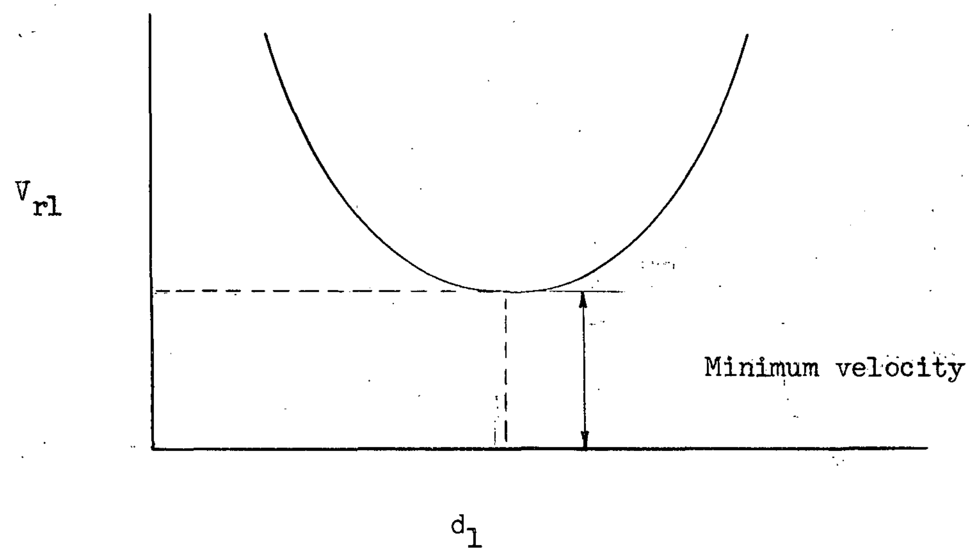


FIGURE IV

In an established flow, whether rotating or straight as in open channel flow, a body must move faster than the established velocity of flow in order to exert any force on the fluid flowing in the same direction. Thus, an impeller vane must move faster than the fluid in order to transmit energy from the impeller vane to the fluid. This means that the pressure on the leading face of the vane should be higher than the pressure on the trailing face of the vane. Due to the lower pressure on the trailing face, there will be a higher velocity there, and a velocity variation across the channel. The result of this is that relative to the vane the fluid leaves the vane tangentially only at the high pressure or leading side of the trailing edge. The fluid has a circumferential component relative to the vane across the channel from one leading face to the trailing face of the adjacent vane, with the result that the fluid is discharged from the impeller at a mean angle relative to the impeller which is less than the vane angle. The absolute discharge velocity is less than that assumed by using the vane angle itself;

this deviation being called the slip of the impeller. Figure V illustrates the difference between the case of slip and no slip. It is important to realize that changing the vane discharge angle from B_2 to B_2' will only mean that the fluid will again lag behind the vane and discharge at some smaller angle B_2'', less than B_2'. The net result of the non-uniform velocity and the slip is to reduce the theoretical head based on the simple ideal velocity diagram. It may be shown (11) that the head produced by a varying velocity distribution is less than that produced by a uniform velocity, given the same rate of flow.

The occurrence of slip in an actual pump has been studied by Peck (13). After information from Pitot tube traverses was analyzed, a difference was noted between the mean absolute velocity discharge angle and the calculated angle assuming that the relative velocity leaves the impeller parallel to the actual vane angle.

CHANGE IN DISCHARGE VELOCITY DIAGRAM DUE TO SLIP

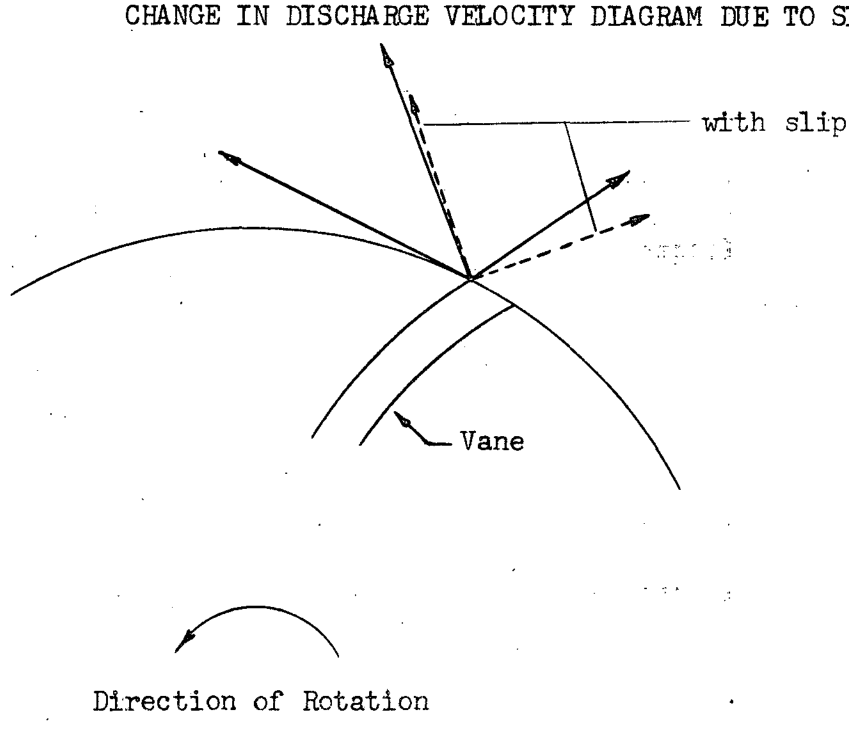


FIGURE V

Actual observations of flow in impeller channels have been made using a pump constructed of a transparent material (15). The flow was seen to be far from the ideal at most capacities except those at the design rate of flow where it was often suprisingly good. Fisher's investigations into flow in impeller channels revealed a large area of separation from the trailing face of the vane. This dead water region formed a considerable portion of the passage area at low rates of flow. Reverse flow in these areas was also present in some instances. Similar work by Binder and Knapp (16) showed almost uniform absolute discharge velocity across the width of the impeller passage at normal capacity with small gradients existing at other capacities.

The Number and Shape of the Impeller Vanes

Theoretically an infinite number of vanes is required to produce the head indicated by the ideal velocity triangle. In an actual pump the head and efficiency increase with the number of vanes (17) until the additional losses produced by the larger number of vanes reaches some point where the efficiency is a maximum. The available flow area is also reduced due to the finite thickness of the vanes. This is especially critical at the inlet where space is limited and the problem of cavitation may occur. Friction losses in a duct are a minimum for the largest hydraulic radius. For a quadrangular passage this is best suited by a square cross-section. For the impeller under study, the present vane spacing to passage width ratio is nearly 1.7 at the radius of the impeller corresponding to mid-distance along the vane. With the addition of another vane, making a total of six, this ratio would be 1.4. Of course, it must be kept in mind that the flow conditions in a duct are not strictly those occurring in the impeller channels as some of the investigations mentioned above have shown. It is suggested that if it is practical from the standpoint of sufficient clearances (for passage of debris, rocks, etc.) that an additional vane be added. Only experiment will determine whether or not the additional vane will increase efficiency.

An incompressible fluid with radial and rotational motion will ideally follow a logarithmic spiral path in which the streamlines (imaginary lines every point on which is tangent to the velocity vector at that point) at any point have a constant inclination with the tangent to the radius at that point (12). As this has a certain rational basis and as a logarithmic spiral is geometrically simple, pump vanes are often designed in this manner. Manufacturing cost and simplicity are also factors to be considered. Sometimes pump vanes are curves which are a portion of a simple circular arc, although it is known that this does not give the best results. The circular arc does not give as smooth a flow path as the logarithmic spiral. Since the present dredge pump impeller has vanes defined by a circular arc, it is suggested that the vane profile be changed to that of a spiral.

The present vane tips are blunt as shown in Fig. VI. This will tend to cause disturbances in the volute. This effect may be partially or entirely eliminated by tapering the vanes as shown. (13) A number of different vane shapes and profiles have been tried by researchers in order to improve pump efficiency. One of these trials consisted of trying to utilize only the active flow part of the impeller passage. Therefore, the areas of dead water mentioned previously were removed by shaping the vanes to occupy this area. "Club-headed" vanes, the width of which were greatly increased at the outlet to reduce the water passage to that expected to be occupied by useful flow, did not produce the improvements which might be expected in efficiency and power at small flows (18).

ROUNDING OFF OF TRAILING EDGE OF BLUNT VANE TIPS

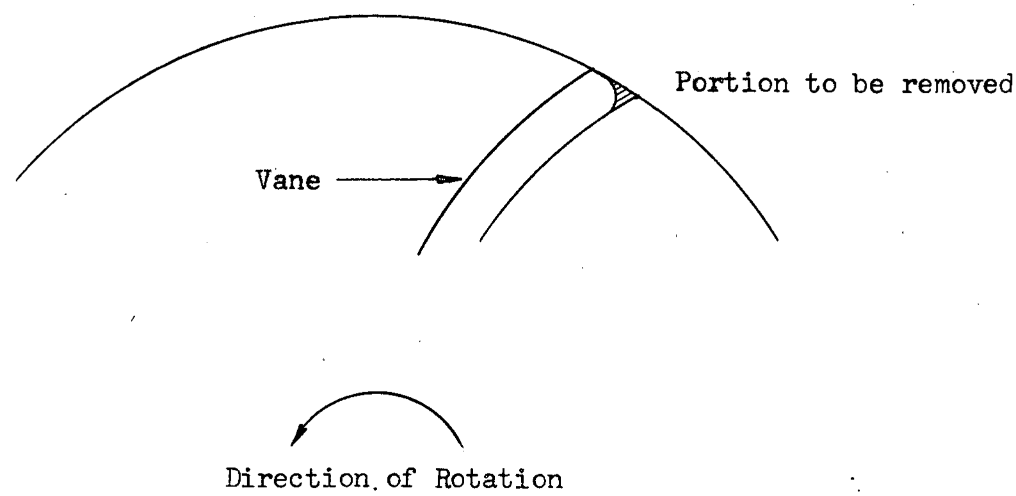


FIGURE VI

The discharge angle is one of the most important aspects of impeller design. It has been explained earlier that a fluid does not leave the vane tangent to the surface of the vane. Stepanoff (11) suggests 22 1/2° as the best vane exit angle for water. This angle is the acute angle formed by a line tangent to the vane at its end and a line tangent to the impeller periphery at the point. Angles from 17 1/2 ° to 27 1/2° are the usual limits for centrifugal water pump vane exit angles. For pumping viscous liquids an increase in the discharge angle up to 60° is suggested. The discharge angle has an effect on the head-capacity characteristic curve of the pump. Studies were made (19) on a water pump keeping every variable except vane discharge angle and vane profile constant. Vane profile had to change to result in the various discharge angles tested. Moderate values of the angle, 20° to 30°, gave relatively flat head curves with good efficiencies. As the angle was increased, head produced increased a little and efficiency fell off a little. The head curve tended to be more rounded with some maximum value. This would not be desirable if pumps of this type were to operate in parallel. The discharge angle of the present pump is 35°. Tests on the model have shown the head capacity curves to be rising to a maximum value and then falling off. A slight decrease in angle, say to 30°, may tend to produce a flater curve and still not affect the efficiency.

Another factor related to the impeller which influences the efficiency of a pump is the clearance between the impeller and the cut-water or volute tongue. An excessive gap here leads to a reduction of efficiency (11). The gap on the present pump is quite large and if practical considerations would allow this distance to be decreased, an increase in efficiency might be expected. However, the clearance should be at least twice that recommended for a water pump since friction effects will be more pronounced with a liquid of high viscosity.

SUMMARY

On the basis of the factors discussed above, the following changes in the design of the dredge pump impeller are suggested.

1. Change the profile of the vanes from a circular arc to a spiral.

2. Increase the number of vanes from five (5) to six (6) if practical considerations will allow it.

3. Decrease the vane exit angle from 35° to a value of 33° to 30°.

4. Round off the blunt trailing edges of the vanes with a small radius fillet.

5. Decrease the impeller-volute tongue clearance to a value at least twice that recommended for centrifugal water pumps if practical considerations will allow it.

These suggested changes are made only for the model dredge pump studied when handling the mud-like fluid mentioned previously. In this study impeller changes were made on the basis that no other components of the pump were to be changed. Additional factors would have to be considered if for instance, the volute were to be changed. Even with these limitations, many of the factors discussed should be applicable to other situations where centrifugal pumps are used to handle liquids containing suspended solids.

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