Civil Engineering Department
Fritz Engineering Laboratory
Hydraulics Division

PROJECT REPORT NO. 38

SUCTION DREDGING
LITERATURE SURVEY

Hugh D. Murphy
John B. Herbich

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CIVIL ENGINEERING DEPARTMENT
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HYDRAULICS DIVISION

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SUCTION DREDGING
LITERATURE SURVEY

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Prepared for
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P R E F A C E

A contract between Lehigh University, Institute of Research, Fritz Engineering Laboratory, Hydraulics Division and the Ellicott Machine Corporation provided for a survey of literature on dredging in general and dredge pumps in particular.

The survey was written under the direction and supervision of Professor John B. Herbich, Chairman of the Hydraulics Division at the Fritz Engineering Laboratory. Acknowledgments are also due to Professor Herbich for his assistance in providing source materials for the survey.

Professor W. J. Eney is the Head of the Civil Engineering Department and Fritz Engineering Laboratory and Professor L. S. Beedle is the Director of Fritz Engineering Laboratory.
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This report is a brief review and summary of selected literature pertaining to equipment and methods associated with dredging practice and laboratory studies of dredge pumps. It consists of four parts.

1. Summary or discussion section

2. Selected Abstracts

3. Annotated Bibliography

4. Bibliography

The discussion section consists of two parts. Part one discusses dredging equipment and dredging in general. Part two discusses dredge pumps.
INTRODUCTION

Dredging is the process of excavating and removing unwanted material from the bottom of harbors and waterways. Dredges, especially the hydraulically operating types, are admirably suited for this purpose. As a means of earthmoving, the hydraulic dredge has never been surpassed at sea. Probably no other machine invented by man performs this task as efficiently and economically as the hydraulic dredge.

Considering the important role which dredging plays in keeping our harbors and waterways navigable, it is surprising that very little research has been done on this subject until recently. The purpose of this report is to present summaries of that work which has been done and to indicate the scope of studies now in progress.

Due to the great importance of the dredge pump in hydraulic dredges and the special problems which this type of pump represent, dredge pumps will be discussed in a separate part distinct from the body of the report.

Thus the discussion consists of two parts: dredging vessels and dredge pumps. Part 1 describes the different types of dredging vessels and their related equipment; such as suction and discharge assemblages, distribution systems, and hoisting and control machinery. Part 2 describes the dredge pump--its function, design and operation.
I PART 1. DREDGING VESSELS

1.1 BASIC TYPES

All dredges, regardless of type, can be classified into two main types: mechanically operating and hydraulically operating.

1.1.1 Mechanical Dredges

Due to their simplicity and analogy with land based excavating machines, mechanical dredges were the first to be developed. Mechanical dredges can be further classified into the grapple dredge, the dipper dredge, and the bucket-ladder dredge.

The grapple dredge consists of a derrick mounted on a barge and equipped with a "clamshell" bucket. Its best usage is obtained in very soft underwater deposits.

The dipper dredge is the floating counterpart of the more familiar land based mechanically excavating shovel. Due to its great leverage and "crowding" action it works best in hard compact material or rock.

The bucket-ladder dredge consists essentially of an endless chain of buckets, the top of the chain being thrust into the underwater deposit to be dredged so that each bucket digs its own load and carries it to the surface. Since the work cycle is continuous, bucket-ladder dredges are more efficient than either the grapple or dipper dredge. Bucket-ladder dredges are particularly useful to sand and gravel suppliers since the
end of the bucket-ladder can be terminated high above the supporting barge and the buckets made to discharge their contents onto vibrating screens. Thus, the different material sizes may be separated and stored on the barge, all by gravity.

Mechanical dredges are all characterized by their inability to transport the dredged materials for long distances; their lack of self-propulsion; and their relatively low production. Their chief advantage lies in their ability to operate in restricted locations such as docks and jetties.

1.1.2 Hydraulic Dredges

Hydraulic dredges, which are the primary concern of this report, are self-contained units and handle both phases of the dredging process. That is, they not only dig the material; but also dispose of it either by pumping the material through a floating pipeline to a spoil area or by storing it in hoppers which can be subsequently emptied over the spoil area. It can be seen that hydraulic dredges are more efficient, more versatile, and more economical to operate due to this continuous, self-contained digging and disposal principle of operation.

With a hydraulic dredge, the material to be removed is first loosened and mixed with water by cutter heads or by agitation with water jets, and then pumped as a fluid.

The three basic units in a hydraulic dredge are the dredge pumps, the agitating machinery, and the hoisting and hauling equipment. The latter is used primarily to raise and lower the cutter and suction dragheads. Due to its relatively great importance and complexity, the dredge
pump will be discussed in a separate part of this report.

Hydraulically operating dredges can be classified into three basic types -- The Dustpan Dredge, the Hydraulic Pipeline Cutterhead Dredge, and the Self-Propelled Hopper Dredge.

1.1.2.1 The Dustpan Dredge - is named because its suction head resembles a large vacuum cleaner or dustpan. The dustpan dredge is a hydraulic, plain suction, self-propelled dredge. It consists essentially of a dredge pump which draws in a mixture of water and dredged materials through the suction head which is lowered by winches to the face of the deposit to be removed. The suction head, which is about as wide as the hull of the dredge, is outfitted with high velocity water jets for agitating and mixing the material. After sucking the mixture to the surface, the dredge pumps it to a spoil area, either at sea or shore, through a floating pipeline. Due to its lack of a cutterhead, which loosens up hard compact materials, the dustpan dredge is suited mostly for high volume, soft material dredging.

A particular use for which this type is well suited is in conjunction with a hopper dredge. The hopper dredge makes its cycle, returning to empty its hoppers next to a dustpan dredge. Next,
the dustpan dredge sucks up the deposited material and pumps it ashore to a spoil area.

1.1.2.2 The Hydraulic Pipeline Cutterhead Dredge (Fig. 1)

This is probably the most well-known dredging vessel as well as the most efficient and versatile. It differs from the dustpan dredge in that it is equipped with a rotating cutter apparatus surrounding the intake end of the suction pipe. These dredges can efficiently dig and pump all types of alluvial materials, and also compacted deposits such as clay and hardpan. The larger and more powerful machines are used to dredge rocklike formations such as coral and the softer type of basalt and limestone without blasting. Some of these dredges have been known to excavate and transport boulders in sizes up to 30 inches in diameter.

This dredge is generally equipped with two stern spuds. These spuds are used to advance the dredge into the cut or excavating area. A well-designed 30-inch dredge (size is given by the diameter of the discharge pipe) with 5000 to 8000 hp. on the pump and 2000 hp. on the cutter will pump 2000 to 4500 cubic yards per hour in soft material, and 200 to 2000 cubic yards per hour in soft to medium hard rock through pipeline lengths up to 15,000 ft.
HOW A HYDRAULIC PIPELINE CUTTERHEAD OPERATES

(Solids are freed by cutterhead, suspended in water, and drawn up through suction pipe to pump. From the pump they are forced through a pipe line to fill or spoil areas. Operation is continuous.)

Courtesy of Ellicott Machine Corporation
1.1.2.3 **The Self-Propelled Hopper Dredge** - A hopper dredge of the seagoing type has the molded hull and lines of an ocean vessel and functions in a manner similar to that of the plain suction type dredge. The bottom material is raised by dredge pumps through dragarms which are connected to the ship by trunnions. The lower ends of the dragarm have suction dragheads for contact with the bottom material. The dragarms are raised and lowered by hoisting tackle and winches. The pumps lift the mixture through the dragheads to the surface where it is discharged into hoppers. As pumping continues, the solid particles settle in the hoppers while the excess water passes overboard through overflow troughs. After the hoppers have been filled, the dragarms are raised and the dredge proceeds at full speed to the spoil area or disposal site and empties the loaded hoppers through bottom doors. The door then closes and the dredge returns to the dredging area to continue the cycle.

American dredges operate with dragarms trailing at a ground speed of 2 to 3 miles per hour. Hopper dredges range in size from approximately 180 to 550 feet in length and have hopper capacities between 500 and 8,000 cubic yards. They are equipped with twin propellers and twin rudders to provide the required
maneuverability. Dredging depths vary from 10 to over 70 ft. (79)

Dredges of this type are necessary for maintenance work and improvement in exposed harbors and waterways where traffic and operating conditions rule out the use of stationary dredges. These dredges can also be equipped for "agitation dredging" whereby soft or free flowing alluvial materials are sucked up and then discharged through a suspended discharge pipe directly overboard without storing in the hoppers. The material is then carried out of the dredging area by currents and stream action.

One of the largest hopper dredges, the ESSAYONS (shown in Fig. 2), was built by the U. S. Army Corps of Engineers for dredging along the eastern seaboard. This seagoing dredge has two 36-inch suction pipes, twin dragheads, and a hopper capacity of 8000 cubic yards. It is 525 feet long and carries a crew of 120. Twin screws and high power give it excellent maneuverability and a 16-knot loaded speed. Twin 1850 hp. centrifugal pumps dredge up to a depth of 70 feet and handle a million cubic yards a month; up to a million and a half when operating with shorter hauls.
FIG. 2 The Hopper Dredge ESSAYONS (Courtesy of the Corps of Engineers, U. S. Army)
1.2 AUXILIARY EQUIPMENT

1.2.1 Suction and Discharge Piping

1.2.1.1 Friction Losses - Laboratory experiments and theory indicate that when the flow in dredge pipelines is fully turbulent, the head loss, expressed in terms of a column of the fluid being pumped, will be the same for any mixture of water and solids as it is for clear water.\(^{(99)}\)

However, flow in dredge pipelines seldom, if ever, complies with the conditions of complete turbulence.\(^{(99)}\) Further, since the density of the mixture varies from the top to the bottom of the pipe, and from point to point along the line, attempts to apply theory to operating conditions are not convincing. However, by resorting to the concept of an average coefficient of friction and an average density of mixture pumped, computations may be made which have checked well with observed results. Vaughn\(^{(99)}\) gives the loss of head as

\[
h = K \frac{V^{1.9}}{D^{1.1}} \tag{1}\]

where \(h\) is the loss of head in feet of mixture being pumped per 1000 feet of pipeline, \(V\) is the velocity in feet per second, \(D\) is the inside diameter of the
pipe in feet, and $K$ is a constant having a value of 0.4 for dredge pipelines containing the usual number of elbows and flexible connections.

Gregory(28) has shown that when pumping clay slurry there exists a critical velocity which varies with the concentration of solid particles in the mixture. For velocities lower than this critical velocity it was found that the head loss does not vary appreciably with the velocity.

1.2.1.2 Connections - Within recent years the use of pontoon line ball joints which eliminate the usual bolted connections have come into great use in the dredging field. Besides permitting greater flexibility in the pipeline, the use of ball joints cuts the time consumed in connecting almost in half.

In addition, pontoon line ball joints permit the line to lie lower in the water, thus making it more stable and less affected by wave action.

1.2.1.3 Booster Pumps - To determine the effects of booster pumps on the capacity of suction dredges the National Bureau of Standards(61) has run tests on a sand-water mixture. It was found that, in general, the booster increased the flow, increased the concentration of sand in the mixture, and decreased the vacuum at the
dredge pump as long as it operates above a certain critical speed. Below the critical speed, the booster acts as an obstacle to flow.

1.2.2 Suction Heads

Suction heads are divided into two main types - cutterheads and dragheads.

1.2.2.1 Cutterheads (Fig. 3) - Cutterheads are attached to the end of the suction pipe and consist essentially of a rotating set of curved teeth or blades. Usually the blades have a spiral shape and are removable for easy maintenance and replacement. High alloy steels are used in their manufacture to prevent excessive wear and abrasion.

Although the horsepower consumed by the cutter can be considerable (up to 2,500 hp. on a 30-inch dredge), its use is well worthwhile when hard, compact material is to be dredged.

One of the more important developments in dredging has been the introduction of the direct suction pipe-cutter drive which eliminates the costly cutter shaft and its maintenance.

1.2.2.2 Dragheads (Fig. 4) - Dragheads differ from the cutterhead in that they have no rotating blades, although they may have teeth attached to the bottom of the drag which partially loosen up the material to be removed.
FIG. 3 Basket-type Cutterhead with Replaceable Serrated Blades. (Courtesy of Ellicott Machine Corporation)

FIG. 4 California type Draghead for Use on Hopper Dredges (Courtesy of the Corps of Engineers, U. S. Army)
The bottom of the draghead consists of a rectangular grate having openings which are of such size so that they reject objects which will not pass through the pump, but which are large enough to prevent clogging of the openings. It has been found that for maximum efficiency, the area of the grate should be approximately three times the area of the suction pipe opening. (79)

Dragheads are primarily used on hopper dredges and can be classified into the following types:

(a) **Ambrose Draghead** - The Ambrose draghead is a general purpose drag employed for dredging mud, silt, clay, gravel, loose sand, and stones. It is non-adjustable and adapters must be installed at the after end of the dragarm pipe so that the grate contact of the drag will be suitable for bottoms of different depths.

(b) **California Draghead** - This type of drag was developed primarily for dredging in sand. It has a movable self-adjusting after body, supported on hinges in such a manner that regardless of the angle of the dragarm, the flat grate of the after body will automatically adjust itself to provide overall contact with the bottom.
(c) **Coral Draghead** - Developed for dredging Pacific atolls, this drag will dredge coral after disintegration by blasting. It has a fixed main and after top body, but the grate section is hinge connected and telescopes into the fixed body part, and is equipped with teeth to aid in breaking up coral.

(d) **Newport Bay Draghead** - This dredge was specifically evolved for the purpose of effectively dredging a sloping bank of fine hard-packed sand into which the drags previously described cannot settle, and down which they tend to slide.

1.2.3 Distribution Systems

The distribution system of a hopper dredge affects its dredging efficiency to a major degree. When the solids discharged by the dredge pump are not distributed evenly throughout the hoppers and retained to an optimum degree, the effectiveness of dredging is reduced; also, when dredges are operated with overflowing hoppers, minimum turbulence in the hoppers is essential to reduce the loss of solids overboard.

Distribution systems are of three main types, mainly: Open Trough, Closed Pipeline, and Closed Manifold.

1.2.3.1 **Open Trough** - In the open trough system of distribution of the dredged material, distribution is made by the initial velocity of flow and trough gradient. The pump discharge enters at one end of the trough which is dimensioned and sloped as necessary to maintain
the velocity of the flowing mixture at a value sufficient to distribute it through the nozzles leading to the various hoppers. The troughs are usually partially covered to prevent splashing of the contents.

1.2.3.2 Closed Pipeline - The system is a simple extension of the discharge pipe from the pump. Thus, pump pressure is available for discharging material at any point of the pipeline. Its main disadvantage is in the fact that the velocity of the material entering the hopper from the pipeline is high and turbulence is produced, thus increasing the settling time of the suspended solids in the mixture.

1.2.3.3 Closed Manifold - The manifold type distribution system provides centralized control of distribution. The discharge from the dredge pumps is received by a manifold, sometimes called a distribution box, from which it is distributed to the hoppers through independent ducts extending from the box to each hopper. Although initially more costly than the previous system, this system has the advantage of reducing splashing and turbulence in the hoppers and equalization of hopper loading rates.

1.2.4 Hoppers

The hopper section of a modern hopper dredge in most cases is a single unit located between the pump room and the power plant, and because
FIG. 5 Distribution System of Dredge TAYLOR Showing Stilling Plate  
(Courtesy of the Corps of Engineers, U. S. Army)
of stability requirements, is divided into a number of compartments, each of which has one or two dump doors and is designated as a "hopper".

The area of the surface of the hoppers at the overflow level has considerable effect on the amount of suspended solids retained in the hopper. The objective is to provide as low a surface velocity and as long a path as possible between the distribution system nozzles and the overflow troughs. Due to practical limitations, not all the fine suspended solids can be settled into the hoppers. However, it has been found empirically that hoppers having a surface area of 1 square foot for each 2-1/2 cfm of pump capacity will yield satisfactory results.\(^{(79)}\)

To prevent short circuiting of the settling chamber of the hopper, baffle and stilling plates have been installed in many hoppers. These devices render the flow in the hopper more tranquil by absorbing the energy of the mixture discharged from the nozzles.

Each hopper is usually equipped with one or two hydraulically operated bottom doors for the discharge of the dredge material. To prevent sticking of the material to the hopper sides during discharge it is usual to slope the lower part of the hopper 50 to 60 degrees from the horizontal.

Rectangular or square shaped hoppers appear to be more efficient than conical shaped hoppers.

1.3 NEW DEVELOPMENTS AND TECHNIQUES

In recent years many devices and techniques have been introduced to the dredging field which aid in making dredging a simpler and more economical operation. Some of the more important innovations are as follows:
1.3.1 Hydraulic Swell Compensators

These devices automatically adjust the height of the dragarm so that the suction head is always in intimate contact with the bottom material, regardless of conditions at the surface. Many of these installations have been designed to compensate for swells up to 6 feet in height.

1.3.2 Electronic Control Consoles

Many of the recently built dredges have been so designed that the dredge operator can automatically control and adjust the height of the suction draghead, the distribution of the dredged materials to the hoppers, the concentration of suspended particles in the mixture, the speed and capacity of the dredge pump, the operation of the hopper doors, and other innumerable tasks, all from one electronically operated control point.

1.3.3 Ultrasonic Measurement of Suspended Sediment

A laboratory instrument recently developed for the determination of the size distribution and concentration of suspended sediment in streams may soon be adapted for use on hopper dredges.

The principle of operation is based on the attenuation of ultrasonic energy at frequencies from 2.5 to 25 mc. (98)

1.3.4 Convertible Dredges

Recently built, the trailing cutter suction hopper dredge FINCOSITA(70) is a prime example of a convertible dredge. The dragarm and suction tube of this dredge have been constructed so as to enable easy changeover from stationary dredging with a cutterhead and suction tube to dredging with a trailing suction draghead. Thus, since the FINCOSITA is equipped with both
hoppers and a floating pipeline, it can operate either as a hydraulic pipe-
line cutter head dredge or as a hopper dredge.

1.3.5 Pneumatic Breakwaters

It has been found that a curtain of air bubbles rising from a
submerged pipe will significantly reduce the heights of waves whose length
is less than 5 times the depth of water at the location of the submerged
pipe. (27)

Thus, for those dredges which are not equipped with swell compen-
sators it may prove feasible to utilize a pneumatic breakwater, allowing the
dredge to work in choppy or disturbed waters.
PART 2. DREDGE PUMPS

2.1 GENERAL FEATURES

The dredge pump is the heart of any dredging vessel. It is responsible not only for excavating and lifting the dredged material to the surface, but also, in the case of pipeline dredges, for the transportation of this material to the spoil area.

Dredge pumps are of the centrifugal radial type and must be designed to withstand the heavy wear and abrasion which they are subjected to by stones and solid objects in the dredged mixture. Although pump requirements vary widely - from the high head and low capacities required for pipeline dredges to the low head and high capacities for hopper dredges - the basic design considerations remain the same.

2.1.1 Basic Pump Components

2.1.1.1 Volute Casings - Nearly all new dredges now use dredge pumps with fabricated or cast outer casings, with all the wearing surfaces lined. Usually the liners are of special alloys, 400-650 Brinell Hardness. The completely lined pump on large dredges reduces operating costs 20% or more, depending on the materials handled. For easy access to the working parts of the pump the top half of the casing should be removable.
In the centrifugal pump, kinetic energy is imparted to the fluid by the action of the impeller vanes on the fluid. The purpose of the casing is to convert this energy due to velocity (kinetic energy) to energy due to pressure (potential energy). This is accomplished by gradually increasing the area of the casing so that the velocity is progressively decreased while the pressure is increased. To avoid sudden fluctuations in velocity, which lead to radial thrusts on the pump shafts and losses in pump efficiency, this transition from a small to large casing area should be as smooth and gradual as possible. To this end modern casing profiles are volute in shape and are laid out with a multi-radii development.

2.1.1.2 Impellers (Fig. 6) - Impellers are generally cast of alloy steel for long wear. Large impellers may have the hub cast and the shrouds and vanes fabricated from alloy steel and assembled by welding. Impeller speeds vary from 900 rpm for 8-inch (inlet diameter) pumps to 150 rpm for 36-inch pumps. Higher speeds cause excessive wear.

Discharge vane angles at the tip, sometimes referred to as exit angles, vary from 22-1/2° to 35° with the trend being toward lower angles. Entrance vane angles vary from 37° to 45°.
Fig. 6  TYPICAL DREDGE PUMP IMPELLER
Usually there are 4 or 5 impeller vanes in the average size dredge pump. This is a compromise between the opposing precepts of hydraulic efficiency and dredging practicality. Using less than this number will decrease the efficiency of the pump; while using more will reduce the clearance at the eye of the impeller where the vanes come together, and the large size materials may not pass through.

2.1.2 Primary Movers

A dredge pump is required to pump a wide assortment of material ranging from fine silt and clay to coarse sand and gravel and stones. It has been found that for any particular type of mixture a pump operates best at a particular speed.\(^{(79)}\) For this reason the pump motor should have a variable speed drive. In addition, it is distinctly advantageous to have a moderate speed range over which the pump can deliver full capacity so that solids of varying density can be pumped at a maximum rate.

Since a direct current, shunt wound motor with variable voltage most nearly satisfies these criteria, such a motor is preferred on dredges.\(^{(84)}\)

2.1.3 Location

To facilitate priming of the pump and to reduce the height of the suction lift, the dredge pump should be situated as low as possible in the hull of the dredge. Usual practice on hopper dredges is to locate the pump just forward of the hoppers.\(^{(79)}\)
2.2 EFFECTS OF SUSPENDED SOLIDS ON PUMP CHARACTERISTICS

2.2.1 Rheological Studies

When one pumps suspensions of solids, such as muds, the properties of the mixture are very distinct to that of water alone. The presence of suspended solid particles increases the viscosity and density of the water. This, of course, will affect the characteristics of any pump, pumping such a mixture.

Rheological studies of typical silt-clay-water mixtures which are often found in rivers and harbors have shown that these materials exhibit the properties of a Bingham body such that below a shearing stress of approximately 500 dynes per square centimeter the rate of shear is nearly proportional to the shearing stress; but at values of stress above 500 dynes per square centimeter, a very sharp increase in rate of shear is observed. Many of these materials have a certain value of shear stress, called the yield value, below which flow ceases.

Observations at Lehigh University indicate that when the concentration of solids is low (up to 1,200 grams per liter) the resulting mixture is essentially water with solids in suspension, and the solids settle rapidly. However, when the concentration is high (up to 1,400 grams per liter), the mixture appears to be homogeneous. It has the properties of a non-Newtonian fluid and the solids do not settle as rapidly.

Due to their very rapid settling, sand-water mixtures are a different case altogether, and exhibit none of the properties described above. Apparently sand-water mixtures do not affect the performance of the dredge
pump as markedly as silt-clay-water mixtures.\(^{(3)}\) However, the pitting and wear of the dredge pump is greater for sand particles than for silt or clay particles.\(^{(63)}\)

Fortino\(^{(49)}\) has shown that for any particular density of material being pumped there is an optimum velocity at which the solid material flow rate is a maximum. Curves have been presented which allow one to determine this optimum velocity for most types of bottom materials and all densities.

2.2.2 Pump Characteristics

It has been definitely established that there exists a difference between the pump characteristics of a centrifugal pump, pumping a water and suspended material mixture, and water alone. Experiments by Gregory\(^{(28)}\) show that as the concentration of solid materials in a mixture increases:

- (a) Head developed at a given capacity decreased.
- (b) Power input at a given capacity increased.
- (c) The efficiency at a given capacity decreased.

In a theoretical consideration of this problem Fairbank\(^{(19)}\) makes the following additional conclusions:

- (a) The drop in the constant speed-head-capacity characteristics varies not only as the concentration, but also as the particle size.
- (b) The ordinary affinity relations of centrifugal pumps are valid within small ranges of speed when pumping materials in suspension.
- (c) The power input at a given capacity varies directly with the apparent specific gravity of the mixture.
These conclusions have been verified only for sand-water mixtures and cannot justifiably be applied for high concentrations of silt-clay-water mixtures, which as previously discussed, exhibit the properties of a homogeneous non-Newtonian fluid.

Tests on a model dredge pump (shown in Fig. 7) at Lehigh University\(^{(32)}\) on various concentrations of a silt-clay-water mixture have resulted in the following conclusions:

(a) The total head (expressed in feet of water) developed by a pump increased linearly with the increase in fluid density. In some cases the increase was non-linear for densities above 1300 grams per liter. The total head (expressed in feet of fluid) is not affected by changes in density. This conclusion contradicts previous research.

(b) Brake horsepower increases linearly with fluid density.

(c) The efficiency at a given capacity decreases as the fluid density increases.

2.3 DESIGN OF DREDGE PUMPS

2.3.1 General

Every design of a dredge pump is a compromise in that, the impeller and the volute casing characteristics are determined not only by
Fig. 7
what is most efficient, but also by the requirement that fairly large objects must pass through the pump.

2.3.2 Design Considerations

2.3.2.1 Determination of Head - The total head or lift against which a dredge pump works is the sum of the static suction head, the vertical lift above the pump, the velocity heads, and friction, inlet and outlet losses.

(a) The static suction head can be calculated as

\[ H_s = S_2 \cdot B - S_1 \cdot A \]  

where:

- \( H_s \) = Static suction head in feet of fresh water
- \( S_1 \) = Specific gravity of the mixture being pumped
- \( A \) = Distance of the centerline of the pump above the channel bottom, in feet
- \( S_2 \) = Specific gravity of the water in which the dredge is operating (1.025 for sea water)
- \( B \) = Distance of water surface above the face of the dredged deposit

(b) The vertical lift is simply measured in feet and multiplied by \( S_1 \) to convert it to feet of fresh water.
(c) The velocity head can be expressed as

\[ S_1 \frac{y^2}{2g} \]

The net velocity head which the pump works against can be computed as:

\[ H_v = S_1 \frac{V_d^2 - V_s^2}{2g} \]  \hspace{1cm} (3)

where:

- \( H_v \) = Net velocity head in feet of fresh water
- \( V_d \) = Discharge velocity in feet per second
- \( V_s \) = Suction velocity in feet per second
- \( g = 32.2 \) feet per second per second

(d) The problem of friction, inlet and outlet losses in pipes carrying suspended materials has been the subject of many articles by various writers\(^{(3)(7)(10)}\)\(^{(28)(29)(69)}\) and enough data exists in the form of graphs and formulas that these losses can be estimated with sufficient accuracy.

The total suction head on a dredge pump is usually limited to between 22 and 23 inches of mercury.\(^{(21)}\)
2.3.2.2 Determination of Pump Capacity - The capacity of a pump, for any given dredge, is primarily determined from a consideration of the power available. Knowing the power available and the head which the pump must work against, the capacity of the pump required can be obtained from the following:

\[ Q = \frac{737 P}{w \cdot H \cdot e} \]  \hspace{1cm} (4)

where:

- \( Q \) = Pump capacity in cubic feet per second
- \( P \) = Power available in kilowatts
- \( w \) = Specific gravity of fresh water
- \( H \) = Total pump head in feet of fresh water
- \( e \) = Efficiency (usually 70% for well designed dredge pumps)

2.3.3 Piping and Pumping Layouts

2.3.3.1 Diameters - The diameter of both the suction and discharge piping should be selected so that the velocity in each is approximately 15 to 25 fps. It has been found that for most materials the optimum pumping velocity lies somewhere within this range. Usually the suction pipe is slightly larger than the discharge pipe. (79)
2.3.3.2 Pumps in Parallel - Many dredges now carry more than one dredge pump. Usually these pumps are connected in parallel so that their respective discharge pipes are led into a connecting wye where the combined flow is then discharged into the distribution system or a floating pipeline.

An investigation into such a layout was recently conducted at Lehigh University. Operation of pumps in parallel at different speeds and with different concentrations of suspended solids was compared to the operation of the pumps separately. Results showed only small effects on discharge and efficiency for the conditions tested. If one pump was held at constant speed while the speed of the other was varied it was found that the reduction in efficiency and discharge in the constant speed pump was very small. The minor disadvantageous effects that the change in conditions for one pump might have on the other pump in a combined flow system appear to be considerably outweighed by the added head loss which would have been incurred had the pumps been pumping through separate pipelines.
2.3.4 Design Methods

2.3.4.1 Theoretical Methods - The theory of the centrifugal pump is based on the time rate of change of the moment of momentum of a fluid particle with respect to the axis of rotation. If one considers the classical derivation, based on the above theory, for the head developed by a theoretical impeller having an infinite number of vanes and applies the same steps to the solid and water particles of a dredged mixture, we can obtain the expression for the head developed in feet of mixture as:

\[
H = \frac{S_s p u (U - V_s \cot \beta) + (1 - p) U (U - V_w \cot \beta)}{g S_a}
\]

where:

- \(H\) = Theoretical head in feet of mixture produced by a theoretical impeller having an infinite number of vanes.
- \(S_s\) = Specific gravity of the solid particles.
- \(p\) = Concentration of solid material, by volume.
- \(U\) = Peripheral velocity of the impeller.
- \(S_a\) = Apparent specific gravity of the mixture.
- \(V_s\) = Radial relative velocity of a solid.
particle at exit from impeller with respect to the impeller vane

\[ V_w = \text{The same as above applied to a water particle} \]

\[ \beta = \text{Acute angle between tangents to the impeller and impeller vane at exit (exit angle)} \]

\[ g = 32.2 \text{ feet per second per second} \]

It should be noted that the above equation is based on assumptions which are only approximately true and which are completely false in some cases. However, it can be used along with other classical pump equations by the designer to yield very approximate predictions of pump performance. For more accurate predictions the designer must resort to more practical methods which are based on empirical and "rule of thumb" rules.

2.3.4.2 Practical Method - The practical method of dredge pump design utilizes certain design parameters which enable the designer to size the pump; that is, these parameters allow one to tentatively assign values to the speed, diameter, width, and other dimensions of the pump.

(a) Specific Speed and Speed Factors - The specific speed of a pump is defined as the speed of a homologous pump operating under a head of 1 foot and pumping one
gallon per minute. It can be expressed as:

\[ N_S = \frac{N Q^{1/2}}{H^{3/4}} \]  \hspace{1cm} (6)

where:

\[ N_S = \text{Specific speed} \]
\[ Q = \text{Discharge rate, gallons per minute} \]
\[ N = \text{Impeller speed, rpm} \]
\[ H = \text{Total pump head, feet of fresh water} \]

The specific speed has a certain range of values for well-designed dredge pumps. Although these values often depend upon the judgment and experience of the designer, it has been long known that the range lies between 1200 and 2000 for large dredge pumps. Utilizing this range, and knowing \( H \) and \( Q \) from preliminary considerations, the operating range of the impeller speed can easily be computed.

The speed factor, \( \phi \), is defined as the ratio of the actual peripheral velocity of the impeller to the theoretical velocity of the fluid. Since the actual peripheral velocity in feet per second is:

\[ \frac{\pi DN}{12 \times 60} \]

where \( D \) is the diameter of the impeller in inches;
since the theoretical velocity is:

\[ \sqrt{\frac{2 g H}{2}} \]

then:

\[ \varphi = \frac{\pi DN/12 \times 60}{\sqrt{2 g H}} = \frac{DN}{1840 \sqrt{H}} \] \hspace{1cm} (7)

The constant, \( 1840 \times \varphi \), is again known to have a certain range of value for well designed dredge pumps (1500 - 1900 for large dredge pumps). \( (79) \)

Thus, knowing this value, and determining the range of \( N \) and \( H \) as before, the impeller diameter can be calculated.

(b) Width of Impeller - In determining the width between shrouds it must be remembered that there is a minimum width which can be tolerated, so that solid objects may pass through the pump. To insure this, the width is usually taken as some percentage (approximately 60%) of the discharge pipe diameter. \( (79) \).

The width should also be selected so that the radial velocity between impeller vanes is not excessively large.

Although dredge pump impellers were at one time single-shrouded, modern dredge pumps are double-shrouded because it has been shown that the increase
in efficiency produced in double-shrouded impellers due to better guidance of the flow more than offsets the loss due to the disk friction of the second shroud. (82)

(c) **Eye of the Impeller** - The diameter of the eye should be slightly larger than the diameter of the suction pipe.

In a centrifugal pump the flow must suddenly change from an axial to a radial direction as it enters the pump. To make this change gradual and thus reduce eddies and turbulence to a minimum the suction shroud should be given a generous radius of curvature. (79) This curvature is extended into the mouth of the suction pipe where it becomes tangent to the pipe. The discharge shroud should also be given a similar curvature.

(d) **Casing** - The casing design greatly affects the efficiency of a pump. In it, the velocity energy created by the impeller must be converted into pressure energy by gradually increasing the cross-sectional area of the casing. (23) As previously mentioned, this is best done by laying out the circumference of the casing as a spiral or volute curve. (35) The latter curve can be empirically determined by radii whose origins are slightly offset from the pump centerline. The more radii
used, the better the final design. The outlet section of the casing should have an area approximately equal to the area of the discharge pipe while the throat or cutwater section should have an area about 65% of the discharge pipe area in order to pass solid objects. It is important, however, to keep this area as small as possible in order to reduce recirculation and losses in volumetric efficiency.

Regardless of how good the casing design is, the presence of axial and radial thrusts is never eliminated. Since these thrusts are dynamic and repeating in nature, bearings and shafts should be constructed of high fatigue strength material.

(e) Impellers - The problem of determining the shape of impeller vanes, along with entrance and exit angles, has always been a difficult one in dredge pump design. This problem is compounded by the effects of pre-rotation of the fluid in the suction pipe and by the fact that the flow of the suspended particles in the fluid seldom conforms to the curve of the vane. Tests at Lehigh University on a silt-clay-water mixture show that, at the present time, the most promising vanes tested are involute in shape with a 45°
entrance angle and a 22-1/2° exit angle; or spiral in shape with a 45° entrance angle and a 28-3/4° exit angle. These angles were measured between tangents to the vane and the impeller circumference.

As stated previously, the number of vanes on a large dredge pump is usually 4 or 5. This is a compromise between the requirements of hydraulic efficiency and the requirement that large objects must pass through the pump.

Service water is usually supplied to the seals of a pump in order to cool them and keep them from wearing too quickly. The use of vanes or radial ribs on the outside of the shrouds of the impeller causes flow of this service water into the clearance space between the shrouds and casing with consequent generation of pressure opposing the head developed by the impeller, thereby reducing leakage. This opposing pressure caused by the shroud ribs also tends to reduce the amount of sand or other solids in the space between impeller and casing, thus reducing wear in this area.
2.4 SIMILITUDE OF DREDGE PUMPS

2.4.1 General

The equations and concepts presented in the previous section of this report may be used to predict, with fair accuracy, the performance of a specific pump. However, for a complete and accurate verification of these predictions the dredge pump designer is left with only two recourses. The first is to test the completed full scale pump in operation; while the second consists of testing a scaled-down version, or model of the prototype pump. Obviously this second method is to be preferred due to its relatively lower cost as compared with the first method.

2.4.2 Similitude

Similitude, as applied to hydraulic models, goes considerably beyond the superficial aspects of geometric similarity with which it is sometimes erroneously identified. Similitude can be defined as a known and usually limited correspondence between the behavior of a model and that of its prototype. (60)

Similitude may be divided into three major divisions:

(a) **Geometric Similitude** - exists when the ratio of all homologous dimensions are equal.

(b) **Kinematic Similitude** - exists when there is similarity of motion between model and prototype.

(c) **Dynamic Similitude** - exists when there is similarity of masses and forces between
model and prototype.

In any model investigation, in order to use the results, reliable prediction equations must be developed. In the development of a prediction equation two classical methods are used. The first method, referred to as the experimental method, consists of establishing the effect of pertinent variables upon the quantity to be predicted by observation and measurement. The second method, usually referred to as the analytical method makes use of the data determined experimentally and from this data proceeds to develop relationships among the significant variables based on the pertinent natural laws.

The general form of the prediction equation for any phenomenon may be determined by dimensional analysis. Neglecting the effects of viscosity, which are assumed to be small, dimensional analysis yields the following relationships for pumps.

\[
\frac{Q}{N D^3} = C_1 \quad (8)
\]

\[
\frac{g H}{N^2 D^2} = C_2 \quad (9)
\]

\[
\frac{P}{\rho N^3 D^5} = C_3 \quad (10)
\]

where \(Q\) is the flow, \(N\) is the impeller speed, \(D\) is the impeller diameter, \(g\) is the gravitational constant, \(H\) is the pump head, \(\rho\) is the fluid density and \(P\) is the power. Dimensional units can be of any system as long as these systems correspond when similar pumps are compared.
The above relations should be equal to the same constant for homologous, or similar pumps.

From these equations can be derived the specific speed, \( N_s \):

\[
N_s = \frac{N Q^{1/2}}{H^{3/4}}
\]  

(11)

where:

\( N \) = revolutions per minute
\( Q \) = gallons per minute
\( H \) = feet

Again, for similarity, the specific speeds for a model and its prototype should be equal.

2.4.3 Prediction Equations

Usually for model pump testing, the pump head and velocities are kept equal for both the model and its prototype and the fluid in each pump is identical.

To obtain a relationship between model and prototype discharge it is only necessary to consider that

\[
Q \propto V L^2
\]

(12)

Since \( V \), the velocity, is the same in model and prototype

\[
\frac{Q_p}{Q_m} = \left( \frac{L_p}{L_m} \right)^2
\]

(13)
where the subscripts \( p \) and \( m \) refer to refer to prototype and model respectively. The ratio \( L_p/L_m \) is known as the scale ratio and is the ratio to which all corresponding dimensions of the model and prototype are reduced.

Using the equality of specific speeds in the model and the prototype we can find the prototype-model speed relationship as follows:

\[
(N_s)_m = (N_s)_p
\]  
\[\left( \frac{N Q^{1/2}}{H^{3/4}} \right)_m = \left( \frac{N Q^{1/2}}{H^{3/4}} \right)_p
\]

since \( H_m = H_p \), then

\[
N_m = \left( \frac{Q_p}{Q_m} \right)^{1/2} N_p
\]

or

\[
N_m = \frac{L_p}{L_m} N_p
\]

The next characteristic to be considered is that of power.
The assumption made here which is often used is that model and prototype efficiencies will be equal. This is a conservative assumption since the actual prototype efficiency is higher than the model's as is explained later.

In order to relate the brake horsepower of the model to that of the prototype, it is recalled that the water horsepower, WHP, is equal to the brake horsepower, BHP, times the efficiency.

\[
WHP = \frac{\gamma Q H}{550} = e \times BHP
\]
Since the fluid densities, heads, and efficiencies are assumed equal in the model and the prototype, the resulting relationship is

\[ (B H P)_p \frac{Q_p}{Q_m} = (B H P)_m \]  \hspace{1cm} (19)

or

\[ (B H P)_p = \left( \frac{L_p}{l_m} \right)^2 (B H P)_m \]  \hspace{1cm} (20)

There are other similitude relationships which can be used to predict other prototype factors, but the ones presented here are of primary interest to the designer.

2.4.4 Scale Effects

Due to the effects of viscosity and the impossibility of controlling all the variables involved, complete pump similitude can never be accomplished. (36)(42)

One of the main reasons this is so, is because it is impossible due to practical limitations, to reduce the interior surface roughness of the model to the ratio of the prototype roughness required by the criterion of geometric similitude. This difference will cause variations in the proportional losses of head due to hydraulic and mechanical friction which in turn will result in differences in efficiency between the two pumps. (47)

Moody(37) has proposed the following empirical formula to express the relation between the efficiencies of model and prototype pumps.
\[ e_p = 1 - (1 - e_m) \left( \frac{L_m}{L_p} \right)^{1/4} \left( \frac{H_m}{H_p} \right)^{1/10} \]  

(21)

When the heads in the model and the prototype are equal, this becomes:

\[ e_p = 1 - (1 - e_m) \left( \frac{L_m}{L_p} \right)^{1/4} \]  

(22)

2.4.5 Model Pump Test Results

In recent years, the prediction of prototype behavior by testing model pumps has been amply verified by comparison of actual prototype test data with model test data.(30)(87)(104) In practically every case it has been shown that model testing is a reliable method of predicting prototype performance.

In addition, it might be pointed out that testing a model is relatively inexpensive due to its smaller size, capacity and power consumption. Whereas most hydraulic laboratories can conveniently test a model, most would find it difficult, if not impossible, to test the large prototype pumps being manufactured at present due to lack of adequate facilities.

For these reasons, model testing remains the most important method for:

(a) Guiding and verifying theoretical developments in design.

(b) Evaluating performance under special conditions such as cavitation.

(c) Performing acceptance tests in lieu of full-scale tests.
(d) Aiding the evaluation of hydrodynamic loading for mechanical design.

Tests on such model pumps have already proved to be of great value in the field of dredging. As an example, the following conclusions based on the performance of a model dredge pump were reached at Lehigh University. (34)

1. The efficiency of the dredge pump may be substantially increased, possibly by 8 to 12 percent in absolute value by redesigning the pump impeller. The most promising impeller forms tested for design conditions were:
   (a) That having a 45° entrance angle, a 22-1/2° exit angle, with an involute vane profile.
   (b) That having a 45° entrance angle, a 28-3/4° exit angle, with a logarithmic spiral vane.

2. The model dredge pump performance was not appreciably affected by slight changes in the characteristics of the silt-clay-water mixture being pumped.

3. It is possible to pump silt-clay-water mixtures having a density of 1410 grams/liter (specific gravity = 1.41)

4. Test results show no significant dependence upon the mixture velocities (or a Reynold's number).
2.5 CAVITATION OF DREDGE PUMPS

2.5.1 General

Cavitation in a hydraulic structure occurs when the pressure at some point is reduced to the vapor pressure of the flowing fluid (about 0.5 psi for water). Cavitation is the action which takes place in this low-pressure region and consists of the formation, transportation, and collapse of vapor cavities. The vapor cavities which form in the regions of vapor pressure are carried downstream by the flowing fluid to collapse or implode as they reach a zone of higher pressure. The tremendous forces that accompany these implosions which occur on solid boundary surfaces cause disintegration of the boundary material. This destruction is termed "cavitation-erosion". (48)

Cavitation may be induced in many ways. The following causes are encountered most frequently: (16)(92)

1. Curving a boundary surface too rapidly away from the normal path of high-velocity fluid streams.
2. Permitting irregularities or discontinuities in boundary surfaces which are subject to high-velocity flow.
3. Using extreme variations in elevation in a conduit system which produce siphonic action if not controlled properly.
4. Moving unstreamlined objects through liquids at high speeds.
5. Expanding the flow area too rapidly in the direction of motion.

2.5.2 Theories of Cavitation

2.5.2.1 Cold Water - One of the more important parameters in dealing with cavitation is the NPSH - the net positive suction head produced by a pump for any given discharge. The NPSH is defined as the total head available to the pump above the vapor pressure and is computed as:

\[ \text{NPSH} = H_{\text{atm.}} + \frac{P_s}{w} + \frac{V_s^2}{2g} - H_v \]  

(23)

where:

- \( H_{\text{atm.}} \) = Atmospheric head in feet of liquid
- \( P_s \) = Pressure measured at the pump inlet in pounds per square foot
- \( w \) = Density of liquid in pounds per cubic foot
- \( V_s \) = Velocity in the suction pipe at the pump inlet in feet per second
- \( g \) = 32.2 feet per second per second
- \( H_v \) = Head of liquid in feet corresponding to the vapor pressure of the liquid

Values of the minimum NPSH for incipient cavitation can be determined by tests for different rates of discharge and these minimum values for incipient cavitation can be plotted as a function of discharge or specific speed to form a smooth curve.
Values of the minimum NPSH for incipient cavitation, referred to as the critical NPSH, can then be determined for any rate of discharge by interpolation from the graph. If a centrifugal pump is operated below this critical NPSH, cavitation will develop and the total head will be reduced by an amount $\Delta H$ and the efficiency decreased.

Usually, in order to insure the safe operation of a pump, the pump is operated at a value of NPSH, referred to as the plant NPSH or $(\text{NPSH})_p$, which is higher than the critical NPSH, $(\text{NPSH})_c$, by the amount $\Delta \text{NPSH}$. In equation form, the plant NPSH is equal to the sum of the critical NPSH and $\Delta \text{NPSH}$ or:

$$(\text{NPSH})_p = (\text{NPSH})_c + \Delta \text{NPSH} \quad (24)$$

Dividing this equation by the total pump head, $H$, and calling the resulting ratios sigma, $\sigma$, we have:

$$\sigma_p = \sigma_c + \frac{\Delta \text{NPSH}}{H} \quad (25)$$

Until recently, it was generally accepted that for all similar pumps operating at corresponding points on the Q-H curve the value of plant sigma, $\sigma_p$, was constant, regardless of head. This statement
was frequently referred to as "Thoma's Law". Recently, however, many cases have been discovered which deviate considerably from this rule.

However, laboratory tests prove that when the ratio $\frac{\Delta H}{H}$, where $\Delta H$ is the amount of head lost through cavitation effects, is held constant then the critical sigma will also be constant.

When two similar pumps are operated at different heads at their best efficiency point so that $\Delta H$ in each case is equal to a constant value, then Tenot's equation holds:

$$\frac{\frac{\sigma_{p1} - \sigma_c}{\sigma_{p2} - \sigma_c}}{H_2} = \frac{H_1}{H_2}$$  \hspace{1cm} (26)

where the subscripts 1 and 2 refer to the first and second pump, respectively and $\sigma_c$ is equal for both pumps if they are similar. Rearranging:

$$\frac{\sigma_{p1} - \sigma_c}{H_1} = \frac{\sigma_{p2} - \sigma_c}{H_2}$$  \hspace{1cm} (27)

or

$$\text{(NPSH)}_p1 - \text{(NPSH)}_c = \text{(NPSH)}_p2 - \text{(NPSH)}_c$$

$$\Delta \text{(NPSH)}_1 = \Delta \text{(NPSH)}_2$$

thus proving that both points have the same margin of safety against cavitation.
If the two similar pumps are operated at the same head, as is usual in model-prototype testing, then this equation reduces to "Thoma's Law".

\[ \sigma_{p1} = \sigma_{p2} \]  

(28)

Thus it is seen from the above that when \( \sigma_c \) and \( (NPSH)_{p1} \) or \( (NPSH)_{p2} \) are determined for any head by experiment, the plant NPSH for any other head can be directly calculated.

2.5.2.2 Liquids other than Cold Water\(^{(76)(77)(86)(91)}\) - For liquids other than cold water, we can use essentially the same theories as above, but we utilize a "corrected" value of the cold water NPSH which can be obtained from plots of empirical and thermodynamical data. Thus, if the cavitation characteristics of any pump are known for cold water, then these characteristics can be obtained for the same pumps handling oil, butane, etc.

2.5.3 Cavitation of Dredge Pumps

The material which passes through a dredge pump is essentially a mixture of water and suspended particles. As such it is separate from the two classifications found above.

It is neither pure water, nor a homogenous liquid entirely different from water, such as butane, oil, etc. whose thermodynamic properties are known at all times.
Therefore the analysis of cavitation in pumps handling a suspended solids mixture cannot follow the analysis described for pumps handling liquids other than water. Instead, separate tests must be performed for each type and concentration of dredged mixture in order to determine the cavitation behavior of a dredge pump for all conditions.

Tests on a model dredge pump at Lehigh University\(^{(38)}\) show that the cavitation characteristics of a dredge pump can be improved by designing the impeller vanes in an involute or spiral profile and decreasing the exit angle so as to lessen the resistance to flow. Tests are also underway to determine the effects of the concentration of suspended solids on cavitation characteristics.
APPENDIX I

ABSTRACTS
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1. Agostinelli, A., Nobles, D. and Mockridge, C. 
AN EXPERIMENTAL INVESTIGATION OF RADIAL THRUST IN CENTRIFUGAL 
PUMPS, 
American Society of Mechanical Engineers, Paper No. 59-Hyd.-2, 
1959

Unbalanced radial forces in centrifugal pumps are of concern to the 
author in order that shaft and bearing designs can be modified for the re-
sulting shaft deflections and cyclic loading of the rotating shaft. Experi-
ments were run on sixteen conventional design, constant velocity, volute 
casing pumps of specific speeds between 25 and 165.

In general, the resultant radial forces were a maximum at shut off; 
decreased to a minimum near the best efficiency point, but not necessarily 
zero; and increased with flow beyond this point. For relatively high 
specific speeds ($N_s > 100$) the minimum thrust occurred at a higher capacity 
than 100% of normal, while lower specific speeds ($N_s < 50$) showed a minimum 
force at less than 100% of normal. Also, for higher specific speed, the 
minimum forces occur at a single point and on either side of this point 
the rate of increase is high. For the lower specific speed, there is a 
larger capacity range over which the minimum force exists.

The authors suggest that if the forementioned characteristics 
indicate the degree of hydraulic mismatch between impeller and casing, high 
specific speed pumps have a very small range over which the two are compatible 
while for lower specific speed pumps, the opposite appears to be true.

Three other series of tests were run by the authors. One was run 
with concentric circular casings, another with a modified concentric casing 
and the third with double volute casings. From the tests on circular casings 
it was found that certain combinations of impellers and casings allow ef-
fective minimizing of unbalanced radial forces. For casing to impeller
diameter ratios above about 1.3, the unbalanced radial forces were much lower than those with the standard volute except in the best efficiency range where they were still only about 40% of the radial thrust at shut off.

Double volute casings are effective in minimizing the unbalanced radial thrust on impellers; but this is often an unsuitable solution for small pumps. The authors, therefore, attempted the design of a modified casing for small pumps which would improve the radial force problem without the use of the double volute. Their modified collector is circular for a portion of the angular distance from the cutwater before the radial extent of the casing wall is increased to the discharge nozzle. For comparison the same impellers were tested with single, double and the modified volutes.

The results of this testing showed that for all specific speed tests, the double volute casing gave the lowest radial loads on the impeller. However, for specific speeds of 55 and 165 the modified casing gave results comparable with the double volute; and for all specific speeds showed a very good improvement over the volute casing.
This paper deals with intensive studies which were made to determine internal flow characteristics of high efficiency, high head centrifugal pumps of commercial design. The experimental objectives of this study were to obtain a complete analysis of both instantaneous and average values of pressures and velocities in the volutes of the pumps investigated. The results of these tests are:

1. There is practically no instantaneous velocity variation in the impeller discharge at normal capacity and only slightly more at low or high capacities during the time of passage of one vane space past a measuring station.

2. There is a strong circulation between the volute and the impeller clearance space which apparently acts as an energy pump and helps to minimize losses.

3. For low-capacity conditions, double peak-velocity profiles were found, probably due to a centrifuge action of the shroud.

4. For normal capacity, the radial-velocity distribution around the volute is relatively uniform, while for high or low capacities large variations are found.

5. For low capacity, a high ratio of inflow is observed in the region of the tongue, while for high capacity a high outflow occurs in the same area.

6. Non-uniform velocity and static-pressure distribution combine to produce unbalanced radial forces on the impeller.
Maximum values exist during low-capacity operation, while the forces are at a minimum for normal rates of discharge.

7. A considerable variation in the deviation between the vane exit angle and the relative velocity was observed. The average deviation was greatest for low-capacity conditions and least for high-capacity conditions. At normal discharges it was 9 degrees.
3. Chevalier, J.
REVIEW, DESCRIPTION AND CRITICAL EXAMINATION OF CAVITATION TEST
METHODS ON SCALE MODELS,
La Houille Blanche, a report of Working Group No. 1 of
Societe Hydrotechnique de France, No. 4, September 1962

The first part of this article gives a description of experimental
means and installations, especially with respect to the measurements of the
amount of air contained in water. The second part discusses scale effect
and cavitation similitude at length. The third part of the article dis-
cusses the presentation of results and analyzes present methods of deter-
mining critical sigma. In view of their diversity, a graphical method is
suggested whereby comparisons between laboratories can be made, and which
does not require existing installations to be modified for its application.
This article presents a general description of the modern hydraulic dredge. Outlined are the latest improvements which have greatly increased production in both soft and hard materials.

Nearly all new dredges now use dredge pumps with fabricated or cast outer casings, with all the wearing surfaces lined. Usually the liners are of special alloys, 400-650 Brinell hardness. The completely lined pump on large dredges probably reduces pump operating costs 20% or more, depending on the materials handled. On large dredges average pump efficiency varies from 60% to 75%.

Impellers are generally made of alloy steel for long wear. Large impellers may have the hub cast and the shrouds and vanes fabricated from high-alloy steel and assembled by welding. Impeller speeds vary from 900 rpm on 8 inch pumps to 300 rpm on 36 inch pumps. Higher speeds cause excessive wear. Specific speeds vary from 1200 on 10 inch dredges to 800 on 36 inch dredges. Discharge vane angles at tip vary between 20° and 30°, and entrance angles vary 16° to 24°. Pump shaft diameters vary from 6 inches to 18 inches, with shafts usually made of SAE 4340, or equal, high alloy steel.

Since about 1925, horsepower on dredge pumps and cutters has almost doubled. On dredge pumps the horsepower varies from 600 hp on 10 inch discharge dredges to 10,000 hp on 36 inch dredges. Cutter horsepower varies from 200 hp to 400 hp on 10 inch and 14 inch dredges; 1500 hp
to 2500 hp on 28 and 30 inch dredges.

The cutter is probably the most important item on a dredge. Dredge cutters are now usually built with removable wear edges, cast in two or three sections, with various types of high alloy steel teeth cast directly on the edges. Anchor handling booms that heretofore were used on few dredges are now in almost universal use. They will, under adverse operating conditions, reduce down time by several hours per day.

Probably the most important new development is the direct suction pipecutter drive which combines the function of the cutter shaft and the suction pipe and eliminates the costly cutter shaft and its maintenance. In addition, pontoon line ball joints that eliminate the usual bolted connections are coming into use, permitting the line to lay lower in the water and making it more stable and less affected by wave action. Pontoons can be smaller and the connecting time for the ball joints is cut almost in half.

As a result of all these modern improvements, dredge production has increased significantly in recent years. As an example, a well-designed 10 inch dredge, with 400 hp to 600 hp on pump and 75 hp to 200 hp on cutter, on 2000 feet of pipeline, will conservatively pump 250 to 400 cu. yds. per hour in soft material and 50 to 1000 cu. yds. per hour in soft rock. A well-designed 30 inch dredge, with 5000 hp to 8000 hp on pump and 2000 hp on cutter, will pump 2000 to 4500 cu. yds. per hour in soft materials, and 200 to 2000 cu. yds. per hour in soft to medium hard rock, and on pipeline lengths up to 15,000 feet and longer.
5. Fairbank, L. C.
THE EFFECT OF MATERIAL IN SUSPENSION ON THE CHARACTERISTICS
OF CENTRIFUGAL PUMPS,
Submitted as a Master's thesis at University of California,
1940

Tests conducted by the Mississippi River Commission have de-
finently established that there exists a difference between the pump
characteristics of a centrifugal pump pumping a water and suspended materials
mixture and water alone. Experiments by Gregory show also that as the con-
centration of solid materials increased:

1. Head developed at a given capacity decreased.
2. Power input at a given capacity increased.
3. The efficiency at a given capacity decreased.

The theory of the centrifugal pump is based upon the time rate
of change of the moment of moment of a fluid particle with respect to the
axis of rotation. Using this theory we can derive the so-called "affinity
laws".

\[ Q \propto N \]
\[ H \propto N^2 \] \hspace{1cm} (29)
\[ HP \propto N^3 \]

The flow of a suspension through the impeller of a centrifugal
pump can be considered as consisting of two parts.

1. The flow of water, \( Q_w \), with a relative velocity
   with respect to the impeller blade, \( \dot{V}_w \).
2. The flow of solid materials, \( Q_s \), with a relative
   velocity with respect to the impeller blade, \( \dot{V}_s \).
From a theoretical consideration of an impeller of infinite number of blades it can be seen that if the impeller is operating with a discharge pressure of zero, the velocities of the solid and water particles will be the same. However, if there exists some finite discharge pressure, the resultant of the forces acting on the water will be less, whereas the forces acting on the solid particles will remain nearly the same. This would indicate then that the tangential component of the velocity of the suspension at impeller exit is less than that for water alone, and hence the head developed by the impeller, as given by the classical equation would be less for the suspension than for clear water. A method of approximating this decrease in head is now developed as follows.

The assumptions made in the development of the classical theory were retained and the following additional assumptions made:

1. The solid particles follow a path with respect to the impeller vanes that corresponds with the contour of the vane.
2. The solid particles leave the vanes at the exit angle at all rates of discharge.
3. The resistance force due to the relative velocity between the solid and water particles is of a viscous nature and follows Stokes Law.

Considering the classical derivation for the head developed by an impeller of an infinite number of vanes and applying the same steps to the solid and water particles, one can obtain the expression for the
head developed in feet of mixture as:

\[
H = \frac{S_s p U (U - V_s \cot \beta) + (1 - p) U (U - V_w \cot \beta)}{g S_a} + \frac{1}{g S_a}
\]

where:

- \(S_s\) = spec. gravity of the solid particles
- \(p\) = the conc. of solid material by volume
- \(U\) = peripheral velocity of the impeller
- \(S_a\) = apparent specific gravity of mixture
- \(g\) = 32.2 ft./sec.\(^2\)
- \(\beta\) = acute angle between tangents to impeller and impeller vane
- \(V_s\) = radial relative velocity of a solid particle at exit from impeller, with respect to the impeller vane.
- \(V_w\) = same as above applied to a water particle.

Experiments were carried out for a sand-water mixture and analytical results were found to be in good agreement with experimental results (approximately 6% in error at 20% concentration).

The writer makes the following conclusions:

1. 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.
2. The drop in the constant speed-head-capacity characteristics varies not only as the concentration, but also as the particle size of the suspended material.

3. The fall velocity of the suspended material is the most important property in predicting the effect of the material on the pump performance.

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

5. The power input varies directly with the apparent specific gravity of the suspension being pumped.

6. The capacity for maximum efficiency of a centrifugal pump remains constant for all concentrations and particle sizes.

7. The ordinary affinity relations of centrifugal pumps are valid within small ranges of speed when pumping materials in suspension.

8. The quantitative expression for H will serve as a first approximation in predicting pump performance.
6. Fischer, K.
INVESTIGATION OF FLOW IN A CENTRIFUGAL PUMP,
Mitterlungen des Hydraulischen Institutes der Technisches
Hochscule, Munchen, No. 4, 1931.
Translation, National Advisory Committee for Aeronautics,
Technical Memorandum No. 1089

The actual output of a centrifugal pump is considerably less than
that stipulated theoretically on the assumption of an infinite number of
vanes. The assumptions which are necessary for the prediction of the
flow pattern, especially the one that impeller passages run always full
with active flow lead to erroneous conceptions of the flow distribution.
The pump used in this investigation was constructed such that the flow
could be observed during actual operation. This was accomplished by
a glass wall on the pump. The flow was observed during the operation of
the pump by means of a rotoscope (this instrument is based on the phe-
nomenon that the reflected image of a stationary object rotates when the
plane of reflection is rotated). The flow was photographed by means of a
camera operated by chains and counter shaft from the pump, simultaneously
with the rotoscope. This simultaneous use provided accurate timing of the
flow attitude for photographing. The pump was equipped with a set-up to
introduce dye to the flow to improve the visibility of the mixture. This
report shows photographs of various flow conditions and presents the re-
lationships between theoretical values and actual values.
7. Fortino, E. P.
VISCOSITY AND PUMPABILITY OF BOTTOM MATERIAL IN A DREDGING CHANNEL,
Marine Design Division, United States Corps of Army Engineers,
Philadelphia District, 1961

A rheological study of Delaware River bottom materials shows that on the average, these materials have the properties of a Bingham body such that below a shearing stress of 500 dynes/cm$^2$ the rate of shear is nearly proportional to the shearing stress; but at values of stress above 500 dynes/cm$^2$, a very steep increase in rate of shear is observed.

Results of a laboratory investigation into the properties of this material show that the behavior of the mud samples depends on their previous history. Properties also changed under constant load. All these influences caused the rate of shear for any one shear stress to vary in an indefinable manner.

Experiments show that for any particular density of material being pumped there is an "optimum velocity" at which the flow rate is a maximum. Curves are included in the report which allow one to determine this "optimum velocity" for most types of bottom materials and all densities.
8. Fujie, K.
THREE DIMENSIONAL INVESTIGATION OF FLOW IN CENTRIFUGAL IMPELLER WITH STRAIGHT-RADIAL BLADES,

The flow through the passage of rotating impellers with single and double shrouds has been studied experimentally by means of a test rig, so designed that it is possible to measure directly the relative velocity, flow direction, total and static pressure on several points in the passage while the impeller is rotating. A yaw meter was used to obtain these pressures in the flow of air through the impeller.

The theoretical analysis was made under the assumption of two-dimensional potential flow (irrotational motion of an incompressible ideal fluid where the flow is uniform across the passage normal to the impeller shroud surface).

The calculated results are compared with the measured results in terms of relative velocity under the given operating conditions. The velocity distribution and the flow direction obtained in the experiment indicate that secondary flows in the boundary layer tend to shift the low-energy air toward the negative or suction surface of the blade in the passage. In the case of the single shrouded impeller, there was observed a counter effect along the casing surface apparently caused by the leakage through the clearance space between the blade and the casing. This results in a vortex flow in the passage. It is concluded that secondary flows consequently dominate the flow condition, and make it so complicated as to be impossible to conjecture it by means of theory.
9. Gregory, W. B.
PUMPING CLAY SLURRY THROUGH A FOUR-INCH PIPE,
Presented to American Society of Mechanical Engineers, New York,
March 9, 1927

This paper gives the results of experiments in pumping a clay
slurry, and the friction losses in pumping the material through a 4 inch cast
iron pipe over a 200 ft. length. The author is concerned with the velocity
which is the most economical for pumping a particular slurry with various
concentrations of particles through a given pipe over a given length. The
test setups are described in great detail.

The experiments show that the relationship between total head lost
in 200 ft. and the velocity is almost the same for various concentration
of solids as it is for water (murky), as long as the velocities are high.
For velocities lower than the critical velocity, the head lost for a given
concentration does not vary appreciably with the velocity. Another inter-
esting result was that the critical velocity varies with the concentration
of particles. The critical velocity being higher for a greater concen-
tration of particles.

Pump tests were made for various concentrations of particles, and
plots of Hp input versus gallons per minute, efficiency versus gallons per
minute and head versus gallons per minute are included in the paper. It
was found that the material had some properties of a viscous fluid, but
that in many ways it behaved like a plastic solid. Pumping was found to
be most efficient for a given concentration of particles at a definite
critical velocity.
The clay slurry is non-viscous in the following ways:

1. The $V_c$ (critical velocity) is not determined by the formula applicable to viscous fluids.

2. The critical velocity is greater as the density increases.

3. The velocity does not influence friction loss below $V_c$.

The author suggests that the presence of the solids causes the resistance to vary in some opposite way to that of the viscous fluid, so that the sum of the effect of viscosity and of the solid particles cancel each other at velocities lower than the critical. Further, the body of slurry has a starting resistance which varies with the concentration of particles.
The principle intention of the writer is to supplement the paper and to briefly summarize the current research program at Lehigh University aimed at improving the efficiency of dredge pumps, particularly for pumping silt-clay-water mixtures.

In describing various hydraulic dredges mention might also be made of the "portable" type dredges. These are hydraulic pipeline cutter-head types with the main dredge pump driven by a high-speed Diesel engine through a reduction gear. The hull is approximately 52 ft. by 20 ft. by 4 ft. and they can be disassembled and transported overland. The pump has a 13-1/4 inch diameter suction and a 12 inch discharge pipe and is operated by a 260 hp motor. It has a maximum digging depth of 26 feet and is capable of pumping 100 to 300 yds. per hour.

Mr. Erickson mentions that the discharge vane angles at tip vary between 20° to 30° and entrance angles vary 16° to 24°. The writer finds that the discharge vane angles varied anywhere from 22-1/2° to 35° with the trend being towards lower angles. Similarly, the writer finds that the entrance vane angles vary from 37° to 40°, the S. S. ZULIA and ESSAYONS having an angle of 45°.

There appears to be a great research need to determine the effect of concentration of the solids in water, the grain size and distribution on the friction factor f. Observations at Fritz Engineering Laboratory, Lehigh University, indicate that when the concentration of solids is low...
(up to 1200 g/l) the resulting mixture is essentially water with solids in suspension, and the solids settle rapidly. However, when the concentration is high (up to 1400 g/l), the mixture appears to be homogenous. It has properties of a non-Newtonian fluid and the solids do not settle rapidly.
11. Herbich, J. B. and Valentine, H. R.
EFFECT OF IMPELLER DESIGN CHANGES ON CHARACTERISTICS OF A MODEL DREDGE PUMP,
Fritz Engineering Laboratory Hydraulics Report No. 277-P. R. 33, September 1961

On pages 88 to 90 of this report are some observations which are of interest in connection with prototype predictions from studies of model performance of centrifugal pumps. According to the authors the prototype performance of a pump is usually better than that of the model. This is due to the difficulty of reproducing all aspects of geometrical similarity in the model, and to the fact that losses due to viscous effects are relatively lower with the flows of high Reynolds Numbers in the prototype. There are no established procedures for predicting the efficiency of the prototype from the model efficiency. However, the Moody formula, used in turbine practice

\[
\frac{1 - E}{1 - e} = \left(\frac{d}{D}\right)^{1/4} \left(\frac{h}{H}\right)^{1/10}
\]

(31)

where

- \(E\) = Efficiency of prototype pump
- \(D\) = Diameter of prototype pump
- \(H\) = Head on prototype pump
- \(e\) = efficiency of model pump
- \(d\) = diameter of model pump
- \(h\) = head on model pump

is sometimes used for pumps.

Another approach, due to Weslicenus, makes use of a chart based on statistical averages of maximum efficiencies of pumps of the same specific speed, with the sizes being represented by design discharges.
12. Iverson, Rolling and Carlson
VOLUTE PRESSURE DISTRIBUTION, RADIAL FORCE ON THE IMPELLER
AND VOLUTE MIXING LOSSES OF A RADIAL FLOW CENTRIFUGAL PUMP,
American Society of Mechanical Engineers Paper No. 59-Hyd-10,
1959

Through tests and analysis on a standard volute type radial flow,
centrifugal pump, the authors tried to determine if:

1. There is a sufficiently accurate relationship for
design purposes between the measured pressure
distribution and the radial force.

2. There is a possibility of an analytical prediction of
the radial force.

3. There is a relationship between volute pressure
distribution and pump head.

The experimental program was run with a 3 inch single stage centrifugal
pump with a specific speed of 1000. Two types of pressure taps were used
in the volute casing, simple side wall taps and static-probe extensions
which were directed into the flow issuing from the impeller. In order to
read the loads on the impeller and the shaft, this part of the pump was
supported only by special supporting beams under the bearing blocks which
were connected with SR-4 strain gages. A similar horizontal arrangement
was devised thus allowing isolation of the three shaft force components:
vertical, horizontal perpendicular to the shaft, and axial parallel to
the shaft.

Experimental Results: - When the volute pressures were integrated around
the casing the resultant radial load and its direction showed fair agree-
ment with the loads and directions indicated by the calibrated mounting
beams, thus indicating a correlation between the volute static head
distribution and the force transmitted to the pump shaft.

Mathematical analysis of the pressure distribution and radial forces with the use of moment balance equations did not give as good results as the experimental data. The general trends were indicated in the analyses, but the authors say that direct comparisons are not possible because of the major factor of internal recirculation from the volute through the wear rings and into the suction of the impeller. This recirculation was appreciable in the facility due to a slight increase in clearance.

Analysis of losses in the volute and subtracting them from the classical formulas for impeller head yielded the pump head curve. In comparison to the characteristic curve of the pump tested, the derived formula gave a curve which was of similar shape, but shifted to a higher capacity. This again was due to the neglecting of the internal recirculation and friction in the impeller.
13. Knapp, R. T.
CAVITATION MECHANICS AND ITS RELATION TO THE DESIGN OF HYDRAULIC
EQUIPMENT,
Engineering, Vol. 173, p. 566, 1952

The author derives the following formula from test results on scale
effects on cavitation inception with regard to hemispherical and ogival
bodies:

\[(K d_i + C_{P_{min.}}) (L V_1)^{1/2} = (K d_i + C_{P_{min.}}) (L_2 V_2)^{1/2}\] (32)

where:

\[K d_i = \text{incipient cavitation number} = \frac{h - \left(\frac{\rho V^2}{2}\right) - p_d}{\rho V^2}\] (33)

\[\rho = \text{density}\]

\[h = \text{total flow pressure, psf}\]

\[p_d = \text{liquid vapor pressure, psf}\]

\[C_{P_{min.}} = \text{pressure coefficient at point of minimum pressure on body} = \frac{p - \left(\frac{\rho V^2}{2}\right)}{\frac{\rho V^2}{2}}\] (34)

\[p = \text{liquid pressure, psf}\]

\[V_1, V_2 = \text{undistributed free-stream velocity of liquid following at infinite distance from the bodies}\]

\[L = \text{representative body dimension, feet}\]
The trailing type, drag suction dredge KAIRYU MARU was recently completed in the Yokohama Shipyards and Engine Works, mainly for the purpose of dredging in the Harbor of Nagoya.

The KAIRYU MARU is a twin-screw and twin rudder ship. The pump room is in the forepart; six section hoppers are located amidships; and the engine room is aft. Both the propulsion engines and dredging pumps are diesel-electric driven, the constant-current control system being adopted. Each hopper is fitted with two hydraulically operated hopper doors.

One steel plate drag-arm, approximately 2 feet in diameter is fitted on each side of the ship. They are long enough to dredge the sea bottom 59 ft. deep at a 45° inclination. Three kinds of dragheads are provided. They are interchangeable depending upon the consistency of the mud.

Principle dimensions are as follows:
Length B.P. 278 ft.-10-1/2 in.
Breadth, moulded 47 ft.-11 in.
Depth, moulded 22 ft.-11-1/2 in.
Draught, designed 18 ft.-4-1/2 in.
Deadweight 3200 tons
Gross tonnage 2647 tons
Hopper capacity 2227 cu. yds.
Speed 12.79 knots
Crew 70
Two dredging pumps of the single stage, single suction, volute type are installed in the pump room, each having a capacity of 1,083,220 gal. per hour at 59 ft. total head. They gave an efficiency as high as 74% at the rated point on the test bed. The power generating equipment consists of two sets of 1000-kw, dc generators driven by two Yokohama M.A.N.G. 8V 40/50 AL type diesels each developing 1800 hp at 360 rpm. The two propulsion motors and the two dredging pump motors are direct-connected, dc electric motors; the rating of each being 900 kw at 300 rpm and 450 kw at 220 rpm.
15. National Bureau of Standards
TESTS OF DREDGE SUCTION BOOSTER,
National Hydraulics Laboratory, Washington, D. C., December 1941

The subject of this report are tests which are performed with the purpose of studying the effect of a suction booster on the capacity of a dredge pump. It is well-known that the energy expended in the suction line by a dredge pump is equal to the sum of the entrance loss at the mouth of the suction pipe, the frictional resistance of flow in the pipe, the velocity head, the energy required to raise the dredge material to the water level, and from the water level to the pump. The material dredged was sand. The variables involved in these tests were the sand characteristics, the depth and speed of dredging, the wear of the dredge pump, the wear of the booster impeller, submergence and lift, booster speed, sand concentration, discharge, velocity in the suction line, the output, the discharge pressure of the dredge pump, the vacuum, the differential head across the booster, the virtual vacuum, and the power.

It was anticipated that the addition of a booster would increase the capacity of the pump. Numerous tests were run from which the following conclusions were derived:

1. For any set of conditions there is a critical booster speed below which the booster acts as an obstruction.
2. For any given concentration a booster increases the velocity of the flow, and the output of sand.
3. For a given sand output a booster decreases the vacuum at the dredge pump.
In general therefore a booster increases the maximum output of sand, increases the maximum concentration of sand, and decreases the vacuum at the dredge under these maximum conditions, as long as it operates above a certain critical speed.
16. Oshima, R.
THEORY OF SCALE EFFECTS ON CAVITATION INCEPTION ON AXIALLY
SYMMETRIC BODIES,
Transactions, American Society of Mechanical Engineers,
Journal of Basic Engineering, Vol. 83, Series D, No. 3,
September 1961

On the assumption that the motion of incipient cavitation bubbles
on geometrically similar bodies is dynamically similar, the relation be-
tween incipient cavitation number and Reynolds number has been obtained
from the dynamical similarity law deduced from the equation of the motion
of spherical bubbles. A comparison of calculated values based on this
theory with experimental data shows good agreement at values of Reynolds
number greater than the critical Reynolds number. Also, by comparing
this theory with the formula by R. T. Knapp, concerning scale effects on
cavitation inception, it is shown that Knapp's formula is a special case
of the present theory.
17. Phillipoff, W.
RHEOLOGICAL CHARACTERIZATION OF SILT,
May 21 to July 2, 1958

The subject of this report is the determination of the rheological properties of Delaware River mud. This investigation was motivated by difficulties encountered in the dredging of the above mentioned mud by the Army Corps of Engineers. The mud was found to have the properties of a Bingham body, and has a yield value of 600 dyn/cm². At the condition of operation of 100 cu. ft./sec. through a 28-inch pipe, whose length is 100 ft., the calculated rate of shear was 80 sec⁻¹, assuming Laminar flow, and the pressure drop 5.2 psi. It was therefore concluded that the pumping operation under the existing conditions did not cause a prohibitively high pressure drop in the suction pipe. The main problem is that undue water is being sucked in addition to the mud dredged.

When one pumps suspensions of solids, such as muds, the properties of the material are very distinct to that of water. The presence of particles of silt increase the viscosity and density of water. There is a yield value below which flow ceases.

The value of σ must be determined experimentally for each concentration of mud. In order that the values obtained experimentally be independent of the dimensions of the test apparatus (in the case of capillaries, the radius, etc.), the plots are made with respect to parameters whose dimensions are those of \( \frac{dy}{dy} \) and \( \tau \). The parameters used for a capillary are

\[
D = \frac{4Q}{\pi R^3} \quad \text{and} \quad \frac{R\Delta p}{2L}.
\]
Both capillaries and rotational viscometers were used. The rotational viscometers were presumably calibrated such that the results would be equivalent to that of the capillaries. As the suction pipe under consideration can be considered as a capillary if the flow is laminar, the rate of shear at operating conditions could be computed from \[
D = \frac{4Q}{\pi R^3} \text{ sec}^{-1}
\] and was found to equal 80 sec\(^{-1}\). Also \(\Delta p = 0.0024 \text{ psi}\).

The experimental results show that the behavior of the mud depended on the so-called previous work history. Also the properties changed under constant load.

The results of the investigation show that the operation is possible. The explanation offered for the large quantities of water suctioned is as follows:

Although the yield stress of the material is well below the stress which occurs at operating conditions, the velocity and consequently the rate of shear falls off rapidly at the intake of the suction pipe.

Therefore the rate of shear at a short distance from the intake pipe is probably low enough to impede the yield stress from being exceeded. Consequently, once a small radius around the pipe has been excavated, until the dredge moves to a different spot, only water will be pumped.
18. Ports and Dredging

FINCOSITA IN HER ELEMENT IN ALL KINDS OF SOIL,
Ports and Dredging, No. 34, 1962

The trailing cutter suction hopper dredger "FINCOSITA", measuring 230' x 40' x 20' and having a hopper capacity of 1600 cu. yd., is a striking example of flexibility and usefulness under all conditions.

A flexible suction tube for stationary dredging, a cutterhead installation and a draghead form part of the dredging equipment of the "FINCOSITA" enable her to operate up to depths of 65 ft. and 55 ft. respectively. The hopper contents can be dumped through hydraulically operated bottom doors or alternatively they can be pumped ashore through a floating pipeline.

To increase the practicality of the "FINCOSITA" even more, the ladder and the suction tube have been constructed so as to enable easy changeover from dredging with the cutter installation to stationary dredging with the flexible suction tube. In this case the suction tube can be lowered while the cutter ladder is retained in its highest position. The flexible suction tube is fitted with a hydraulic swell-compensating installation which insures that the draghead will remain in contact with the bottom, even though there may be a swell of up to 6 feet.
Built by IHC Holland, Shipbuilders and Engineers, at the Hague, the "JOHANNES GAHRS" a diesel-electric twin-screw trailing suction hopper is being used by the Federal German Government to maintain shipping channels.

Dredging operations are controlled from the bridge, partly hydraulically, partly electrically. To meet the requirement that the ship should be able to operate under rough sea conditions with waves up to 3 meters, IHC Holland designed flexible suction tubes and trailing heads with hinged visors, together with swell compensators. The soil is discharged from the hoppers through two double rows of bottom doors, hydraulically controlled from the bridge. Discharge can also be effected through a discharge line to the shore.

Four diesel engines of 1200 hp each and two auxiliary diesels of 600 hp drive the generators, which supply the current for the propulsion motors, the two dredge pump motors, the auxiliaries and the 1000 hp bow thrust unit.

The two dredge pumps, each have a capacity of 353,000 cu. ft./hr. They are driven by 675 kw motors and have cast chrome nickel steel impellers.

Principle dimensions and other particulars are:

<table>
<thead>
<tr>
<th>Description</th>
<th>Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length overall</td>
<td>371 ft.</td>
</tr>
<tr>
<td>Beam</td>
<td>59 ft. 1 in.</td>
</tr>
<tr>
<td>Depth</td>
<td>26 ft. 3 in.</td>
</tr>
</tbody>
</table>
Draught at full load 19 ft. 5 in.
Maximum dredging depth 70 ft. 7 in.
Diameter of suction tubes 34 in.
Speed 14 knots
Hopper capacity 4200
Construction Mainly electrically welded, balanced rudders

Special instruments specially developed for this ship are the bottom profile recorder, screens showing the position of suction tubes and trailing heads and the accoustic dredged groove recorder.

Production of this ship is illustrated by the following complete, typical dredging cycle:

9:00 - 9:10  Hopper filled to overflow
9:10 - 9:45  Hopper filled with sand (78,000 cu. ft.)
9:45 -10:03  To discharging site - remaining water still flowing off until cargo is quite dry.
10:03 -10:05  Discharge
10:05 -10:30  To beginning of dredging run

Thus the 90 minute cycle consisted of:

Filling of hopper  - 45 minutes
To and from discharging site  - 43 minutes
Discharge  - 2 minutes
20. Soth, J. M., Brkich, A. and Stahl, H.

SUCTON HEAD CORRECTION FOR CENTRIFUGAL PUMPS,
American Petroleum Institute, Division of Refining,
Presented at the mid-year Meeting, May 26, 1959

This paper gives the results of suction head tests conducted with a single-stage centrifugal pump handling water and butane at various temperatures. To correlate the data an analysis was made of the parameters involved. The results of these tests and others, when plotted against the parameters involved, showed a reasonable correlation; and, that of the common liquids, cold water required the most suction head to prevent excessive cavitation. On the basis of this data an NPSH (net positive suction head) reduction chart is presented for various liquids.

The authors go on to claim that in a pumping system, no cavitation will take place as long as the local pressures do not drop below the saturation pressure corresponding to the liquid temperature. Operation at the saturation pressure will not cause cavitation since any flashing that would try to take place would have to get its vaporization heat from the liquid which would subcool it. A drop in the pressure level below the saturation value, however, would liberate heat for vaporization equivalent to the liquid specific heat times the change in saturation temperatures corresponding to the pressures. Since the enthalpy was also reduced by the amount of specific volume times the change of pressure the actual heat available for vaporization is the difference, or:

\[ X H_{fg} = C \Delta T - \frac{\Delta \rho V_f}{J} \quad (35) \]

where:

\[ X \quad = \quad \text{fraction of one pound of liquid vaporized} \]
\[ H_{fg} = \text{heat of vaporization} \]
\[ C = \text{specific heat of liquid} \]
\[ \Delta T = \text{change of saturation temperature corresponding to the change of pressure} \]
\[ \Delta P = \text{reduction of pressure imposed on the system} \]
\[ V_f = \text{specific volume of liquid} \]
\[ J = 778 \text{ ft.-lbs./B.T.U.} \]

Changing the NPSH by heating at constant pressure would also give the same expression.

The specific volume with partial vaporization is given by this expression:
\[ V = V_f + XV_{fg} \]

where:
\[ V = \text{specific volume of mixture} \]
\[ V_{fg} = \text{change of specific volume during vaporization} \]

Combining the terms and simplifying,
\[ \frac{V - V_f}{V_f} = \left( C \Delta T - \frac{\Delta P V_f}{J} \right) \frac{V_{fg}}{V_f} \frac{H_{fg}}{V_f} = B \quad (36) \]

where \( B \) is sometimes referred to as the "thermal cavitation criterion", and represents the percent increase in the specific volume when partial cavitation takes place. Thus \( B \) can be used as an index of the extent of cavitation. Using Clapeyron's equation:
\[ \frac{dP}{dt} = \frac{JH_{fg}}{TV_{fg}} \quad (37) \]
Combining and simplifying we get:

\[
\frac{V - V_f}{V_f} = \frac{\Delta \text{NPSH}}{J} \left( \frac{C V_{fg} T}{V_f H_{fg}} - 1 \right) \frac{V_{fg}}{V_f H_{fg}} = B \tag{38}
\]

With most liquids, the first term on the bracket is much larger than one so that a further simplification is possible.

\[
\frac{V - V_f}{V_f} = \frac{C T \Delta \text{NPSH}}{J} \left( \frac{V_{fg}}{V_f H_{fg}} \right)^2 = B \tag{39}
\]

This equation shows that we can now calculate the critical NPSH for incipient or controlled cavitation for any liquid. As an example, suppose the index of extent of cavitation of a pump, B, is 0.65 for cold water and that the minimum NPSH for this extent of cavitation is 16 feet.

Knowing the properties of the liquid, and B, we can now calculate \( \Delta \text{NPSH} \) for any other liquid. Subtracting \( \Delta \text{NPSH} \) from the minimum NPSH for cold water will give the minimum NPSH to produce the same extent of cavitation in the pump using the new liquid.
21. Spronck, R.
LA SIMILITUDE HYDRODYNAMIQUE ET LES ESSAIS SUR MODELES EN
HYDRAULIQUE APPLIQUE (Hydrodynamic Similitude and Model
Tests in Applied Hydraulics),
Annales des Travaux Publics de Belgique, pp. 46-101,
February-April 1932, Waterways Experiment Station Translation
No. 40-18

These articles contain a highly diversified collection of discussions relative to model studies. The following presents the topics of interest discussed:

1. Flow experiments in conduits and channels
2. Research on transportation of soil particles by currents. Forms of river beds with movable bottoms. Improvement of estuaries and ports.

Under item 1 the author indicates that the determination of resistance to the uniform flow of liquids in conduits and channels constitutes one of the most complex problems; and exclusive of several recent mathematical developments of limited scope, it always necessitates experiments. In the case of laminar flow the Hagen-Poisseuille law which was mathematically devised is well accepted. In the majority of practical cases however, the movement is turbulent and the laws governing such flow are quite complex. The old formulae of hydraulics are unrelated to the laws of similitude. Most of the recent formulae refer to the Reynolds Number. One great difficulty is wall roughness; while another important question is the flow from circular conduits to those of other cross-sections, completely or incompletely filled, and open channels.

It appears that the friction coefficient depends, apart from the Reynolds Number, on at least two variables characterizing the "relative
roughness" of walls, one is predominant in the case of irregular wall roughness and the other in the case of wall waviness.

With regard to rough conduits, it is necessary to note that perfect geometric similitude can never be attained and we cannot therefore place an unlimited confidence in the roughness coefficient.

Under item 2 the author points out that the problem of transportation of solid particles is exceptionally difficult and remains poorly explained, in spite of the numerous theoretical and experimental studies. With regard to the possibilities of realizing a relative similitude for the transported material these studies do not seem to be accurately defined at present.
The boiling of liquid in the process of cavitation is a thermal process and is dependent on the liquid properties -- pressure, temperature, latent heat of vaporization and specific heat. During cavitation, damage to the pump performance (head-capacity and efficiency) is caused by the appearance and disappearance of vapor cavities in the low-pressure zone which disrupt the dynamic conditions existing during normal pump operation when the flow is all liquid. To make the boiling possible the latent heat of vaporization must be derived from the liquid flow. This necessary flow of heat from the liquid can only take place when the liquid temperature is above that corresponding to the saturation temperature at the prevailing pressure in the low-pressure cavitation zone. This is the same as saying that the pressure in the cavitation region must fall below the saturation pressure corresponding to the liquid temperature.

The extent of damage to the head-capacity characteristics of a pump depends upon the amount of liquid vaporized and the vapor's specific volume at the existing pressure in the cavitation zone. The effect of the liquid properties on cavitation is easier to observe by comparing the performance of the same pump, at a given speed, handling different liquids. In this way, identical dynamic conditions are assured inside the impeller passages.

One of the more important parameters in dealing with cavitation is the NPSH (the net positive suction head produced by a pump for any
given discharge). Values of the minimum net positive suction head for incipient cavitation can be determined for different rates of discharge, Q; and these minimum values of NPSH can be plotted as a function of Q on a smooth curve. If a centrifugal pump is operated below the minimum or critical value of the NPSH for incipient cavitation, cavitation will result and the total head produced by the pump will be reduced by a value, $\Delta H$.

Usually in order to insure the safe operation of pumps, the pump is operated at a value of NPSH, referred to as $h_p$, which is higher by the value $\Delta h_p$ than the critical NPSH, $h_c$. Thus in equation form:

$$h_p = h_c + \Delta h_p \quad (40)$$

In pump and water turbine practice it is customary to express the cavitation characteristics of a plant as a ratio of the plant NPSH to the total head, designated by the Greek letter sigma

$$\sigma = \frac{\text{NPSH}}{H} \quad (41)$$

plant $\sigma = \frac{\text{NPSH}}{H} = \frac{h_p}{H} = \frac{h_c}{H} + \frac{\Delta h_p}{H} \quad (42)$

or plant $\sigma = \sigma_c + \frac{h_p}{H}$

where $\sigma_c$ is the "critical" sigma.

Until recently, it was generally accepted that for all similar pumps operating at corresponding points on the H-Q curve the value of plant sigma is constant regardless of head. This statement was frequently referred to as "Thoma's Law". Recently however, many cases have been discovered which indicate considerable deviation from the rule $\sigma = \text{constant}$. 
However, laboratory tests prove that when the ratio $\Delta H/H$, where $\Delta H$ is the amount of head lost through the effects of cavitation is held constant, then the critical sigma, $\sigma_c$, will also be constant. This applies for any liquids other than water, where the "corrected" value of NPSH is used for the calculation of sigma. This "corrected" value of NPSH is found from charts for that purpose which are reproduced in the paper for certain fluids.

When a pump is operating at its best efficiency point (bep), the absolute value of the head loss $\Delta H$ due to cavitation is called the cavitation measured effect. When the cavitation measured effect for two similar pumps operated at different heads is equal, Tenot's equation for the plant sigma $\sigma_1$ and $\sigma_2$ applies:

$$\frac{\sigma_1 - \sigma_c}{\sigma_2 - \sigma_c} = \frac{H_2}{H_1}$$

(43)

where $\sigma_c$ is the critical sigma for incipient cavitation, which is the same for both pumps if they are similar. Rearranging:

$$(\sigma_1 - \sigma_c) H_1 = (\sigma_2 - \sigma_c) H_2$$

(44)

or $h_1 - h_c = h_2 - h_c$

(45)

or $\Delta h_p_1 = \Delta h_p_2$

(46)

thus proving that both points have the same margin of NPSH against cavitation. This is the same as saying that the absolute pressure at the low pressure zone is the same; and is by $\Delta H_p$ higher than the saturation pressure of the liquid at the prevailing temperature.
Note that when two similar pumps are operated at the same head 
\( H_1 = H_2 \), both Thoma's and Tenot's rule give the same result: \( \sigma_1 = \sigma_2 \)

Equation (43) can be rearranged into a very useful form as follows:

\[
\sigma_c = \frac{\sigma_2 H_2 - \sigma_1 H_1}{H_2 - H_1} \tag{47}
\]

\[
\sigma_c = \frac{h_2 - h_1}{H_2 - H_1} \tag{48}
\]

Thus when \( \sigma_c \) and \( h_1 \) or \( h_2 \) are determined for any head by experiment, the plant NPSH for any other head can be directly calculated.

If cavitation is to be avoided or kept at the incipient point, all liquids require exactly the same NPSH under the same operating conditions. Under the conditions of limited or controlled cavitation, to produce the same cavitation measurable effect (\( \Delta H/H \)), reduction of NPSH value below that of cold water is possible for certain liquids if suitable cavitation resisting materials are used for impellers.

To find the critical NPSH (for incipient cavitation) for liquids other than cold water, first find the critical NPSH for cold water alone and then add to that the correction for the liquid which is found in plots of empirical data.
Mathematically the principle of least resistance can be justified from what is known as Hamilton's Principle. This is defined as follows:

"If we consider all the varied paths along which a conservative system may be guided, so that it will pass in a given time from a definite initial configuration P to a definite configuration Q, we shall find that the course the system actually follows of its own accord, is always such that along it the action is a minimum."

When applied to centrifugal pumps the principle of least resistance states that at capacities other than normal (best efficiency point) the liquid approaching impeller inlet acquires prerotation such that the angle of attack of the flow at vane inlet tips is reduced. In this manner the hydraulic loss due to sudden changes in direction of flow is minimized. Prerotation at partial capacities causes a reduction of head produced and power required. Prerotation at partial capacities causes backflow at the impeller approach and results in discontinuities in the head-capacity characteristics.

In the ordinary flow pattern of the centrifugal pump at partial capacities, a rotary motion in the direction of impeller rotation occurs in two phases: (a) by direct deflection by the impeller vanes in the immediate vane approach; (b) prerotation further ahead of the impeller by liquid avoiding sudden change in direction by the impeller vanes following the principle of least resistance. The energy gradient in the established flow determines the velocity and direction of flow. The liquid in the impeller
approach cannot be pulled, but only is pushed from behind. The impeller creates low pressure zones equal to the impeller vane number and revolving with the impeller speed.

The effects of prerotation can be minimized by a suitable baffling of the impeller approach. Use of high impeller angles in blowers and pumps magnifies all the effects of the prerotation. As a result, the unstable part of the Q-H curve is extended. Adjustable inlet guide vanes are frequently used to improve the stable operating range of blowers and pumps.
As the title implies, the author is more concerned with the practical rather than the theoretical aspects of dredge pumps. However, the article does contain a few points of great interest considering the scarcity of technical information on the operation of suction dredges.

Laboratory experiments and theory indicate that, when the flow in dredge pump lines is fully turbulent, the loss of head, expressed in terms of a column of the fluid being pumped, will be the same for any mixture of water and solids as it is for clear water.

However, flow in dredge pump lines seldom, if ever, complies with the conditions of complete turbulency. Further, since the density of the mixture varies from the top to the bottom of the pipe and from point to point along the line, attempts to apply theory to operating conditions are not convincing. However, by resorting to the concept of an average coefficient of friction and an average density of mixture pumped, computations may be made which have checked well with observed results. Loss of head is calculated by the formula:

\[ h = K \frac{V^{1.9}}{D^{1.1}} \]  \hspace{1cm} (49)

where \( h \) is the loss of head in feet per 1000 feet of pipeline, \( D \) is the inside diameter of the pipe in feet, \( V \) is the velocity in feet per second and \( K \) is a constant having a value of 0.4 for dredge pipelines containing the usual number of elbows and flexible connections.
The total head or lift against which the pump works is the sum of the vertical lift above the pump, the suction head, and the friction head loss in the pipeline. Of these, the vertical lift can be measured and the friction loss computed by the above formula. Owing principally to the high entrance losses, the suction head cannot be determined by any fixed rule. Usually the digging vacuum on a dredge is between 22 and 23 inches, corresponding to a head of 23 feet of a mixture of average density. This figure also includes the velocity head created by the dredge.
25. Watabe, K.
ON FLUID FRICITION OF ROTATIONAL ROUGH DISC IN ROUGH VESSEL,
Bulletin, Japanese Society of Mechanical Engineers, Vol. 1,
No. 1, p. 69, 1958

Experiments were carried out for cases of discs with smooth and with
rough surfaces. The resistance coefficient was found for variations of
roughness, axial and radial clearances.

The main results obtained may be summarized as follows:

1. The resistance coefficient, $C_f$, in the turbulent
flow region tends to a certain value with increasing
the Reynolds Number, $R_n$, for rough surface.

2. For a certain small value of clearance, the coef-
ficient $C_f$ takes a minimum value.

3. When geometrical similarity is held for the dimension
of the apparatus, $C_f$ depends on $R_n$ only.

4. For incremental values of the resistance coefficient
induced by roughness, the relations $C_{f1} = 2$, $C_{f2} = 4$,
$C_{f3}$ hold.

(Where the subscripts refer to: 1, rough disc
in a rough vessel; 2, a rough disc in a smooth
vessel and 3, a smooth disc in a rough vessel).
APPENDIX II

ANNOTATIONS
1. Bollay, W.
THE THEORY OF FLOW THROUGH CENTRIFUGAL PUMPS,
Theodore von Karman Anniversary Volume, California Institute of Technology, 1941.

Using conformal transformation to transform the n blades of an impeller into a single blade, the flow of a fluid through a centrifugal pump having radial or nearly radial impeller vanes can then be calculated by the application of the principles of the "theory of thin wings".

2. Chatley, H.
ENERGY CONSIDERATION IN DREDGING,
The Institute of Civil Engineers, Selected Engineering Papers, No. 125, Session 1930-1931 and Session 1931-1932.

Describes the energy consumption in dipper-grab-bucket dredging and friction forces acting in pipelines carrying mixed flow of water and solids. Shows the computation of energy required when forcing mixed flow through pipelines. (16 pages)

3. Chatley, H.
THE PRINCIPLES OF DRAG SUCTION DREDGING,
The Dock and Harbor Authority, p. 301. August, 1936.

Refers to above article by the same author, with some new computations, such as: Resistance to cutting; resistance in suction pipe, and computations concerning discharge. 4 pages.

4. Chatley, H.
THE PUMPING OF GRANULAR SOLIDS IN FLUID SUSPENSION,

Discusses the theory of the horizontal transportation of solids in air and water, which is illustrated by computations.

5. Chatley, H.
TERMINAL VELOCITIES,
The Union Institution of Engineers, Journal and Record of Transactions, Vol. 50, p. 81, 1931.

Short article on the transportation of solids in water.

6. De Marchi, G.
THE STUDY OF HYDRODYNAMIC QUESTIONS BY THE AID OF LABORATORY EXPERIMENTS WITH MODELS ON A REDUCED SCALE,

The article gives a comparison between results of laboratory experiments with prototype observations for the purpose of establishing the conditions and limits of validity of the laws of similitude.
7. Durepaire, P.
CONTRIBUTION A L'ETUDE DU DRAGAGE ET DU REFOULEMENT DES DEBLAIS A L'ETAT DES MIXTURES,

In French. Contributions to the theory of dredging and pumping of spoil in a liquified state.

8. Green, J.
PNEUMATIC BREAKWATERS TO PROTECT DREDGES,

A curtain of air bubbles from a submerged pipe will materially reduce heights of waves that are not longer than five times the depth. However, tests of pneumatic breakwaters and models indicate their inability to protect against long ocean swells off the New Jersey shore. A British air distributor might improve performance.

9. Hanoco, C.
EXPERIMENTAL STUDY OF LOSS OF HEAD IN A CLOSED PIPE CARRYING CLAY SLURRY,

A theoretical article about loss of head in a closed pipe carrying a fluid with a heavy proportion of solid materials.

10. Hayward, H.
SIDECAST (BOOM) DREDGING, FOREIGN EXPERIENCE AND LOCAL APPLICATION,
Presented at American Society of Civil Engineers Fall Meeting, 1961.

Data on the first years work of the gigantic 4-pump ZULIA, the largest seagoing sidecast hopper dredge and largest earth moving unit in the world.

11. Hazen, A.
ON SEDIMENTATION,
Transactions, American Society of Civil Engineers, Vol. 58, pp. 58-88. 1902.

Shows charts and computations on sedimentation when solids in suspension settle.
12. **IHC Holland Bulletin Third Series, No. 1, 1959.**
   Published by Industriele Handelcombinatie Holland, The Hague, Holland.

   This informative bulletin contains a few very brief notes on the following topics of interest with regard to dredging:


   A report by Mr. D. Laval, Director of the Port of Rouen, includes a discussion of the changes that have taken place in France during the last fifteen years in the construction and use of dredge pumps. Reports upon tests in 1950 and 1952 on the flow of mixtures in pipes, and wear and tear of pumps. (in French)

14. **Jefferson, T. B.**

   **LONGER LIFE FOR DREDGE PUMPS,**
   *Engineering News-Record*, December 2, 1937.

   Concerned primarily with the practical side to dredging, this article describes how wear, abrasion, and down time for repairs can be significantly reduced by the discriminate use of special high alloy liners inside dredge pumps.

15. **Keller, C.**

   **USE OF AIR IN TESTING HYDRAULIC MACHINES,**

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