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Wear phenomena in centrifugal dredge pumps by D. Rachman, Translation

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Dredge Pump Research

WEAR PHENOMENA IN CENTRIFUGAL DREDGE PUMPS

Translation
by
D. Rachman

Fritz Engineering Laboratory Report No. 310.17

CIVIL ENGINEERING DEPARTMENT
FRITZ ENGINEERING LABORATORY
HYDRAULIC AND SANITARY ENGINEERING DIVISION

WEAR PHENOMENA IN CENTRIFUGAL DREDGE PUMPS

(VERSCHLEISSERSCHEINUNGEN AN BAGGERKREISELPUMPEN)

by

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G E N E R A L   C O M M E N T

After a short introduction into the operation, designing and calculation of centrifugal dredging pumps, the main wear phenomena and wear causes are discussed. Finally research into wear and preventive measures are reported.

1. I N T R O D U C T I O N

1.1 GENERAL SURVEY

The following deals with centrifugal pumps used in floating dredger apparatus such as:

Hopper barge - suction dredger,

Cutter head suction dredger,

Soil suction dredger,

Scourer and

Intermediate Pumping Stations,

mainly used for the transportation of mixtures of the carrier fluid, water, and the various soil types such as clay, mud, sand and gravel, Figures 1 to 5.

Dredging pump installations of 150 to 6000 horse power are in use nowadays, the power requirements depending on type of dredger, type of soil and soil ingredients, scourer pipe length and diameter also the effective gross head. $(H_s + H_d)$.

The consideration of the wear of internal pump surfaces, which are in close contact with the excavated material, limit the mean outer rim speed of the impeller to 22 - 30 m/sec., in special cases approximately 40 m/sec., corresponding to a manometric delivery head of 25 to 90 meters of water. The rotational speeds of the impeller vary between 750 to 180 rpm depending on pump size, cavitation and clogging dangers.

The quoted data permit an economic scouring range of between 300 and 6000 m. with delivery rates of 1000 to 12,000 cubic meters per hour, depending on the mixtures, the characteristic velocities of solids and carrier fluids in the pipe lines and the pipe I.D. when the delivery is through the straight horizontal pipes.

The main difference between a centrifugal dredging pump and a centrifugal pump used to pump clean fluids are that:

(a) The impeller vane passages must not contain contractions in between suction and delivery in order to prevent the trapping of solids in the passages and the accompanying detrimental affects.

(b) The number of impeller vanes must be kept small, (1 to 5 vanes) to ensure the freeest possible passage of the contained solids. The entry area between two adjacent vanes is thus kept as large as possible, (taking care to use the correct inlet vane angle and avoiding cavitation sensitivity).

(c) The contour of the impellers outer faces have the minimum of curvature in meridian section to facilitate small side clearances, simplicity of manufacture, the use of wear - resistant lining as well as quick and cheap repairs (deposits - welding, hardfacing).

(d) The closest proximity between impeller O. D. and volute is reduced.

(e) Highly efficient, streamlined impeller vanes and flow passages are exchanged for the simplest possible shapes to facilitate quick and economic replacement of worn-out linings.

The above, besides showing the main differences, also lists some of the principle demands made on a centrifugal dredging pump.

1.2 POINTERS FOR THE DESIGN & CALCULATION OF CENTRIFUGAL DREDGING PUMPS

The laying out and calculation of centrifugal dredging pumps follow for the most part, the standard procedure for the usual types of centrifugal pumps. The principles generally used for the pumping of clean fluids must, however, be modified to allow for the solid content in the fluid of dredger pumps. The break-through in theoretical analysis is just beginning due to the multitude of variables involved.

The designers must therefore fall back on empirical knowledge and practical experience.

The first major difficulty in laying-out a centrifugal dredging pump lies in estimating and obtaining the correct manometric delivery head to satisfy one, or at the most, several operating conditions.

Besides, the usually known delivery height above water level, at least a sufficiently accurate statement can be made on the minimum suction head that can be tolerated at the pump. It can lie between 6.5 and 8.5 meters of water based on cavitation considerations. On the other hand, it is very difficult to calculate with any certainty the head lost in the suction pipe.

The loss is principally dependent on:

(a) The type of bottom and soil composition besides the shape of the various solids.

(b) The solids content of the carrier fluid.

(c) The pipe I.D.

(d) The velocity ratio between solids and fluid

(e) The velocity of the fluid

(f) The pipe joints and the pipe straightness

(g) The quality of the pipe bore.

The estimation of the head lost for silt-water mixtures in pipelines has been the subject of much research in France, Russia, U.S.A. and Germany (1).

From the results it is possible for given conditions, to pre-estimate the manometric delivery head, i.e. the energy per cubic meter of mixture, with sufficient accuracy.

Next the design of the main elements of the dredger pump, the impeller and volute must be adjusted to give the required output with the highest possible efficiency.

The performance of a centrifugal pump for clean fluids can only be pre-estimated with required accuracy at the design condition and a characteristic curve in its neighborhood. For a more extensive characteristic curve, results from models, or full scale tests, are required. This is even more true in the case of dredging pumps.

The prediction of the combined performance of the dredging pump, driving motor and pipeline, is helped by model tests using clean fluids and finally modified by results from full scale tests. The results from the full scale tests are compared with the predictions from the model tests and then corrected, Figures 6 to 8.

It is seen that only small differences exist between performance curves obtained from tests with pure fluids and fluids containing solids of usual concentrations. The exception is where the mixtures contain sludge or geleous sediment, which because of their thixotropic properties follow different laws (2).

1.3 DESIGN OF MANUFACTURED CENTRIFUGAL DREDGING PUMPS

The present day production designs of dredging pumps are mainly four, but for a few small variations. The pumps are chiefly:

(1) Cast Construction with thickwalled casing and an impeller of wear-resistant cast steel, Figure 9 to 10.

(2) Welded construction of the casing containing a lining of wear plate and a cast impeller, Figure 12 to 14.

(3) Welded construction of the casing inlaid with rubber and either fully rubber lined welded or cast impeller, Figure 15.

(4) Cast or welded casing containing an inner Wear-Casing of chilled cast iron and a cast impeller, Figure 16.

There are many reasons why any one of the listed designs may have advantages over the other. Realization of the large size of such dredging pumps leads to the conclusion that a welded construction is preferable to the otherwise necessary large single casting, because:

(a) No Modelmaking is required, i.e. no model costs;

(b) Smaller weight;

(c) Lower prices;

(d) Simpler, more economic replacement of worn-out wear-linings where the wear rate is highest;

(e) Lower maintenance costs.

2. WEAR PHENOMENA IN CENTRIFUGAL DREDGING PUMP

Practically all the parts of the pump which come into contact with the dredging mixture, are subjected to abrasive wear to varying degree. The most affected parts are:

(a) the impeller,

(b) the volute,

(c) the suction and shaft seals.

Under similar hydraulic conditions the composition and type of solids carried decide the wear rate in the pump. The most important properties of the bed materials are indicated versus numerical indices (3). Besides the density of the bed material, the grain-distribution plays a major role, Figure 17. The density is based on the density of the non-porous solid mass, and in case of inorganic bed material lies between 2550 kg/m^3 to 2850 kg/m^3. For quartz it is 2650 kg/m^3 and clay 2720 kg/m^3. Sand and gravel which usually cause very intense wear in dredging pumps, require a further knowledge of grain size and the uniformity of grain distribution. A measure of this is the degree of irregularity:

$$U = \frac{d_{60}}{d_{10}}$$

where d_{60} and d_{10} are grain diameters by 60% and 10% straining respectively.

Expressed numerically it means:

$U < 5$	uniform bed,
$5 \leq U \leq 15$	non-uniform bed,
$U > 15$	very irregular bed.

In transportation of sand, the roughness factor and the hardness of the solids become important (4). Generally it may be said that wear can be controlled when sludge is pumped. For grain in the sift size range, the wear can be seen to increase with increasing grain size, increasing hardness and sharpness. This fact is largely based on the much higher absolute and relative carrier velocities needed for increasing grain sizes to prevent the solids from depositing in the pipelines. High velocities further mean increasing losses and higher delivery head, i.e. higher impeller speeds and greater wear.

At the present there are no known materials which can fully resist abrasive wear particularly as encountered in transportation of sand or gravel water mixtures.

In special instances when especially abrasive bed material is being pumped some parts of the casing, seals or even the impeller must be replaced after 40 to 60 hours operations.

In the case of such short service life the economic operation of hydraulic transportation of solids is not dictated by the price of the worn out components, but by the close down of the whole plant, brought about by their replacement.

It was believed that this tragic fact could be counteracted by even more suitable pump materials or rather material - pairing. It was only later that attention was focussed on an improved streamlining of the pump as an effective counteraction against wear. Older pump designs only reached an optimum efficiency of 45 to 50%. Conversely the optimum efficiency of normal, even faster running centrifugal dredger pumps of approximately the same size, reached between 65 and 70%.

The lower efficiency of older designs is without a doubt caused by adverse shaping and hydraulic flow conditions in the pump, which in turn brought about increased wear. It was therefore unavoidable that improvements in the streamlining of the components to reduce wear, came as a follow-on to the metallurgical wear prevention measures.

2.1 IMPELLER - WEAR

Impellers of centrifugal dredging pumps were earlier almost entirely of the open or half-open type, while nowadays the closed-type impeller designs have won through, Figure 19. There are chiefly three zones on the impeller which are subjected to high wear rates:

(I) Impeller eye and vane inlet edges.

(II) Impeller exit, i.e. vane exit.

(III) Impeller sides at suction end.

2.1.1

At the impeller eye the first contact and energy transfer between the mixture and the impeller vanes take place especially at the impeller edges. There is an appreciable velocity drop between carrier fluid and solids, depending on the operating conditions of the plant and the bed material.

The edge of the impeller eye is highly worn depending on the impact angle between solids and the edge. When the angle between the approaching stream and the impeller inlet vane angle is large, the wear rate is high; large grain sizes causing a bending over of the edges, Figure 20 to 22.

The wear and tear phenomena is largely caused by the fact that the vane inlet is kept small, in order to obtain a large inlet area between each pair of vanes, smaller than the relative approach angle of the transported solids. The advantages of a large inlet area are brought at the cost of increased wear of the vanes at entry. A useful compromise must therefore be sought.

2.1.2

At least for closed type impeller, the passage between inlet and outlet are usually relatively evenly worn, while at exit both the edges of the impeller sides and the vane tips show a much greater wear rate, Figure 23 to 25. The vane tips are in particular badly worn where they join the impeller sides. This phenomena is caused by the fact that the solids in contact with the pressure side of the vanes crowd over to the transition zone between vane and impeller sides. When the pump is running at part-load output, the extreme impeller width encourages diffusive and backward flow in the transition zone between impeller O.D. and the volute, thus increasing the local wear rate.

The high wear rate of the outer impeller edges is mainly caused by vortices induced by the impeller sides.

2.1.3

The increased wear on the suction side of the impeller side walls compared to that on the bearing side, is primarily due to leakage of mixture between the volute and the suction through the end clearance space. The pressure difference between the suction and the delivery must be effectively sealed. This pressure drop causes a leakage depending on the effectiveness of the suction gland. This continuous circulation of solid particles through the clearance space on the suction side of the pump and across the suction gland, resulting in a very high wear, should be avoided where possible.

For the clearance space on the bearing side the same recommendation is valid. Here the problem is much simpler since the seal runs against a very much smaller diameter, and the surface velocity is correspondingly smaller. The pressure drop is also much smaller on that side.

2.2 CASING WEAR

The volute is chiefly subjected to high wear in the area of close proximity, i.e. between the tongue and impeller O.D. This is due to the fact that the pressure difference at the impeller O.D. relative to the operating pressure, is highest across the tongue with a corresponding flow through the clearance space. This increased local velocity of the solid carrying fluid brings about high wear of the tongue and nearlying areas. Figure 26-29.

2.3 SEAL WEAR

The causes of the seal wear have already been mentioned in part 2.1.3. In the practical operation of centrifugal dredging pumps, the sealing glands on the suction side of the enclosed-type impellers are the cause of many problems. There are many designs in use of which only a few will be shown. Figure 30 to 34.

The following are the basic differences depending on the operation and arrangement:

(a) Contact Seal,

(b) Seal free from contact.

In both cases the seal-grroves can be either arranged radially or axially. Axial seal grooves are mainly used for contactless seals which can be manually adjusted. Radial seal grooves are used with self-adjusting seals.

A satisfactory operation life of the seal whatever the design is - from experience - only to be had, when the seals are kept free from solid particles by the use of a sealing fluid. Without the use of sealing fluid, the high rubbing speeds, the high interface pressure and the continuous flow of mixture through the clearance space across the seal, wear it out in a short time.

3. MODEL TESTS ON DREDGER PUMPS

A program of research work was undertaken jointly by the Institute of Design and Hydraulic Flow Machines and Orenstein Koppel and Lübeck Engine Works, Ltd. This included the design and construction of a special test rig and the testing of various dredger pump models with the object of establishing a reliable method of casing the impeller design for large dredger pumps. This rig, shown in Figure 35 was used to establish:

(1) V, H Characteristics (Throttling curves)

(2) Output and efficiency characteristics

(3) Cavitation characteristics

(4) Axial thrust conditions

(5) Pressure and velocity distribution in pump casing

The numerous test results referred to characteristics shown in Figure 7 and the exhaustive results of the Pressure and Velocity Distribution on tongue Z 4 illustrate the performance of two bladed impellers of different diameters used with it. See Figures 36 to 41.

These diagrams illustrate the following details under subtitles:

(a) Position of measuring point in ψ and $\emptyset$ diagram.

(b) The flow round the tongue

(c) The pressure distribution on the tongue in a central radial plane between the housing and walls.

(d) Pressure distribution at runner outlet, measured on both end walls of the casing and the pressure distribution round the circumference of the spiral, midway between the end walls, see also Figure 41.

The nomenclature is as follows:

$\emptyset = \dfrac{4\,\dot{V}}{D_S^2\,\pi\,u_S}$	Flow coefficient
$\dot{V}$ =	Rate of flow
D_S =	Impeller suction diameter
u_S =	Top speed at impeller inlet
$\psi = \dfrac{2\,g\,H}{u_2^2}$	Head coefficient
H =	Delivery Head
u_2 =	Top speed at impeller outlet

g = Acceleration of gravity

η = Efficiency

p_Z = Pressure at a point on tongue

P_{du}'' Pressure at pump outlet

The essential results of this investigation show that:

(1) The increase in the tongue clearance of the impeller produces a smaller reduction in local pressure and hence lower speeds at the tongue under any flow condition.

(2) A blunt rounded tongue shows by comparison with a slender tongue a smaller fall in pressure due to velocities.

(3) The tongue shape has an insignificant effect on efficiency.

(4) The position and shape of the ψ, $\emptyset$ Characteristic (Throttling curve) and the efficiency are considerably affected by the tongue clearance.

The above findings were supplemented by measuring the abolute velocity of the flow in the spiral casing in both magnitude and direction at a number of points, so that the final pressure and velocity distribution in the dredging pumps casing would be sufficiently clear, in order to decide on an optimum hydraulic shape and design, bearing in mind the ncecssary reduction in wear.

4. DESIGN AND MATERIAL CONSIDERATIONS LEADING TO REDUCTION IN WEAR ON THE BASIS OF RESULTS OF MODEL, OPERATION PRACTICE AND WEAR RESEARCH

4.1 GENERAL POINTERS AND FUNDAMENTALS

All observations and measurements carried out so far in connection with wear of dredger pumps in operation do not lead to a generally applicable conclusion about the reasons for wear and its progress. This is due to the numerous variables encountered in operation. The amount of available data is inadequate to come to any far reaching conclusions. In practice the unit is operated principally on an economic basis, which leaves little room for systematic wear research.

The constantly and appreciably changing operating conditions make it difficult to present a true life story of the pump parts subjected to wear, to diagnose the trouble and subsequently to produce a further movement in the unit. There are few units with small changes in duty and operating conditions, so that observations made under such circumstances are applicable only for the specific circumstances at the time.

The wear research, in Germany, has been carried out above others by Wellinger (ref. 5-15), Uetz (7-16) and Wahl (6-18). The aim of this work was to devise test conditions which would enable the drawings of most generally applicable relationships and conclusions. Of particular interest is the relationship between resistance to wear of a material and wear conditions, as well as specific properties of the materials. Here there is a hope of gaining a deeper insight by a searching investigation of various specific cases of wear.

In the meantime the designer has to face the problem of selecting a suitable material for wearing parts. In general he has to take account, apart from purely technical considerations, also of economic factors. For example in the case of subsitution of wear resisting material, will the increased resistance offset increased materials and machining costs? A material 1 is only then more economical than another 2 when

$$\frac{1}{E_1} P_1 < \frac{1}{E_2} P_2$$

where E = wear resistance, P the price including assembly and delay as well as disturbing production.

The correct choice of an economic material demands therefore an adequate knowledge of wear characteristics for any particular application. Unfortunately the wear characteristics cannot be defined by a single characteristic quantity, since it is very closely governed by the operating conditions. In the case of Metal-Mineral wear, which takes place in dredger pumps, there is a general tendency to show a correlation between the wear V of a material and hardness H_S of the particles producing it, such as shown in Fig. 43 (ref. 16) for the case of dry sliding wear.

It is worth noting that the $V = H_S$ characteristic rises steeply from a low value at an H_S corresponding roughly to the hardness of the material to a maximum, and then forms a slightly down sloping curve. The hard material wear remains at a minimum for up to very hard abrasive particles, whereas in the case of relatively softer materials the wear is mostly in the upper range even for relatively soft abrasive materials.

The wear consistency of various materials is also subject to large variations, Fig. 44, and the question whether the insertion of new materials will be economical depends not only on the hardness of the material itself, but also on the hardness of the abrasive particles.

Since the spoil varies in quality and composition from place to place, the economic value of a material will depend on the location of operations at any one time. It is therefore not permissible to use experience obtained at different locations with other spoils without further considerations. The effect of jet striking angle in jet erosion was first investigated by Wellinger (ref. 9 and 13).

The most important findings of these tests are shown in Fig. 45 to 46. These provide a number of directly useful qualitative pointers for a number of wear problems. Fig. 45 shows that the speed of erosion is highest for a small jet incidence (i.e. angle between the jet axis and eroded surface) and is constantly reduced up to a direct impact wear (90^{o}).

In the case of hardened or basically brittle materials, the rate of erosion is smallest at small incidence angles and increases with the increase of the angle of incidence. An exception to this behavior are the soft and ductile materials, which have a maximum rate of erosion at medium striking jet angles, i.e. St. 36, Fig. 45. This behavior is particularly born out by Fig. 46 where the wear ratio W Δ Y of a number of materials is plotted, using St. 37 as a standard of comparison, such as function of the jet angle. (Wellinger - Diagram). It is worth noting that with the change from sliding erosion to impact erosion $\alpha = 0^{o} - 90^{o}$ there is a complete reversal of material properties in the sense explained above.

The brief and incomplete indications show that the wear date available from previous wear research, which was quite numerous, can be used with ones experience and knowledge to draw far reaching conclusions.

4.2 REDUCTION OF WEAR BY IMPROVED SHAPE AND MANUFACTURE

Model tests and wear observations on large pumps in operation have lead to the following modifications in shape and construction which resulted in improved design of new dredger pumps.

Runner

1. Shrouded design.
2. Optimum blade angle, especially at inlet.
3. Extended shrouds, especially at outlet.

Housing

1. Blunt, efficiently shaped tongue.
2. Increased tongue clearance at the maximum impeller diameter.
3. The closest clearance between the impeller shrouds and housing.
4. An efficient proportioning of runner and volute dimensions with reference to the required operation conditions.

Seals

1. Clean water fed suction end seal for closed type of impeller.
2. Self-adjusting suction seal lip.

4.3 REDUCTION OF WEAR BY SUITABLE MATERIALS OR MATING OF MATERIALS

Using as a basis, the results of wear research, a full scale dredger pump was used in operation at a fixed location with a relatively constant duty, so as to control operating conditions, to carry out time tests of various main wearing pump parts, in particular of the suction seal. The results collected during these tests were applied in the current production and have led to the improvement of some of the economics of wearing parts in ratios of 1 : 3 up to 1 : 10 as shown in Fig. 47. The general deductions were as follows.

Runner

1. Thickwalled cast alloy steel. Additions Cr, Ni, Mn, Mo and V. Suitable for molded deposition repair.

2. Welded design completely rubber covered. Rubber skin thickness 30 mm. Shore-Hardness 50 - 65^{o} (useful for pure sand dredgers, sensitive against sharp edged solids).

Housing

1. Welded with an interchangeable wear lining of special steel plate E 22. Alloy composition Cr, Mn, Si. Plate thickness 20-30 mm.

2. Welded design with a vulcanised rubber lining. Lining thickness 30 to 100 mm. Shore-Hardness 50 to 65^{o} (limitations as for runner, manufacture - welded and fully covered).

3. Cast design - as for the runner.

Seals

1.	Material pairing	Rubber/Metal with sealing water feed.
	Rubber -	Shore-Hardness 40^{o} to 60^{o}.
	Metal -	Ferrous Chromium casting (1.4%C, 30%C).
	Arrangement -	Sealing gap may be axial or radial, running in contact or free.
2.	Material pairing	Rubber/Rubber with sealing water seal.
	Rubber -	Shore-Hardness 45^{o} - 60^{o}.

Arrangement - Axial - free of contact.

Adjustment - Manual.

5. SUMMARY

In spite of the fact that suction-dredgers fitted with centrifugal pumps, date back as far as 100 years, and the economics of a dredger pumping plant was strongly influenced by the working life of the wearing parts, up until today the technical advances in the dredger pump design with regard to wear depended exclusively on experience with a few units in operation.

All the available information, up to date, dealing with dredger pump wear including those presented in the paper dealing with general erosion aspects are applicable to strictly defined environmental conditions. In spite of this they may provide useful guidance for dealing with other and similar cases.

It may therefore be concluded that while the quoted erosion data are not sufficiently quantitative and indefinite, covering a few generally applicable relationships, they should provide a basis for future investigations and researches as well as the necessary inspiration.

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D I S C U S S I O N

DR. E. FASCHALLEGG

In a thorough research into all operating conditions and loading conditions should the effects of radial forces, which exist in volute pumps, not be neglected. Figures 36 to 41 of the article show amongst other things the pressure distribution in the volute along the impeller peripheri. If this pressure distribution is integrated over the peripheri, a resultant radial force will result..

For the design conditions of the volute, these forces disappear largely, while in the case of part-load or over-load, large radial forces can occur. A suction dredger pump can but seldom operate only at one operating conditions, it must rather operate safely over a large part of the characteristic curve.

It is now possible to keep the radial force distribution within reasonable limit over the whole operating range by a proper choice of casing dimensions.

Fig. 1 shows the radial force variation for the same impeller in two different volutes. If the pump operates in the range of high radial forces, high shaft loads and one sided bearing loads result. The disadvantages are obvious.

Titles for photographs not reproduced in this translation

Fig. 1 Diesel electric hopper barge suction dredger with traction head. Has two-side-suction pipes and two centrifugal dredging pumps.

MAIN DATA

Overall length	113 meters
Wide across bows	18 meters
Side height	8 meters
Draught laden	5.9 meters
Dredging depth	21.5 meters
Speed	12.6 knots
Hold-capacity	2800 cumeter
Installed diesel motor output	6250 hp (metric)
Nominal discharge of mixture from one pump	9500 cu m/hour
Nominal delivery head of one pump	15 m water

Fig. 4 The machine room of the dredger shown in Fig. 3, showing the suction side of both the dredging pumps and the injection pump.

Fig. 5 Diesel driven intermediate pump

Drive output	640 hp (metric)
Manometric nominal delivery head of pump	55 m water
Nominal delivery of mixture	1,800 cu m/hr
Rotational speed of pump	535 rpm
Suction and delivery fittings	400/400 mm dia.

Fig. 10 A pump like in Fig. 9 with the suction cover off.

Fig. 13. Dredging for a cutter head suction dredger with wear-plate lining and of welded construction.

Drive output	200 hp (metric)
Drive rpm	750
Man. nominal delivery head	22 m water
Nominal discharge	1300 cu m/hr.

Fig. 14 Dreding pump same as before in Fig. 13, only smaller

Drive hp	150
Drive rpm	600
Nominal delivery head	20 m water
Nominal discharge	1000 cu m/ hr.

Fig. 16 Dredger pump with a wear casing of chilled cast iron inserted.

(a) Suction cover,
(b) Impeller,
(c) Wear-casing,
(d) Outer pump casing.

Fig. 19 Enclosed, 3-vane impeller of a dredging pump.

Fig. 20/22. 5-vane dredging pump impeller of cast steel construction. Strongly worn, partly bent-over vane inlet edges, mainly caused by unsuitable angle of approach of the gravel - water - mixture.

Nominal data for pump:

Drive output.........	300 hp/metric
Drive rpm	600
Man. delivery head...	50 m water,
Discharge	1,000 cu m/hr.

Fig. 23 5-vane dredging pump impeller of cast steel construction.

Uniform wear of the vanes and the side plates, indeed increased on the suction side, caused by the transportation of medium sand - water - mixture after a transportation of 300,000 cu m of sand.

Nominal data for the pump

Drive output...............	1,200 hp (metric)
Drive rpm	500
Man. delivery head	54 m water
Discharge of mixture	3,600 cu m/hr.

Fig. 24 & 25 Same impeller as in Fig. 23, with increased wear at the joining of vanes to side plates and on the outside of the side plates.

Figs. 26/28 Wear phenomena at the tongue, on the bearing side of the wear plate and impeller (Pump as in Fig. 23).

Fig. 29 Worn out piece of lining from the volute just behind the tongue (Pump as in Fig. 23).

Fig. 34 Impeller with a damaged rubber lined sealing bush of Fig. 31.

Fig. 35 Partial view of the model testing for dredging pumps in the Institute of Hydraulic Machines of the Technischen Hochschule, Hannover.

(a) Suction pipe,
(b) Multimanometer,
(c) Robot camera,
(d) Pressure pipe line,
(e) Weighing machine for the d.c. dynamometer,
(f) Model pump with suction cover from perspex.

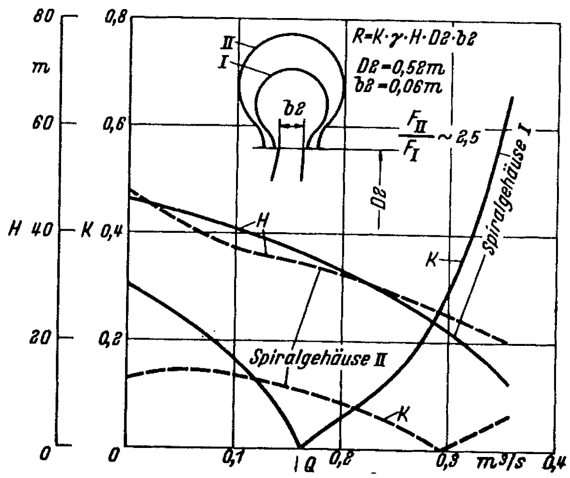


Fig. 1 Discussion - The variation in radial forces for the same impeller in two different volutes (Volute I and Volute II).

Note: This figure accompanies the discussion by Dr. E. Faschallegg

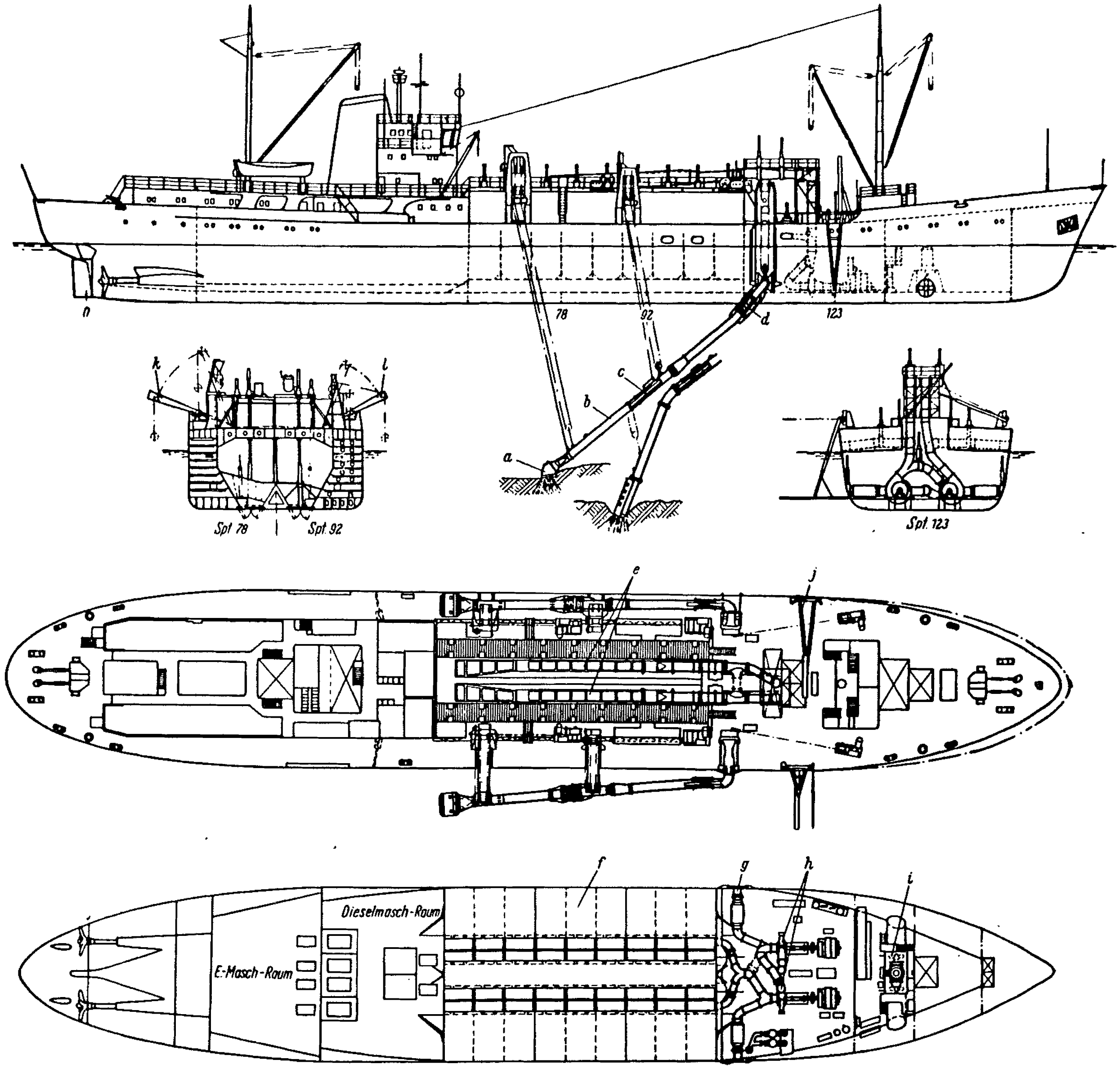


Fig. 2 General Plan of Fig. 2

(a) Fraction head
(b) Side suction pipe,
(c) Middle joint,
(d) Suction pipe bend joint,
(e) Loading channels,
(f) Hold,
(g) Suction pipes of dredging pumps,
(h) Cemtrifugal dredging pumps,
(i) Bow-wave diffuse,
(j) Loom for sea-bed charting device,
(k) Loom for traction head gear.
(1) Loom for middle joint.

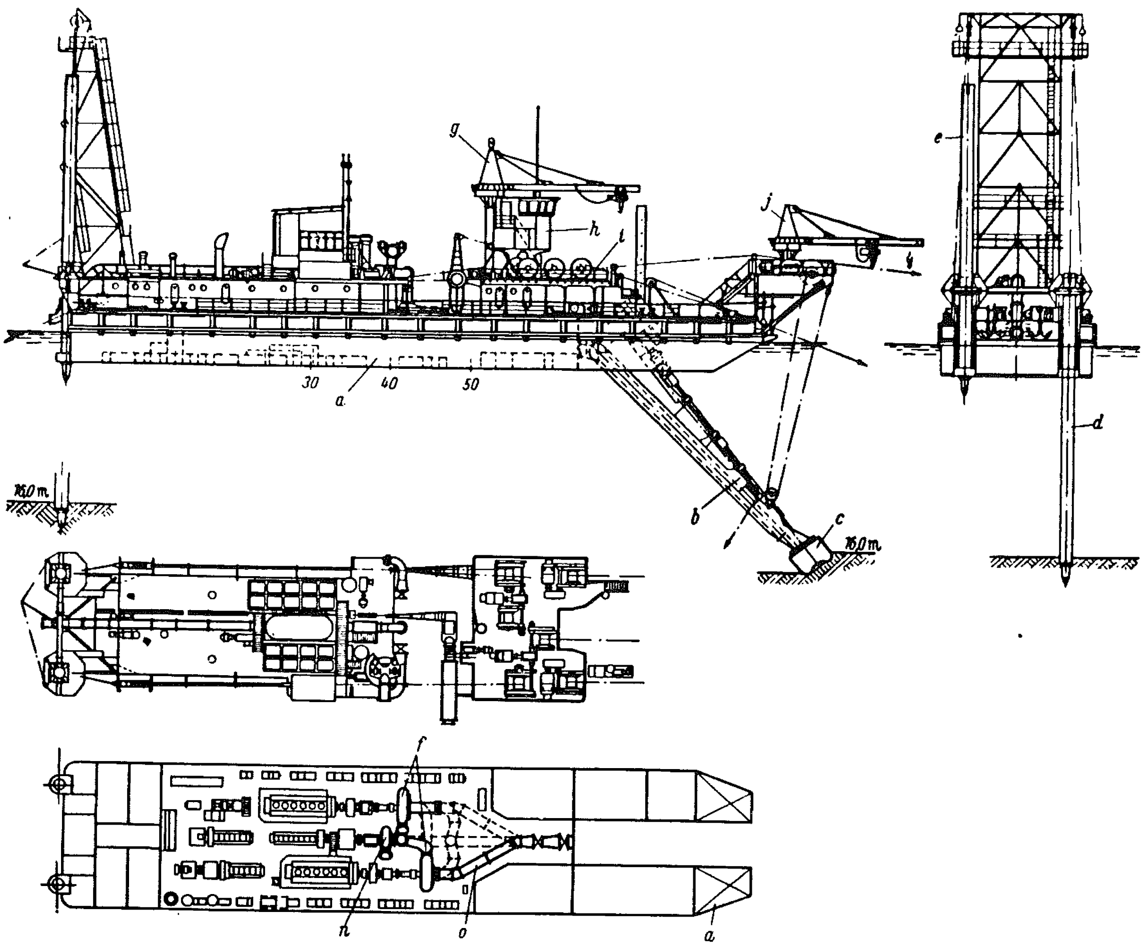


Fig. 3 Diesel driven cutting head suction dredger with shute-loading and emptying device (without self-propulsion).

Main Data

Deck-Length	51 m
Bow width	11 m
Side height	4 m
Draught	2.36 m
Dredging depth	4-16 m
Output of each pump drive	1,260 hp (metric)
Nominal discharge for each pump	5,400 cu m/hr
Nominal delivery head for each pump	43 m water

Dredger Lay-Out

(a) Float,
(b) Cutting head loader,
(c) Cutting head,
(d) Turning pile (lowered)
(e) Turning pile (lifted)
(f) Dredger pumps,
(g) Mid-ship winch,
(h) Control cabin with a central control,
(i) Winch platform,
(j) Vorderbock crane,
(k) Water injection centrifugal pump,
(l) Suction pipe line for dredging pump.

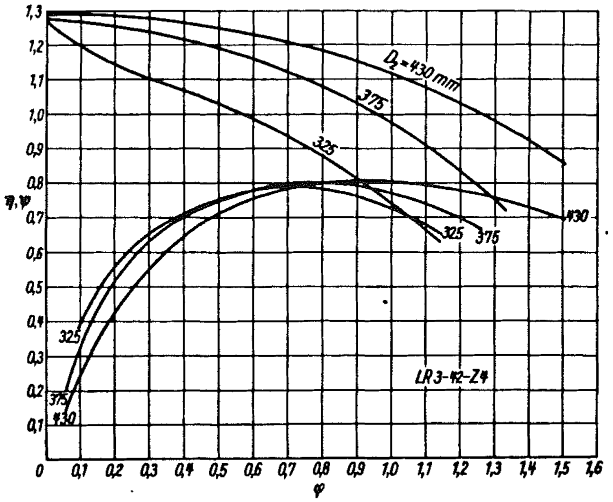


Fig. 6 Characteristic curve of a model dredger pump with 5-vane impeller of various o.d.

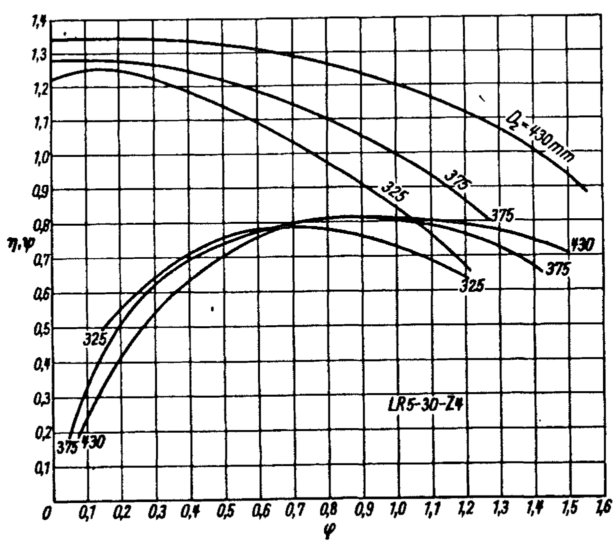


Fig. 7 Characteristic curve of a model dredger pump with 5-vane impeller of various o.d.

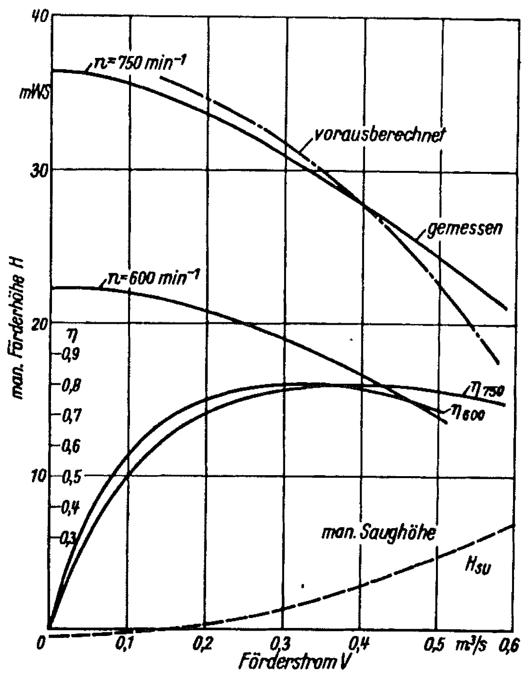


Fig. 8 Measured characteristics of a dredger pump pumping water compared with a predicted throttle curve.

Axis H manometric delivery head,
V discharge

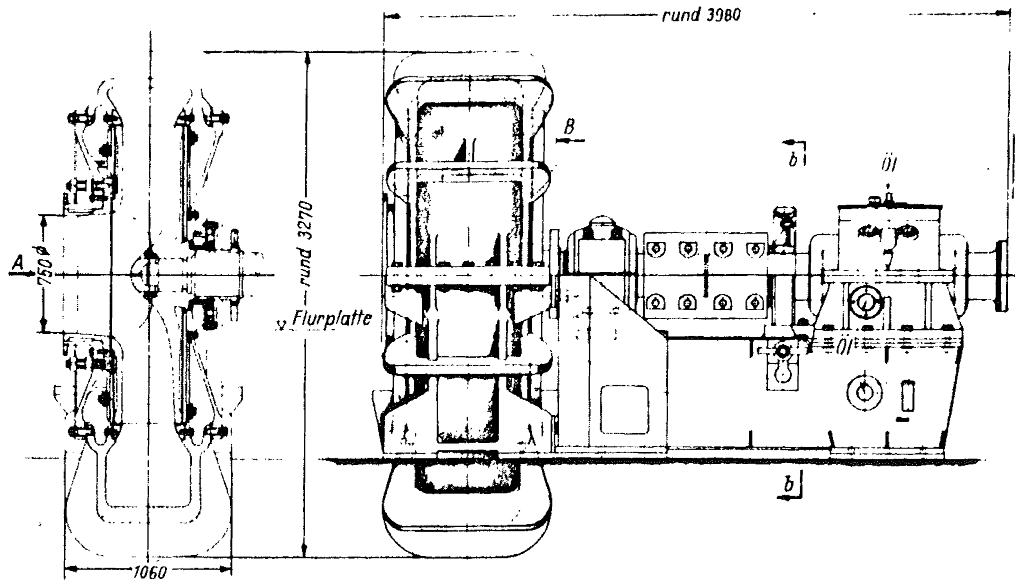


Fig. 9 Centrifugal dredging pump of a cutter head suction dredger of casting construction.

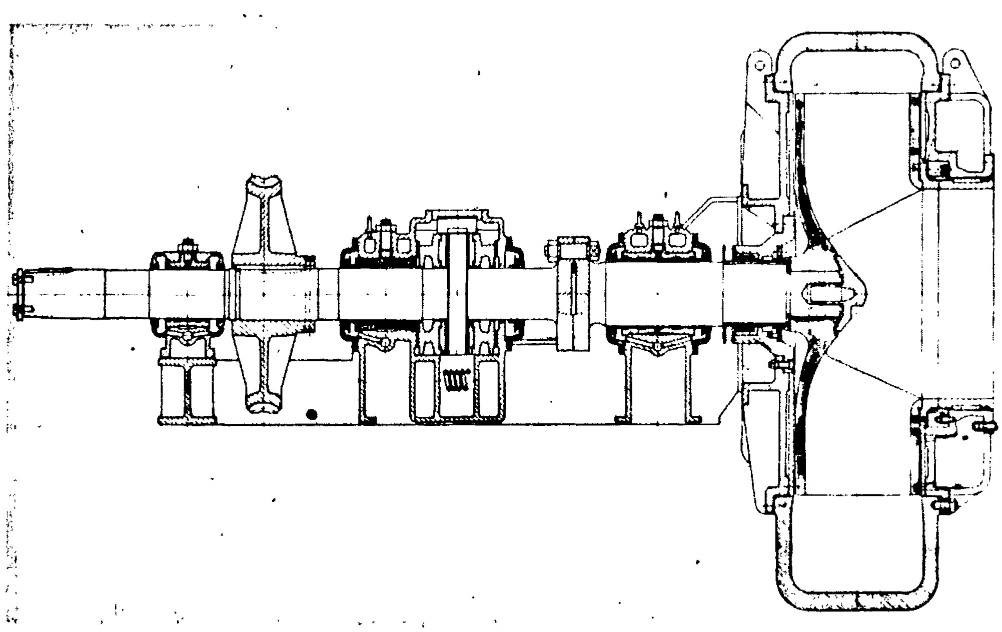


Fig. 11 Dredging pump of a hopper barge suction dredger of cast construction (for output data, see Fig. 2).

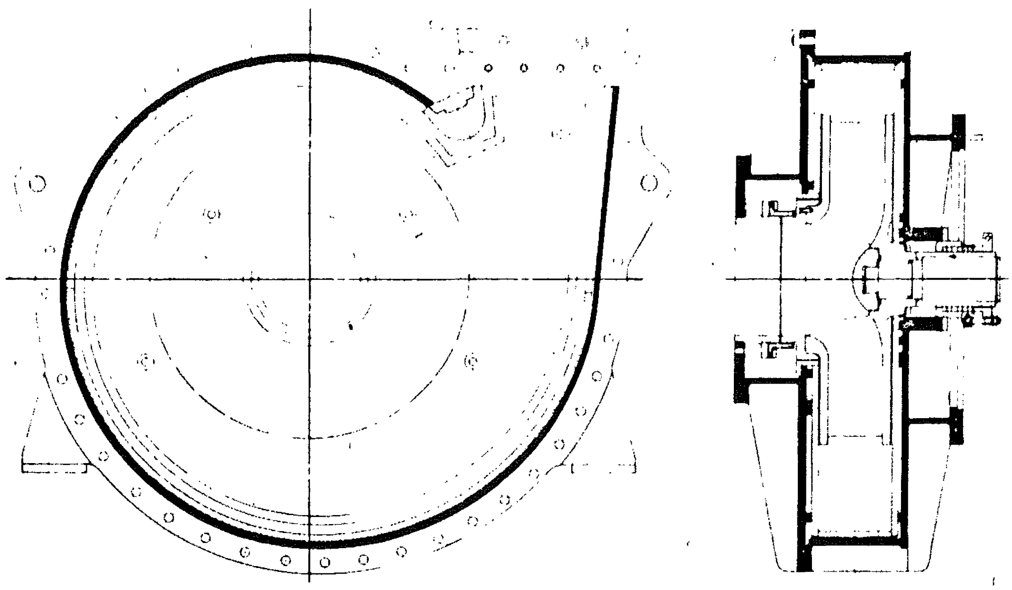


Fig. 12 Dredging pump, which can be dismantled, of welded construction with a wear-plate lining, for a cutter head suction dredger.

Drive output................	450 hp (metric)
Drive rpm	550
Man. nominal delivery head..	48 m water
Nominal discharge	1450 cu m/hr.

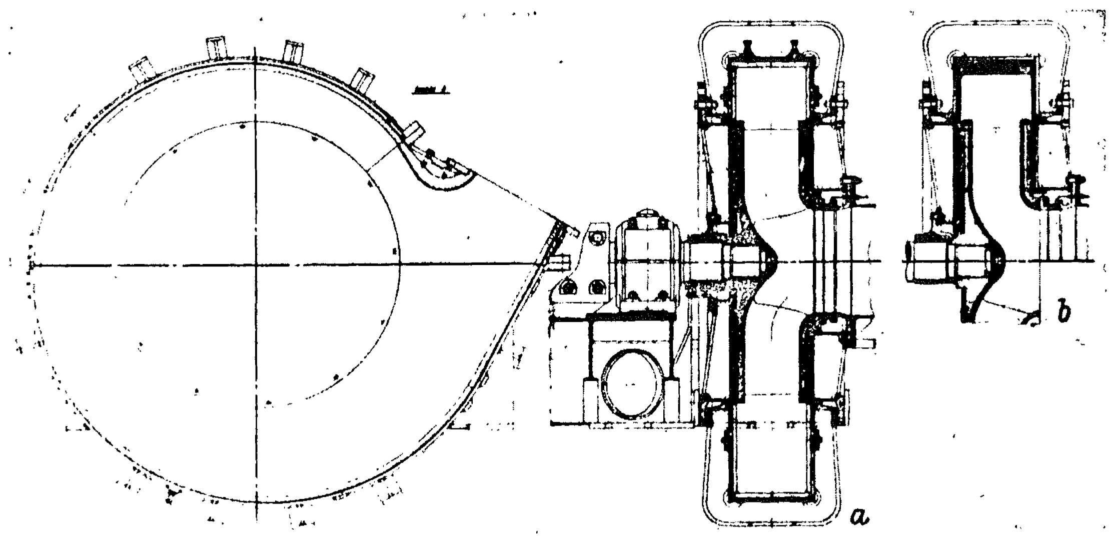


Fig. 15 Dredging pump for a cutter head suction dredger of welded construction with (a) wear-plate lined casing and cast steel impeller, (b) rubber lined casing and rubber lined welded construction impeller.

Drive hp	1260 hp (metric)
Drive rpm	270
Nom. delivery head.........	43 m water
Nom. discharge	5400 cu m/hr.

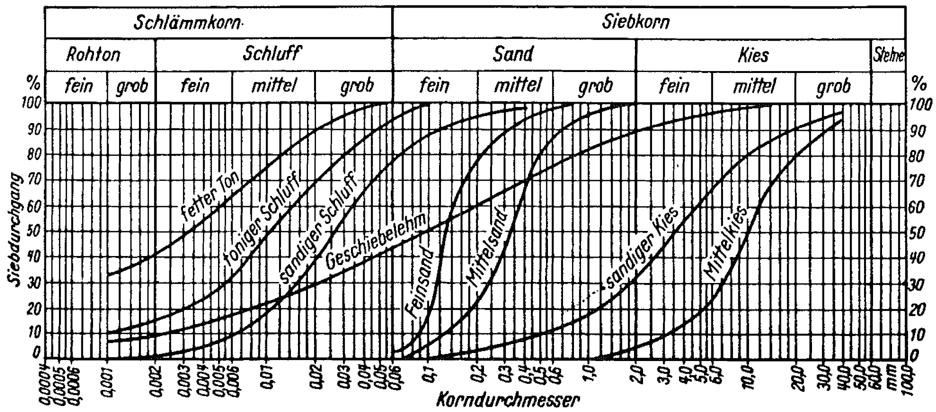


Fig. 17 <u>Typical grain distribution of different beds.</u>

Ordinate: Effective filtering,
Abscissa: Grain diameter.

SILT					SIFT - SIZE - GRAIN					
Coarse clay		Sludge			Sand			Gravel		Stone
Fine	Coarse	Fine	Medium	Coarse	Fine	Med.	Coarse	Fine	Med.	Coarse

Legends to curves from left to right

Fat clay; sludge containing clay; sandy sludge; pebble clay; fine sand; medium sand; sandy gravel; medium gravel.

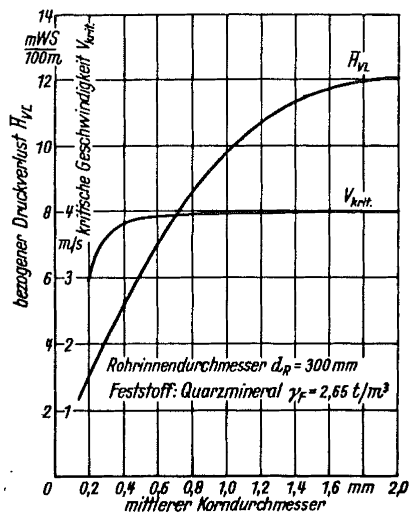


Fig. 18 Dependence of the critical velocity and the resulting pressure losses on the mean grain diameter of the transported solids.

Ordinate:	Resultant pressure loss	
	Critical velocity	
Abscissa:	Mean grain diameter.	
Pipe I.D.	300 m	Specific weight
Solids:	Quartz-mineral	2650 kg/ms

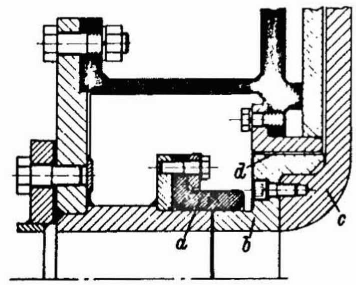


Fig. 30 Suction sealing gland, in radial contact supplied with sealing fluid (water).

(a) Rubber - lip seal,
(b) Sealing bush,
(c) Impeller,
(d) Throttle gland.

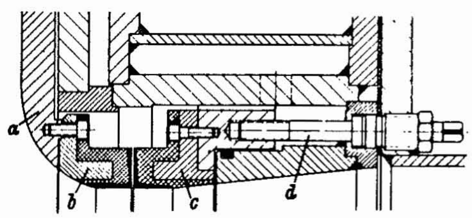


Fig. 31 Suction sealing gland, axially free from contact with sealing fluid (water).

(a) Impeller,
(b) Rubber lined sealing bush,
(c) Axially adjustable rubber seal,
(d) Adjusting screw.

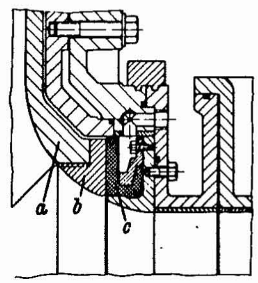


Fig. 32 Suction sealing gland, axially in contact supplied with sealing fluid (water).

(a) Impeller,
(b) Sealing Bush,
(c) Rubber seal.

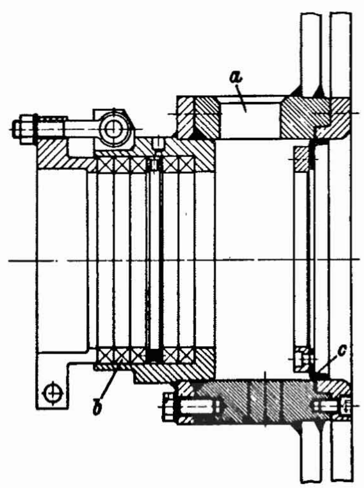
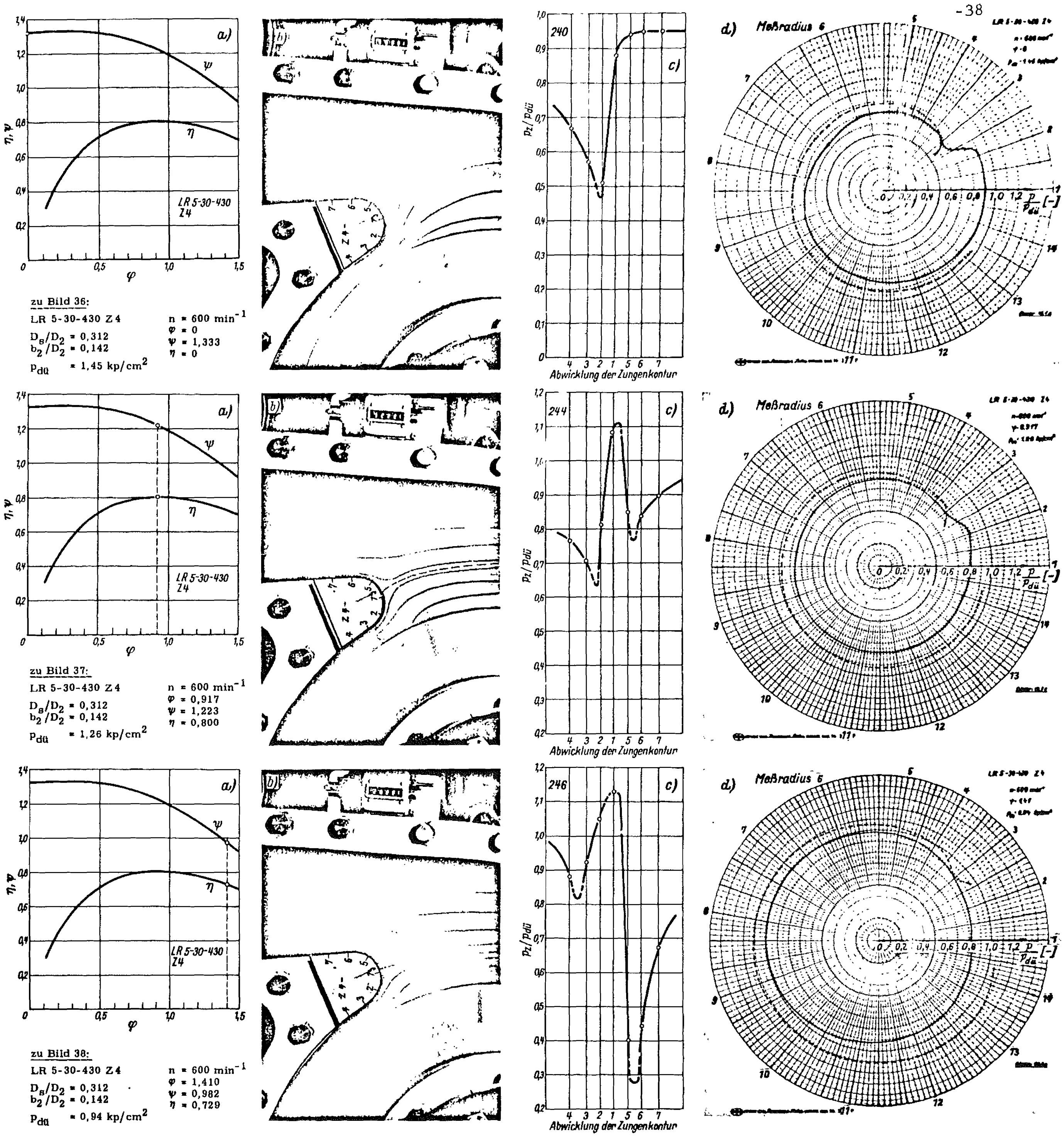


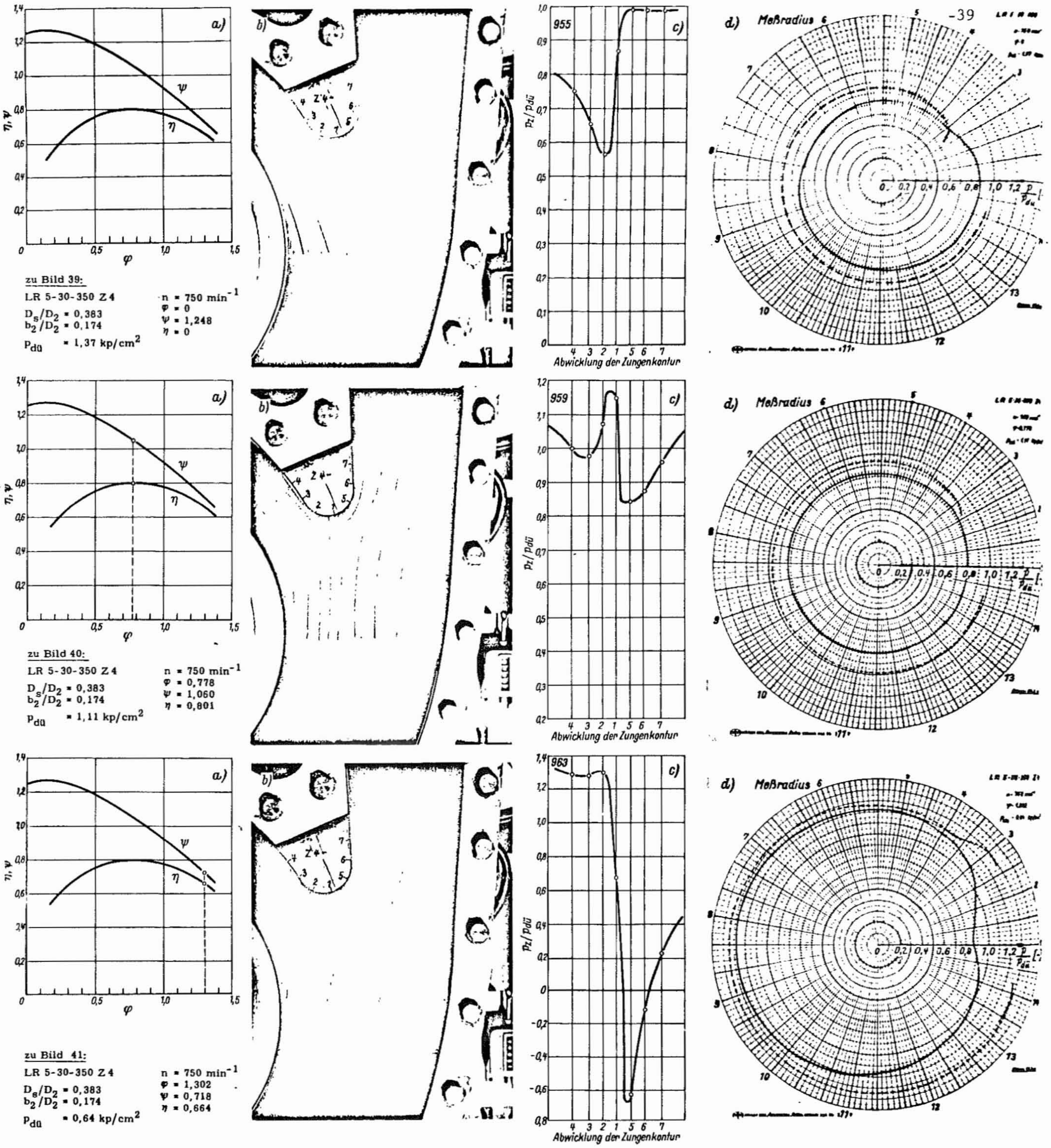
Fig. 33 Journal seal as a combined lip-seal and packing gland with sealing fluid supplied.

(a) Sealing water fitting,
(b) Packing,
(c) Rubber lip - seal rotating with shaft.



Figs. 36/38 Pressure and flow conditions at the tongue and in the volute of the model pump L R 5 - 20 - 430 Z 4 for 3 different operating conditions.

(a) Position of the measuring point,
(b) Flow in the neighbourhood of the tongue,
(c) Pressure distribution on the tongue surface,
(d) Pressure distribution at the impeller exit and around the volute peripheri.



Figs. 39/41 Pressure and flow conditions at the tongue and in the volute of the model pump L R 5 - 30 - 350 Z 4 for 3 different operating conditions.

(a) Position of the measuring point,
(b) Flow in the neighbourhood of the tongue,
(c) Pressure distribution on the tongue surface,
(d) Pressure distribution at the impeller exit and around the volute peripheri.

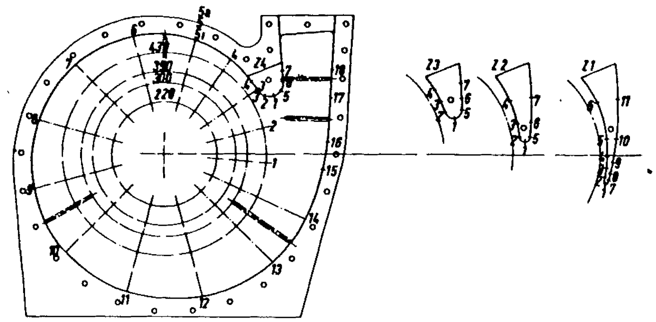


Fig. 42 Distribution and position of the pressure tappings in the volute and around the tongue.

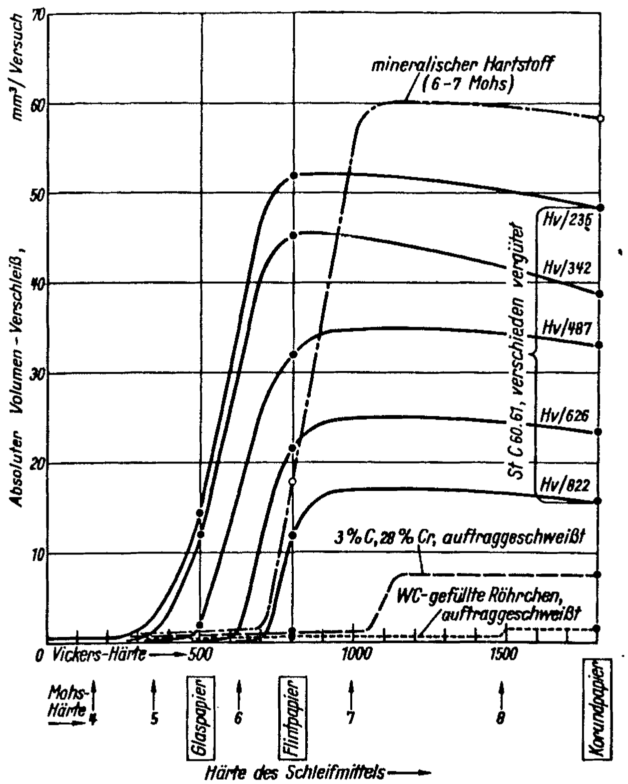


Fig. 43 Relationship between the wear volume V and the hardness of the grinding material by dry grinding using the grinding - plate method of (16).

Ordinate: Absolute wear volume
Abscissa: Wickers - Hardness (hardness of grinding material).

Mohs Hardness, Glasspaper, flint paper, emery paper, identification of curves working upwards.

W C - filled small pipes deposit welded
3% C 28% Cr deposit weld
Several qualtities of St C 6061.
Mineral hard - material.

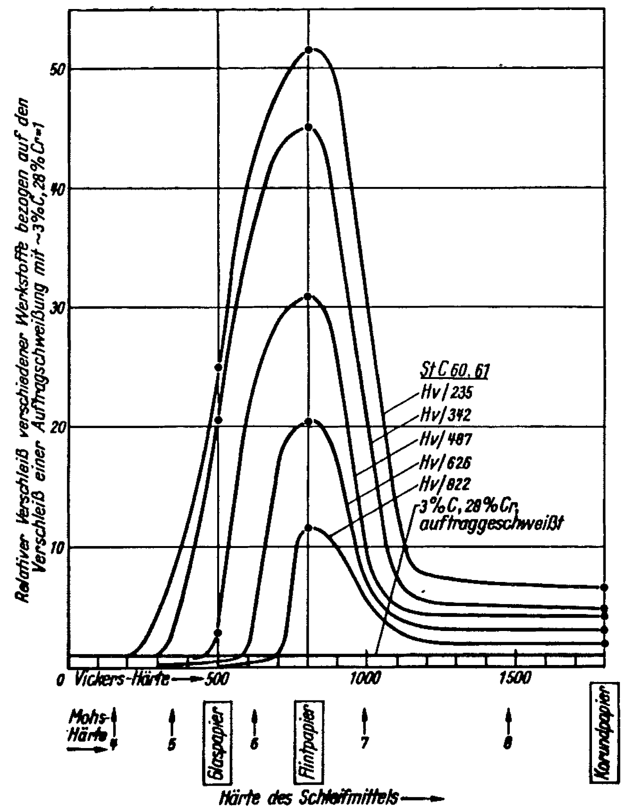


Fig. 44 Relative wear of the materials in Fig. 43 related to grinding material hardness H_s.

Ordinate: The wear of several materials relative to the wear of deposit welded 3% C 28% Cr.
Abscissa: Hardness of grinding material.

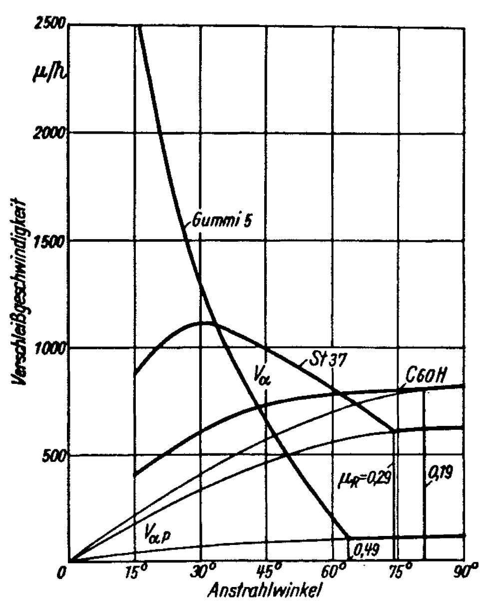


Fig. 45 Relationship between the wear rate and the impact angle using quartz sand (d m > 0.2 < 1.5 mm) in sand blasting device. (p - 1 atm) (o.f.(13)).

Ordinate: Wear rate microns/hour
Abscissa: Impact angle α

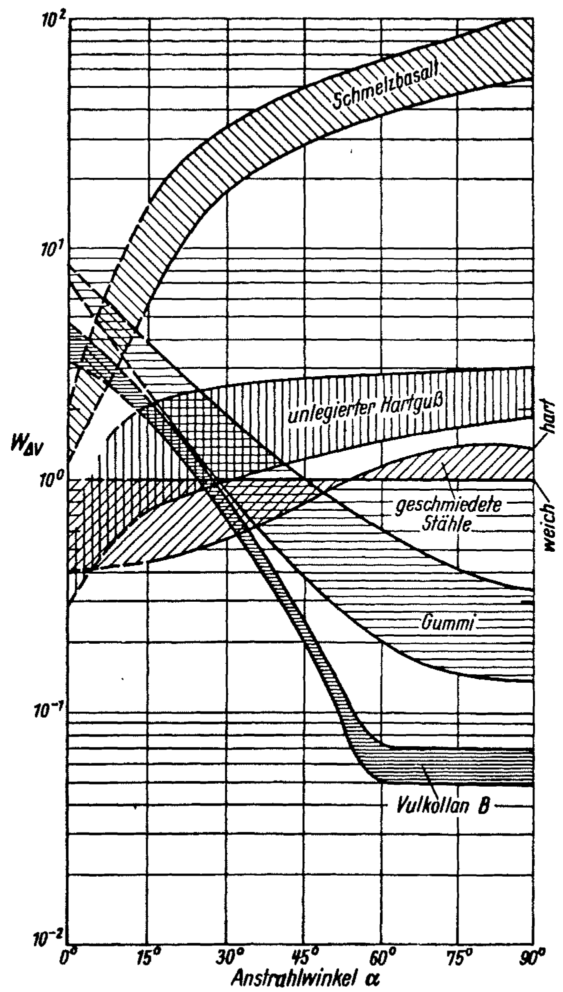


Fig. 46 Wellinger - Curve for various materials (o.f.(13)).

Abscissa: Impact angle α

Identification of curves working upwards, Wulkellan B; Rubber; Wrought Iron - hard, soft; Unalloyed cast iron.

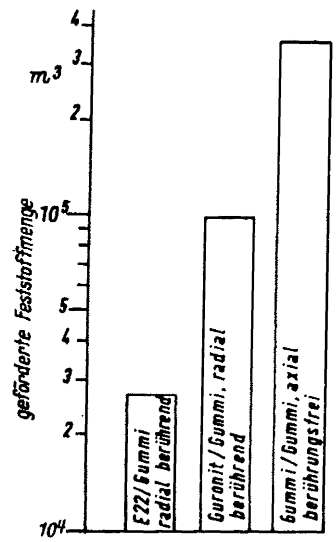


Fig. 47 <u>Improvement of the wear resistance of a suction gland of a dredging pump transporting sand - water mixture, through design and material improvements</u>.

Ordinate: Solids transported cu m.
Colums from left.
E 22/rubber, radially in contact.
GURONITE/RUBBER, radially in contact.
Rubber/Rubber axially free of contact.