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2 Trailing suction hopper dredger



Figure 2-1 Trailing Suction Hopper Dredger (TSHD)

2.1 General description

2.1.1 Characteristics

The characteristics of the trailing suction hopper dredger are that it is a self-propelled sea or inland waterway vessel, equipped with a hold (hopper) and a dredge installation to load and unload itself.

In a standard design the trailing suction hopper dredger is equipped with:

- One or more suction pipes with suction mouths, called dragheads that are dragged over the seabed while dredging.
- One or more dredge pumps to suck up the loosened soil by the dragheads.
- A hold (hopper) in which the material sucked up is dumped.
- An overflow system to discharge the redundant water.
- Closable doors or valves in the hold to unload the cargo.
- Suction pipe gantries to hoist the suction pipes on board.
- An installation, called the swell compensator, to compensate for the vertical movement of the ship in relation with the sea-bed.

2.1.2 Application area

The trailing suction hopper dredger has a very wide application area and is therefore called the workhorse of the dredging industry.

Because it needs no anchorage system to position the vessel when dredging, which can be an obstacle for passing ships, in the early days the trailing suction hopper dredger (TSHD) was mainly used for the deepening and maintaining of waterways. Nowadays the trailing suction hopper dredger is also used for land reclamation. Examples of that type of jobs are the large reclamation works executed in the Far East. Here the non-bearing soil was first removed by the trailing suction hopper dredger, after which the same area was filled again with sand. The reason for a preference of the trailing suction hopper dredger above other types of equipment for this type of work is mainly the fact that the distances to the dump areas for the non-suitable material and distance from the sand pits are too large for a direct discharge and supply with pipelines.

The main advantages of a trailing suction hopper dredger are:

- The ship does not dredge on a fixed position. It has no anchors and cables, but it moves freely, which is especially important in harbor areas.
- The trailing suction hopper dredger is quite able to work under offshore conditions.

The materials that can be sucked are mainly silt and sand. Clay is also well possible, but can give some trouble with congestions in the draghead and **rutting**. Rutting is the slipping back of the **dragheads** in their old rut or trail. Dredging rock with a trailing suction hopper dredger is in most cases not economical. It requires very heavy dragheads, also called **ripper-heads**, and the productions are usually very low.

2.1.3 History

The first TSHD “General Moultry” with a hopper size of 155 cu yard (118.5 m³) was built in 1855 in the United States. Few years later 1959 a trailing suction hopper dredger was build in France for maintenance work in the harbor of St. Nazaire.

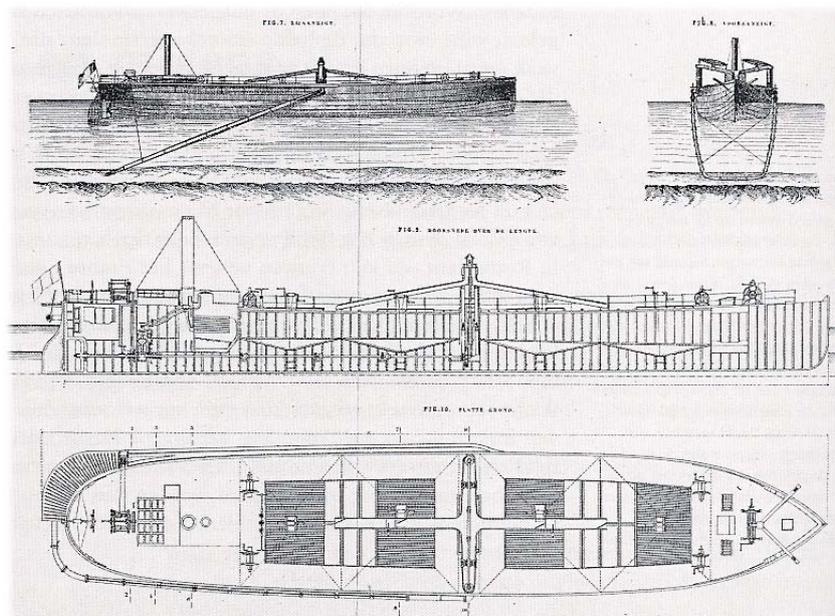


Figure 2-2 French trailing suction hopper dredger from 1859

The ship had two drag suction pipes, which were connected at the bottom by a tube with holes (Figure 2.2). The dredging material, silt, was sucked through the holes in the connection tube by a steam-driven centrifugal pump. The size of the hopper was 240 m³.

In 1962 a dredger was built according to this layout at the yard Fijenoord at Rotterdam, Netherlands. Those types were able to dredge only very light silty material.

The real development of the trailing suction hopper dredger emanated from the **stationary suction hopper dredger**, one of the few Dutch dredge inventions. This self-propelled ship has a hopper and a forward pointing suction pipe. The dredge method is like a stationary suction dredger, working stationary on anchors and cables. At first with a pipe in the well, but the suction pipe was mounted on the side during the excavation of the Nieuwe Waterweg as it appeared not the right solution in waves.

The change from an anchored to a self-propelled dredging ship was a big step ahead. At first the suction pipe on board of a trailing suction hopper dredger was placed in a well behind the ship, but was soon moved to the side. The trailing suction hopper dredger has mainly developed in the USA and reintroduced in the Netherlands in the fifties and improved till its state of today.

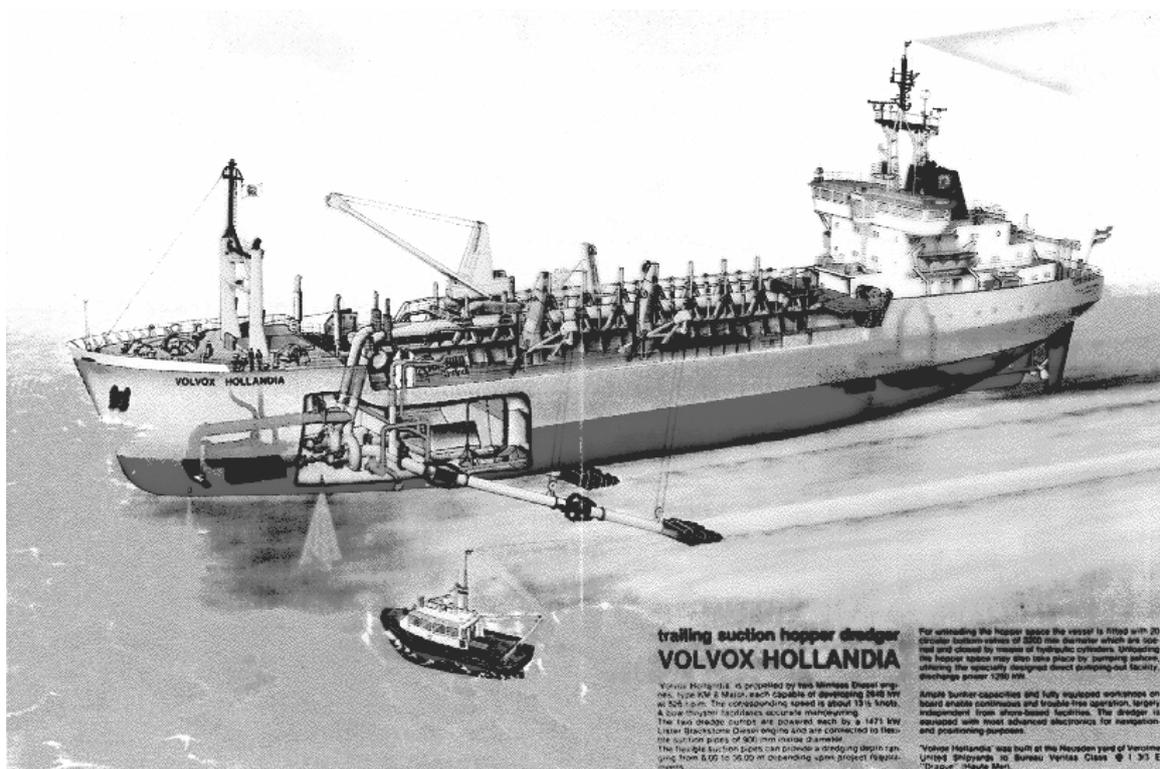


Figure 2-3 Artist impression of TSHD

2.1.4 Work method

When arriving on the dredging area the speed of the trailing suction hopper dredger is reduced to approximately 3 knots (± 1.5 m/s) and the suction pipes are swung outboard. The suction pipes are initially lowered approximately horizontally until the trunnion slide is positioned in front of the suction intake (Figure 2.4).

Next the **intermediate gantry** and the **draghead winch gantry** are lowered such that the pipe rotates like a straight line around the trunnion.

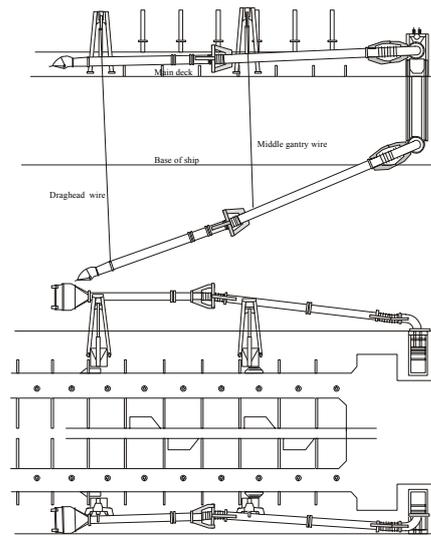


Figure 2-4 Suction pipe lowered

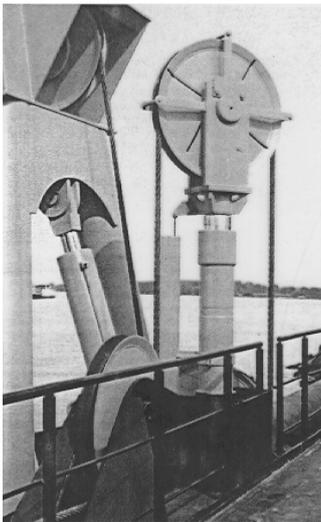


Figure 2-5 The swell compensator

When the suction mouth arrives a few meters above the sea bottom the **sand pumps** are started, the dragheads are lowered onto the seabed (which can be seen by the rise of the swell compensators cylinders (Figure 2.5) and the dredging can start.

Where and how much needs to be dredged is nowadays shown on electronic maps (computer screens). It also shows the position, direction and course of the ship.

The trailing suction hopper dredger sucks the soil from the seabed at a sailing speed of 1 to 1.5 m/s (2 to 3 knots) and deposits it in the **hopper**. For non- or bad-settling soils the dredging is stopped when the surface of the mixture in the hopper reaches the upper edge of the **overflow** (Figure 2.6).

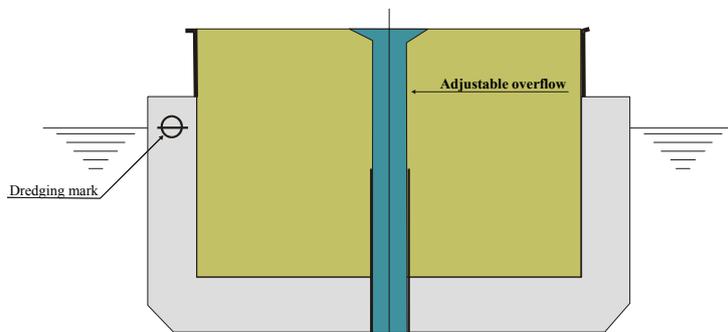
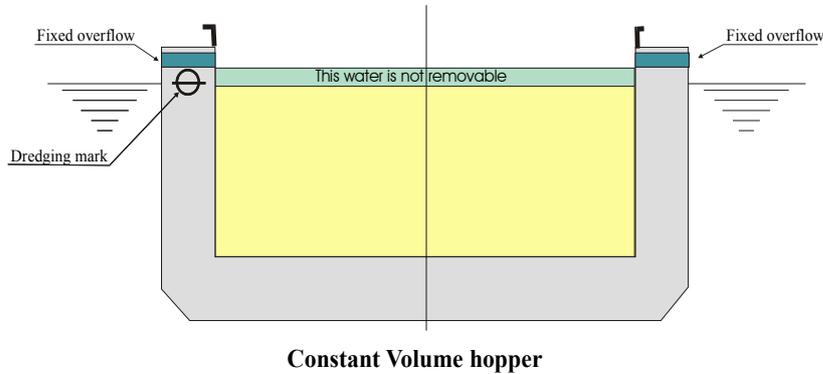


Figure 2-6 Justable overflow

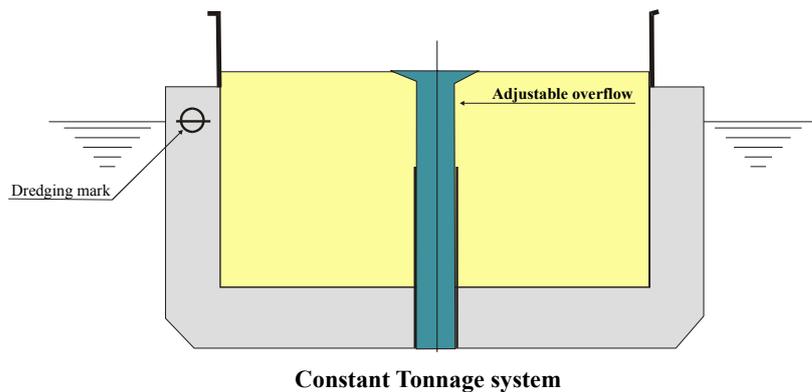
The hopper filling is at maximum or the fill rate is 100%. Usually pumping continues for five minutes more to remove floating water on the mixture through the overflow. When dredging settling soils the dredging continues when the maximum level of the overflow is reached. Most of the solids will settle and the remainder is discharged with the water through the overflow.



If the trailing suction hopper dredger is equipped with a fixed overflow (not adjustable) than the ship is loaded until it reaches its **dredge mark** (a fixed allowed draught) after which the suction is stopped.

Figure 2-7

That case it is said that the ship is designed as a **Constant Volume System (CVS)**.



If the ship however has a height adjustable overflow system, than it is possible, when the hopper is full and the ship is on its mark, to lower the overflow level such that the total weight of the in the hopper present water and soil remains constant.

Figure 2-8

This is called a **Constant Tonnage System (CTS)**.

The dredging is stopped when:

- The hopper is full. Overflow not allowed.
- The maximum allowable draught is reached and the overflow can not be lowered usefully anymore.
- The economical filling rate is reached.

When dredging stops, the suction pipes are pumped clean to prevent settling of the sand or gravel during the hoisting of the pipes causing an extra load for the winches. When the pipes

are cleaned the pumping stops and the pipes are raised. When the dragheads are out of the water the ship's velocity is increased to sail to the discharge area.

The discharge area can:

- Be in its most simple shape a natural deepening of the seabed, **the dumping area** (shortly **dump**), to store redundant material. If the storage capacity is large, there is no concern about the way of dumping. This hardly happens nowadays. The client demands usually a dump plan to fill the dump as efficiently as possible. At all times the draught on the dump needs to be sufficient to open the bottom doors or valves (Figure 2.9).
- Be a storage location for contaminated silt, like for instance the Slufter (Rotterdam harbor). Here the material is pumped ashore using a pump ashore discharge system.
- An area that has to be reclaimed.
- An oil or gas pipe that has to be covered.

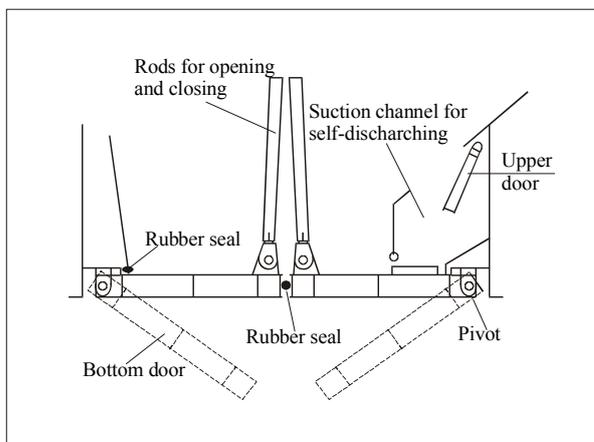


Figure 2-9 Bottoms doors operated by rods

In case of the discharge area is a **dump**, opening the doors or valves in the base of the hopper does the unloading.

This is usually done with an almost non-moving ship, certainly when accurate dumping is required. During the dumping water is pumped onto the load by means of the sand pumps. The eroding water stimulates the dumping process. If the trailing suction hopper dredger is equipped with **jet pumps** connected to a jet nozzle system in the hopper, those will be used too. The jets more or less fluidize the load and improve the dumping process.

If the load is pumped ashore using the sand pumps than only these jets are available to fluidize or erode the load.



Figure 2-10 Pump ashore connection

The **shore connection**, being the connection between the board pipeline and the shore pipeline is currently mostly positioned just above the bow (Figure 2.10). The connection between the ship and the shore piping is in this case a rubber pipeline. The ship remains in position by maneuvering with its main propellers and bow thruster(s).

When the load is either dumped or pumped ashore the ship will return to its suction area and a new cycle starts. In general the ship sails empty, in a non-ballast way, back to its suction section. There is only some residual water and/or load left in the hopper



Figure 2-11 TSHD J.J.F. de NUL picking up the floating pipeline to the shore connection

2.2 The design

2.2.1 The productive capacity

When a dredging company wants to order a new trailing suction hopper dredger usually a market study is performed that about the required production capacity of the new dredger.

The required production capacity is expressed in m^3/week or m^3/month or even cubic meters per year. Besides that insight required about the expected average cycle time of the trailing suction hopper dredger on the different jobs, as well as the type of soils to be dredged. Then the production capacity can be translated to:

- The required payload in ton mass.
- The maximum hopper volume in m^3 .

If the ship is used for a single purpose, for instance the maintenance of a harbor area, than the required production capacity is usually known and therefore the above mentioned ship data.

For an international operating dredging contractor this is different and far more complicated. Answers have to be given to the question how the average cycle and the required production capacity will evolve in the future. For these contractors there is in fact only one requirement and that is dredging cheaper than their competitors. This leads quickly to a demand for large dredgers, which dredge cheaper and therefore more competitive.

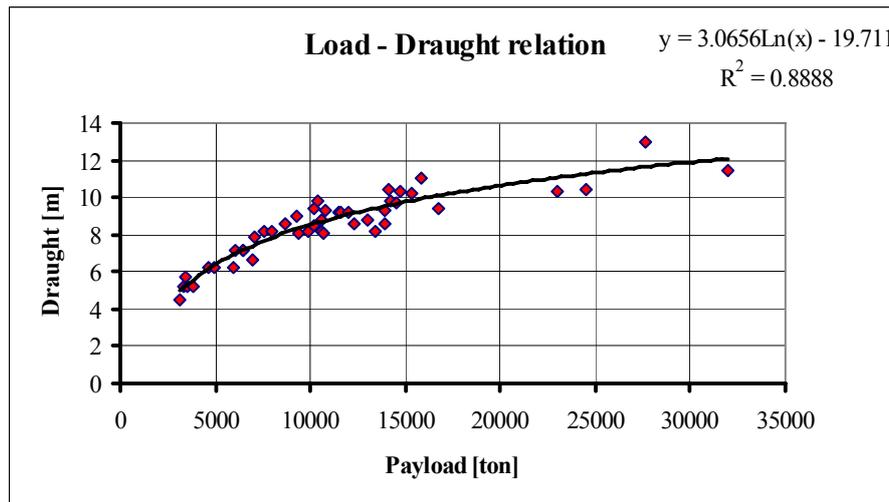


Figure 2-12 Payload - draught relation

The only decelerator on the building of larger vessels is the draught of the ship. When the draught increases, the usability of the ship decreases. The contractor can, dependent on the expected amount of work as function of the (initial) dredging depth, determine the availability of the ship for a certain draught.

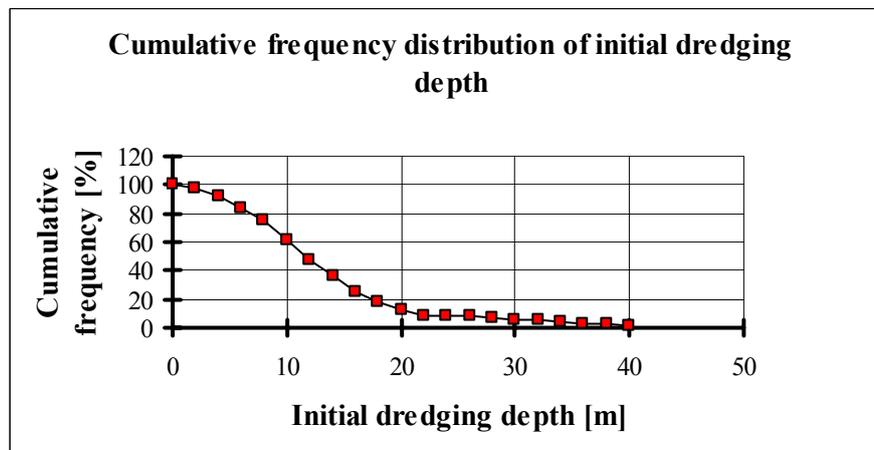


Figure 2-13

Unfortunately it is possible that market expectations of today are totally out-of-date in 5 years. The management chooses for a certain production capacity and later one will just if this choice was right.

The design is usually made a co-operation between the builder and the client is often scaled-up from successful ships. Of course the proper scale rules have to be obeyed when scaling-up. At this moment five classes of trailing suction hopper dredgers can be distinguished:

Small hoppers	deadweight capacity to	± 50 MN (to 5000 ton mass)
Medium size hoppers	deadweight capacity	50-100 MN (5000-10000 ton mass)
Large hoppers	deadweight capacity	100-150 MN (10000-15000 ton mass)
Jumbo hoppers	deadweight capacity	150 250 MN (15000-25000 ton mass)
Mega hoppers	deadweight capacity	>250 MN (above 25000 ton mass)



Figure 2-14 Different scales Fairway (23.347 m³) and the Sospan (700 m³)

2.2.2 The main dimensions

When the choice for the production capacity of the trailing suction hopper dredger to be built is made, the hopper volume is known too. The main dimensions of the trailing suction hopper dredger are determined, as by other ships, by the required payload, draught and speed. It will be clear that a straight correlation exists between these quantities to satisfy the shipbuilding demands. After all a large hopper volume with a limited draught gives wide long ships with possible disadvantages like a poor behavior in swell or problems to obtain the required speed.

Trailing suction hopper dredgers are therefore build according to certain **ship ratio**, such as L/B, B/H and B/T ratio's (L=length, B=width, H=depth and T=draught). Those ratios's depend on market requirements too and therefore change in time (Figure 2.15)

With the remark that a large B/T ratio:

- Results in a large initial stability, resulting in heavy ship motions in swell.
- Has an adverse effect on the resistance of the ship.

With a large L/B ratio a lean ship is obtained with the advantages of:

- A simple construction as a result of the long equal mid-section (cheap).
- A relative low resistance, therefore a higher velocity with the same installed propulsion power.

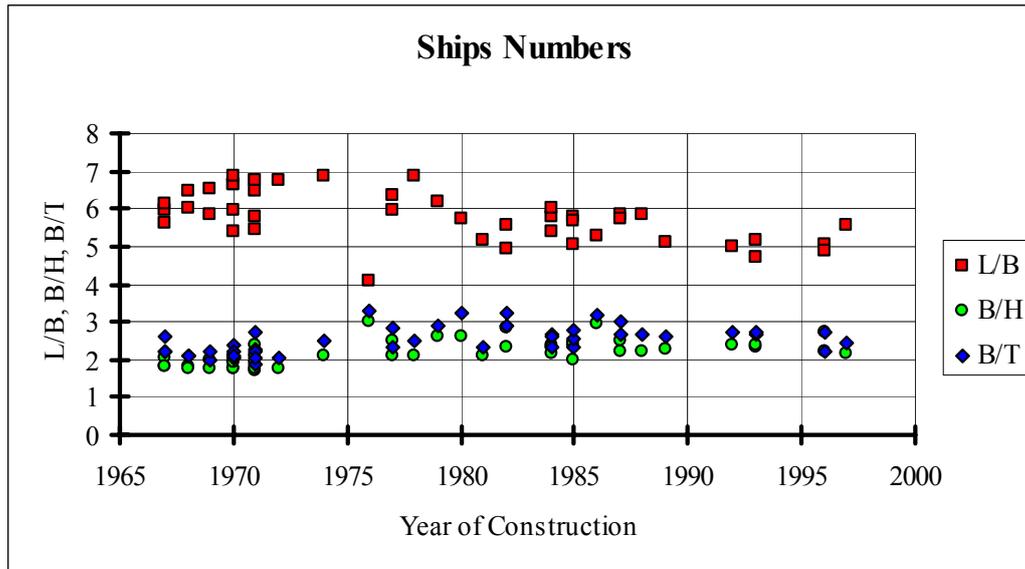


Figure 2-15

On the other hand a small L/B gives a good stability and longitude strength and demands therefore less material, which is also cheaper.

In general a smaller B/H and a larger L/B result in less building costs. So demands for the draught (smaller T) will cost extra money and will have to be earned with a higher usability.

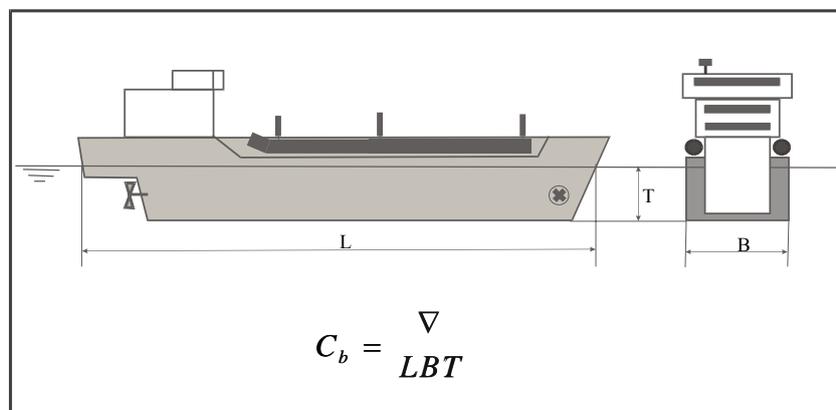


Figure 2-16 Definition Block coefficient

Definition Block coefficient

Of course the required block coefficient $C_b = \frac{\nabla}{L \cdot B \cdot T} = \frac{\text{displacement}}{L \cdot B \cdot T}$ is involved too.

<i>Displacement</i>	=	In m ³
B	=	Width of ship at the main section I m
L	=	Length between perpendiculars in m
T	=	Draught at International mark in m

The lower C_b , the longer the ship will be with the same displacement. For trailing suction hopper dredger C_b lies between 0,78 and 0,85.

Also the required maximum dredging depth can have an influence on the length of the ship. Naturally, the long suction pipe has to be stored on the deck and that requires length.

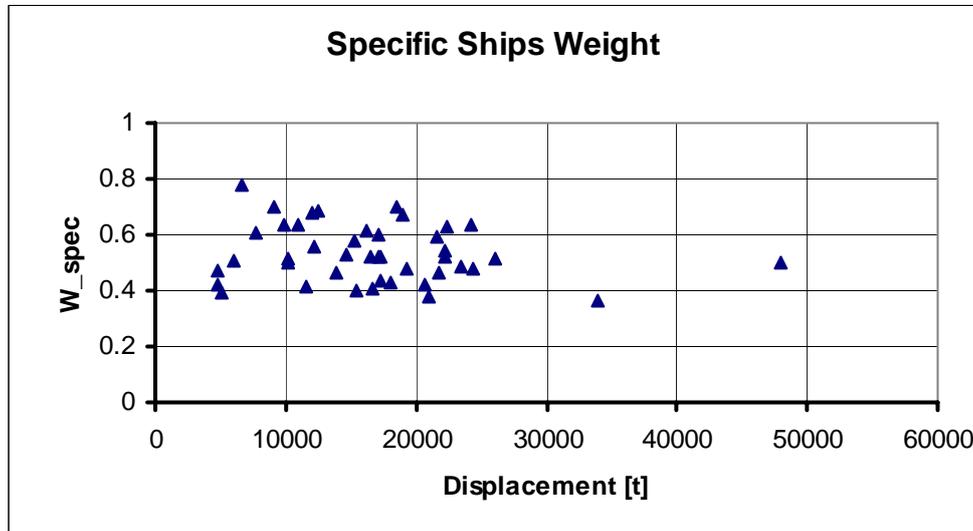


Figure 2-17

A good measure to see if the trailing suction hopper dredger is well placed in the market is to compare its specific weight with that of its competitors. The specific weight can be defined as the ratio between the ships weight and payload. The weight is directly related to the costs and the payload to the profits. In Figure 2.17 the specific weight for a large number of ships is given.

2.2.2.1 The load

As aid, the payload in tons and the maximum hopper volume in m³ determine the amount of soil that a trailing suction hopper dredger is able to carry each voyage. These are of great importance. The payload is the weight of the paying load that the ship may carry on the maximum allowed draught. The payload is often a cause for misunderstandings. As a definition the payload is the ship weight of the loaded ship subtracted with the weight of the empty ship ready for service. This is shown in the hereunder shown chart.

	Dutch term	English term	Explanation
1	Scheepsgewicht	Ships weight	Construction weight and necessary equipment like: anchors, chains, moor cables, rescue equipment, nautical equipment and inventory of the cabins, galley, engine-room and tool-room of the boatswain
2	Toegevoegde gewichten	Added weights	This is the liquid filling of all systems on board including the water in the inlets. Also the outside water situated above the bottom deck for instance under and around the bottom doors is included.
1+2	Gewicht leeg schip	Weight "light" ship	

3	Toelading	Dead weight	Weights of: Crew and their possessions, consumer goods, spare parts, and ballast water and load .
1+2+3	Gewicht van het "geladen" schip	Weight of "loaded" Vessel	
4	Gewicht lading	Weight cargo	Weight of the paying load.
1+2+3 +4	Gewicht bedrijfsklaar schip	Ships weight ready for Service	

Figures below gives some information about "light weight" and "dead weight" of TSHD's

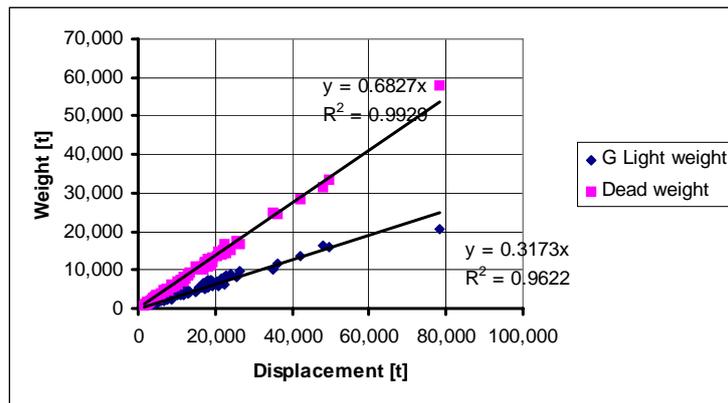


Figure 2-18

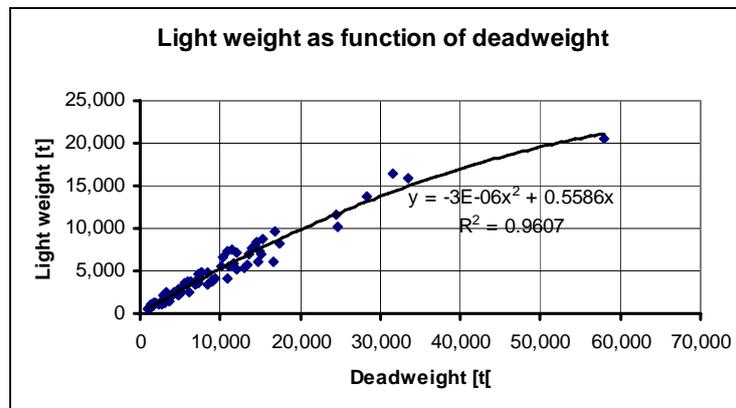


Figure 2-19

Except that there are different names for the payload, it is also apparent that it varies in time and often decreases. The reason is that when the ship has been in use for a while things will be added or reinforced, which causes an increase in the ships weight. Spare parts also tend to remain on board that should be stored onshore. In fact there is only one way to determine the payload correctly:

1. Clear the hopper such that no remaining soil is present.
2. Determine the displacement of the ship with the draught and the trim of the ship, the displacement is the weight of the ship including the water in the hopper.
3. Determine the weight of the water present in the hopper by determining its volume and the specific gravity
4. Subtract the weight of this water the ships weight determined under point 2. This is the weight of the ship ready for service.

5. The payload is obtained by subtracting the ships mass (displacement x water density) in tons on the maximum allowed draught with the weight of the ship ready for service.

It will be clear that the payload is never constant, but varies with the weight of the consumer goods like fuel, lubricants, drinking water etc.

In case of light soils, such as silt and soft clay, the maximum hopper volume can be decisive for production instead of the payload.

2.2.2.2 The hopper density.

As mentioned earlier, the production capacity of a trailing suction hopper dredger is indicated with the quantities:

- Pay-load
- Maximum hopper volume

The quotient $\frac{\text{pay-load}}{\text{maximum hopper volume}} [\text{kg/m}^3]$ is called the hopper density and is a

measure for the average density that a dredging contractor expects to dredge during the economical lifetime of the ship. It also says something over the purpose for which the dredger is designed. Is this for instance maintenance of a fairway in a sandy soil, than the dredges sand in the hopper will have a density of approximately 1900 kg/m^3 . Unfortunately no hopper can be filled to a 100% but approximately to maximum 90%. The maximum hopper density required is $1900 * 0.9 = 1710 \text{ kg/m}^3$

For a gravel trailing suction hopper dredger this is for instance: $2000 * 0.9 = 1800 \text{ kg/m}^3$. And for a silt trailing suction hopper dredger this could be even 1300 kg/m^3 . In Figure 2.20 the hopper density of international operating dredging contractors is shown as function of time. It stabilizes at the end of the eighties and early nineties around 1500 kg/m^3 , but due to the big reclamation works it is increasing again.

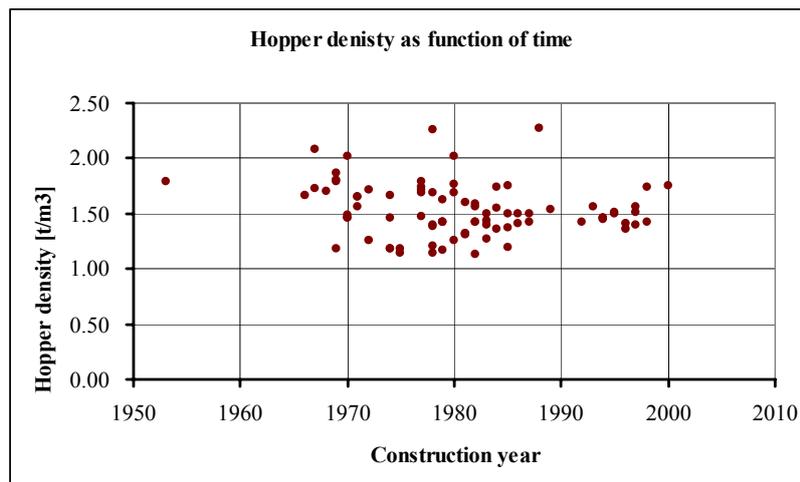


Figure 2-20

2.2.3 The dredge installation

The design of a dredge installation includes the determination of the required main dimensions and required powers of the following dredging components:

- Number of suction pipes
- Pump capacity [m^3/s]
- Suction and discharge pipe diameter [m]
- Type dredge pump
- Sand pump drive and power [W]
- Type and size of the draghead(s)
- Hopper shape
- Jet pump power and drive [W]
- Discharge systems

For the subjects the production should be corrected in a certain way from the average cycle production of the dredger.

For instant, assume that the dredger is designed for a payload of 16000 ton and a hopper volume of 10000 m^3 and a average loading time in sand with a d_{50} of 200μ of 90 minutes. De density of the soil in the hopper is $1900 \text{ kg}/\text{m}^3$. When the hopper is loaded the volume of sand will be 8421 m^3 . The average load rate is in this case $8421/90=93 \text{ m}^3/\text{min}=1.56 \text{ m}^3/\text{s}$.

When cumulative overflow losses of 20% are to be expected, then the dragheads should excavate $1.56/0.8=1.95 \text{ m}^3/\text{s}$ as an average. Every m^3 of sand contains $1-(1900-1025)/(2650-1025)=1-0.538=0.462 \text{ m}^3$ water in the pores. ($\rho_{\text{water}}=1025 \text{ kg}/\text{m}^3$, ρ_{sand} is $2650 \text{ kg}/\text{m}^3$). So a production of $1.95 \text{ m}^3/\text{s}$ equals a sand mass of $1.95*0.538*2650=2780 \text{ kg}/\text{s}$

2.2.3.1 Number of suction pipes

A trailing suction hopper dredger is usually equipped with two suction pipes. For smaller and medium size trailing suction hopper dredgers it is cheaper to use only one suction pipe. With two suction pipes the total efficiency is often better because it is still possible to dredge when one of the pipes fails.

There are also examples of large trailing suction hopper dredger with one suction pipe: the ANTIGOON of Dredging International with a hopper volume of 8.400 m^3 and the VOLVOX TERRA NOVA of Van Oord ACZ with 18.000 m^3 hopper volume. In principal it is an economical consideration, but looking from the process technical side there are some questions. For example: is one draghead as efficient as two dragheads with the same width?

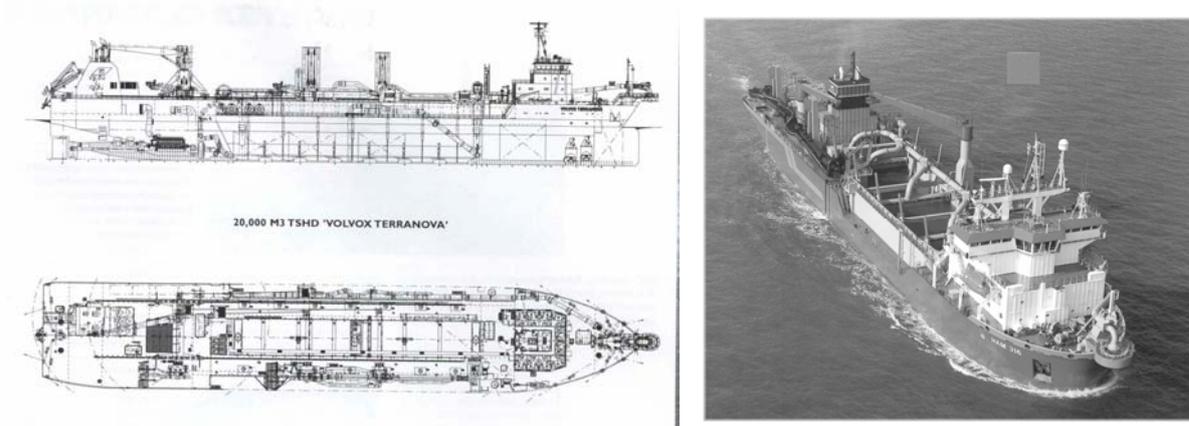


Figure 2-21 Volvox Terra Nova and HAM 316, both with one suction pipe

2.2.3.2 Pump capacity

The sand pump capacity can be determined using several criteria:

1. In a particular type of soil a certain load time is demanded. (for instance 1 hour for sand with a d_{50} of 200-300 μm)

The volume pure sand as function of time is:
$$V_{sand} = \left[\int_0^T C_i Q_i - \int_0^T C_o Q_o \right] dt$$

C_o	=	Volumetric concentration at overflow	[-]
C_i	=	Volumetric concentration at intake	[-]
Q_o	=	Discharge at overflow	[m^3/s]
C_i	=	Flow rate at intake	[m^3/s]
T	=	Loading time	[s]
n_0	=	Porosity	[-]
ov	=	Cumulative overflow losses	[-]
V_{sh}	=	Volume sand in the hopper	[m^3]
ρ_s	=	Density of sand in the hopper	[kg/m^3]

This sand occupied in volume in the hopper of $V_{sh} = \frac{V_{sand}}{1 - n_0}$; n_0 is porosity

For TSHD's having a constant volume system $Q_m = Q_i = Q_o$, with Q_m is the pump capacity; so the mass of the load becomes:

$$L_{mass} = \frac{\rho_s}{1 - n_0} Q_m \int_0^T (C_i - C_o) dt = \frac{\rho_s}{1 - n_0} Q_m \cdot C_i (1 - \text{ov}) \cdot T$$

With ov being the cumulative overflow losses defines as $ov = \frac{\int_0^T C_o Q_o}{\int_0^T C_i Q_i}$

For 1 hour loading the flow rate is: $Q_m = \frac{V_{sand}}{(1-ov)C_i \cdot 3600}$

C_i and C_o are delivered concentrations; so $C_i = C_{dv}$

The expected C_{vd} depends on the particle size, the permeability of the soil and the available jet water momentum. (see 2.5.5.1.3)

If the TSHD is designed as a constant tonnage dredger the incoming mass equals the outgoing mass; so $m = m_i = m_o$.

$$m_i = Q_i \rho_{mi} \text{ and } m_o = Q_o \rho_{mo} \text{ so } Q_i \rho_{mi} = Q_o \rho_{mo} \text{ or } Q_o = Q_i \frac{\rho_{mi}}{\rho_{mo}}$$

ρ_{mi} and ρ_{mo} are respectively the mixture densities at the intake and overflow.

The load becomes now:

$$L_{mass} = \frac{\rho_s}{1 - n_0} Q_i \int_0^T \left(C_i - C_o \frac{\rho_{mi}}{\rho_{mo}} \right) dt = \frac{\rho_s}{1 - n_0} Q_m \cdot C_i (1 - ov) \cdot T$$

Although the formula is the same as for the constant volume system hopper dredgers it doesn't mean that the cumulative overflow losses are the same for both types.

2. In an ascertain type of sand the load rate in m^3/s or in t/s must have a minimum value.

If there would be no overflow losses than the load rate is directly proportional to the flow rate. However, the overflow losses increase with an increasing flow rate, which result in an increasing deviation from the linear relation. (Figure 2.22& 2.23)

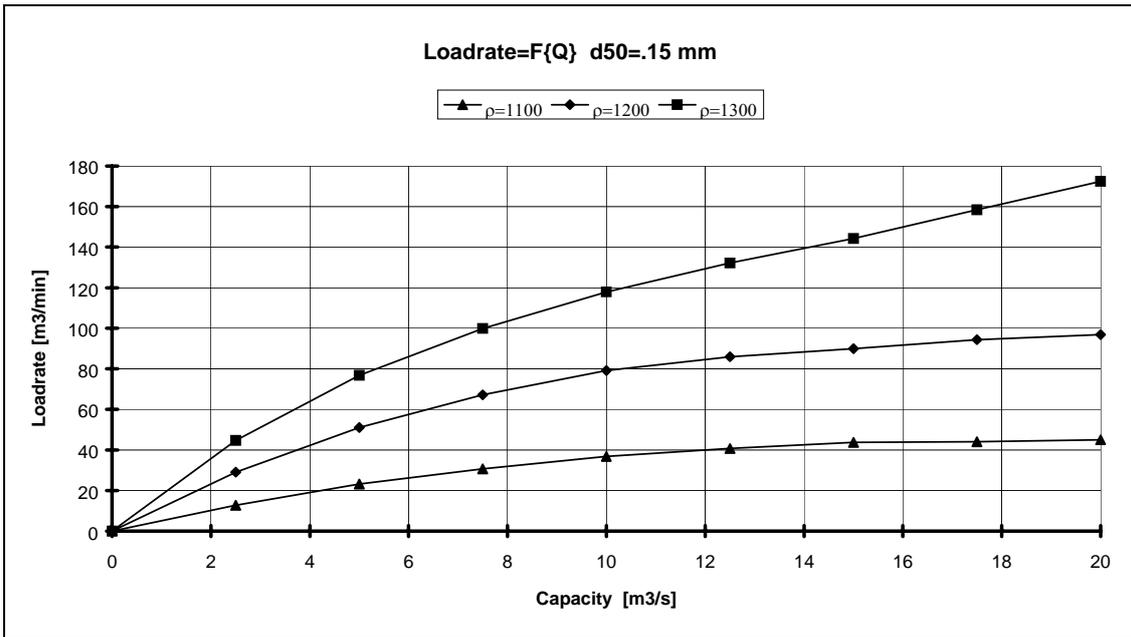


Figure 2-22 Loadrate as function of pump capacity

It can be proven that for certain particle sizes there is an optimum loadrate.

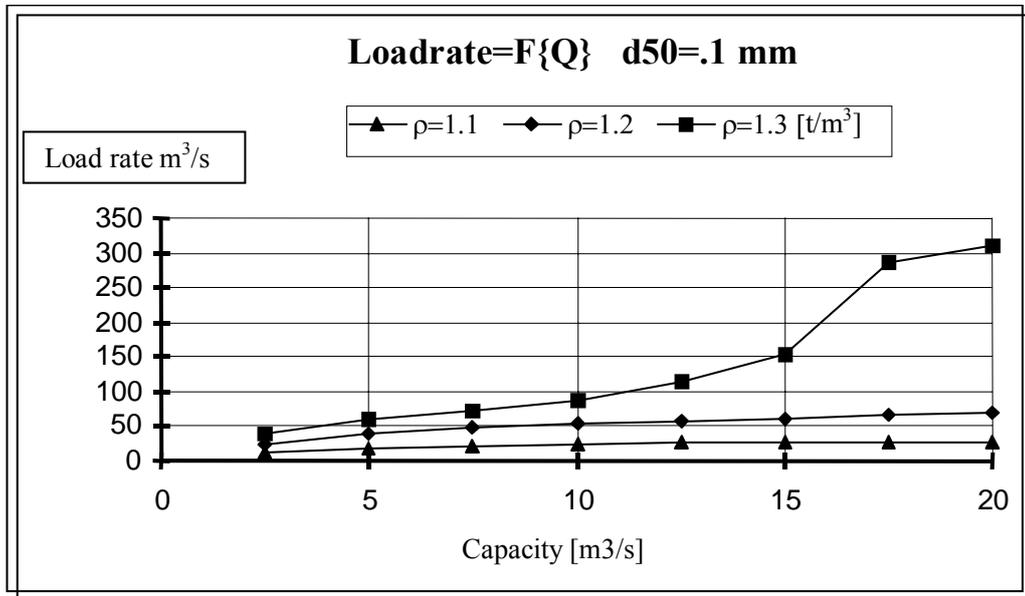


Figure 2-23

The increase of a higher suction production (load rate) must be considered against the higher sand and water pump power, larger suction pipe diameter and dragheads etc.

Remark: In Figure 2.23 the step in the load rate is caused by the fact that for high densities and high flow rates the loading after the overflow is not necessary since the optimal production for the dredge cycle has been reached.

3. When apart from the soil the cycle time is known too, than the flow rate can be chosen such that the cycle production is maximal. The cycle production is defined as

$$\text{the quotient between loading and cycle time, so: } P_c = \frac{\text{load}}{t_{\text{suction}} + t_{\text{non-suction}}}$$

If there are no overflow losses than this formula can be written as:

$$P_c = \frac{\text{load}}{\left(\frac{\text{load}}{Q \cdot C_{vd} \cdot \rho_k \cdot g} + t_{\text{non-suction}} \right)} = \frac{Q \cdot \text{load}}{\left(\frac{\text{load}}{C_{vd} \cdot \rho_k \cdot g} + Q \cdot t_{\text{non-suction}} \right)}$$

This is a monotone ascending function. However the overflow losses cause an optimal flow rate for which the cycle production has a maximum. (Figure 2.24)

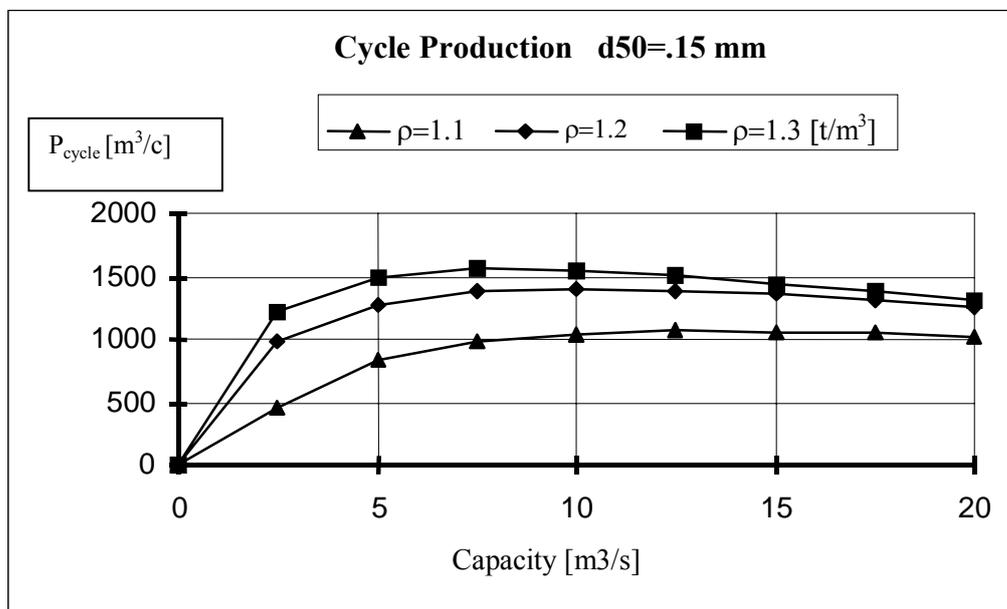


Figure 2-24

4. Also the pump capacity can be scaled from existing "well working" trailing suction hopper dredgers, by using the scale rule from Froude. However overflow losses will not be on scale when using this scale rule.

Above mentioned criterions lead to a design flow rate and a design density.

2.2.3.3 Suction pipe diameters

Old trailing suction hopper dredgers are equipped with relatively large suction pipe diameters. In the past the size of the diameter was mainly based on minimizing the pressure loss in the suction pipe to avoid cavitation of the dredge pump. However it was understood that the concentration distribution was homogeneous over the diameter, which is not always the case.

For a homogenous flow it can be shown that the suction production is maximum for a certain suction velocity. This is done with the so-called suction formula, a force balance over the suction pipe.

For a pump that is positioned k meters under the surface. The pressure at the suction mouth is $\rho_m g H$. The pressure in front of the pump p is equal to the allowable underpressure, vacuum, so $p = -VAC$.

The pressure difference over the suction pipe equals the weight of the mixture and the losses in the pipe.

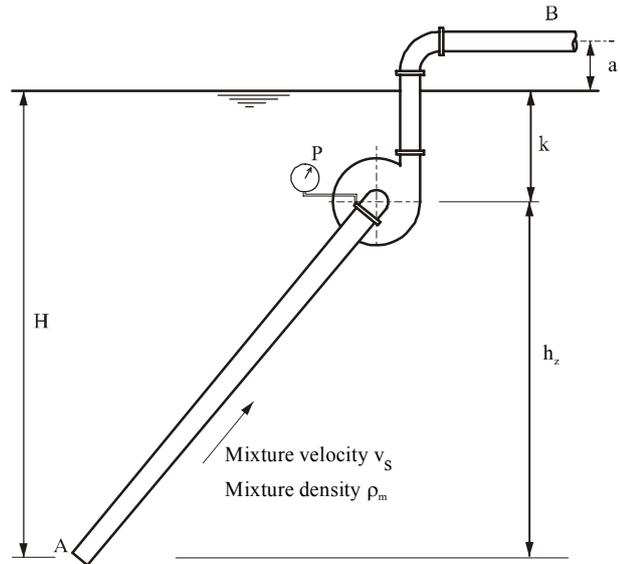


Figure 2-25

$$\rho_{water} \cdot g \cdot H + Vac = \rho_{mixture} \cdot g \cdot h_z + \xi \cdot \frac{1}{2} \rho_{mixture} \cdot v^2 = \rho_{mixture} \cdot g \cdot (H - k) + \xi \cdot \frac{1}{2} \rho_{mixture} \cdot v^2$$

$$\rho_{mixture} = \frac{\rho_{water} \cdot g \cdot H + Vac}{g \cdot (H - k) + \frac{\xi}{2} \cdot v^2}$$

$$Pr = Q \cdot C_{vd} \cdot \rho_k = v \cdot A \frac{\rho_{mixture} - \rho_{water}}{\rho_{grain} - \rho_{water}} \cdot \rho_{grain}$$

This function appears to have, dependent on H , k , V_{ac} and ξ , an optimum for a certain suction velocity v , which is independent of the suction pipe diameter.

ξ can be written as $\xi = \beta + \lambda \frac{L}{D}$ with;

β =entrée loss coefficient [-]

λ =Darcy-Weisbach resistance coefficient [-]

L =length of suction pipe in m

D =suction pipe diameter in m

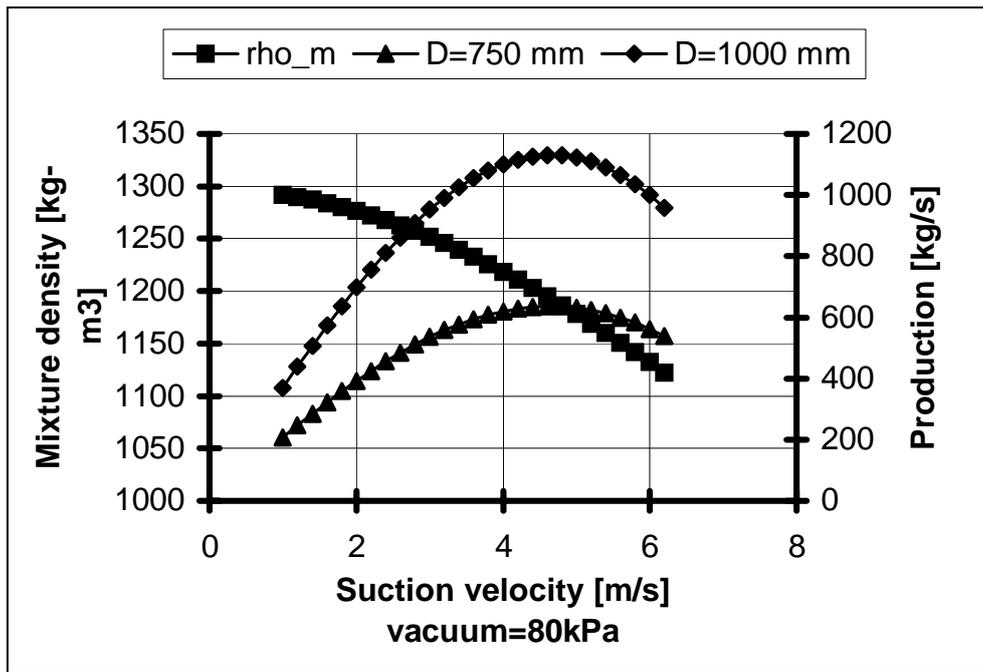


Figure 2-26

Application of the suction formula has several disadvantages:

1. The mixture density, the resistance factor ξ and the suction velocity are not independent of each other, but are determined by the erosion process and the pump characteristics.
2. The flow is only homogeneous for sand types with a $d_{50} < 0.15$ mm.

For coarser materials the flow becomes heterogeneous. As a result the volumetric concentration (the amount of sand in the pipe) increases and therefore also the pressure loss in the pipe. In other words the decrease of the pressure loss by the lower velocity is cancelled out by the increase as a result of the higher volumetric concentration. Therefore the pressure loss in the pipe does no longer behave according: $\Delta p = \frac{1}{2} \cdot \xi \cdot v^2$.

For this reason modern trailing suction hopper dredgers do have relative smaller suction pipe diameter then in the past. Besides that heavier pipes demand heavier winches, gantries and their foundations. This leads to a lower useful deadweight capacity and more investment cost.

Figure 2.27 below shows the relation between the maximum hopper volume and the suction pipes diameters for trailing suction hopper dredgers with two suction pipes. (diameters above 800 mm are round off to 100 mm and under 800 mm to 50 mm)

As can be seen in the Figure 2.27 the spread in the used suction pipe diameters is considerable. This could lead to the conclusion that design process is not yet unambiguous. At present however modern TSHD's have smaller in suction pipe diameter at the same flow rate. This is especially affected by the better insights in the two-phase flow at relative low velocities for inclined pipes.

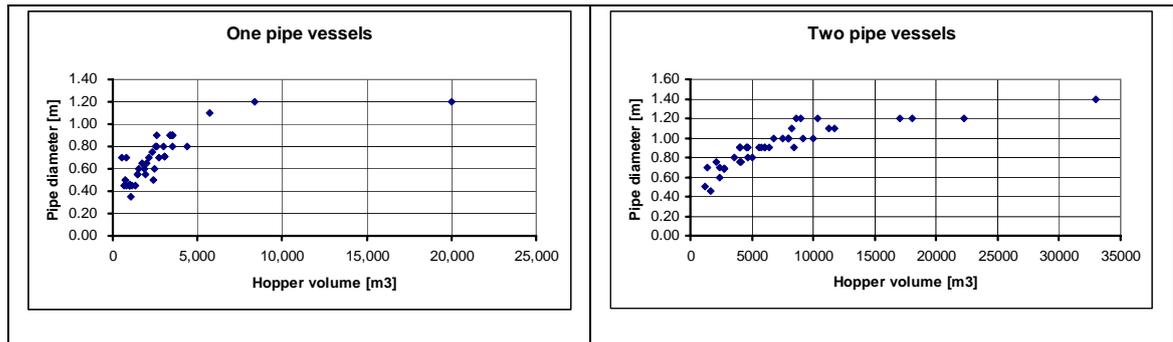


Figure 2-27

From many researches it appears that the velocity for which all soil particles in the pipe are still in motion is dependent on the Froude-value: $\frac{v^2}{g \cdot D}$. (v =velocity and D pipe diameter)

Depending on the grain size and concentration the Froude-value may not become less than a certain value $F_{I,H}$. Adding the maximum average velocities for which no stationary bed is formed in a horizontal pipeline can be calculated using $V_{sm} = F_l \sqrt{2 \cdot g \cdot (S_s - 1) \cdot D}$ or with the demi-McDonald of Wilson, which can be estimated with the formula:

$$V_{sm} = \frac{8.8 \cdot \left[\frac{\mu_s (S_s - S_f)}{0.66} \right]^{0.55} \cdot D^{0.7} \cdot d_{50}^{1.75}}{d_{50}^2 + 0.11 \cdot D^{0.7}} \quad \text{With } d_{50} \text{ in mm and the diameter } D \text{ in meters.}$$

In Figure 2.28 both formulas are drawn (Durant, $F_l=1.4$). For inclined suction pipes V_{sm} has to be raised with a value ΔD dependent of the incline. According Wilson and Tse ΔD reaches a maximum for approximately 30° and is then $\Delta D=0.333$ (Matousek, 1997).

In the design of trailing suction hopper dredgers usually $F_l = 1.00$ is assumed and ΔD is not considered. This implies that the dredger is designed for materials with a d_{50} between 100 and 300 μm and that for coarser materials a stationary bed is accepted.

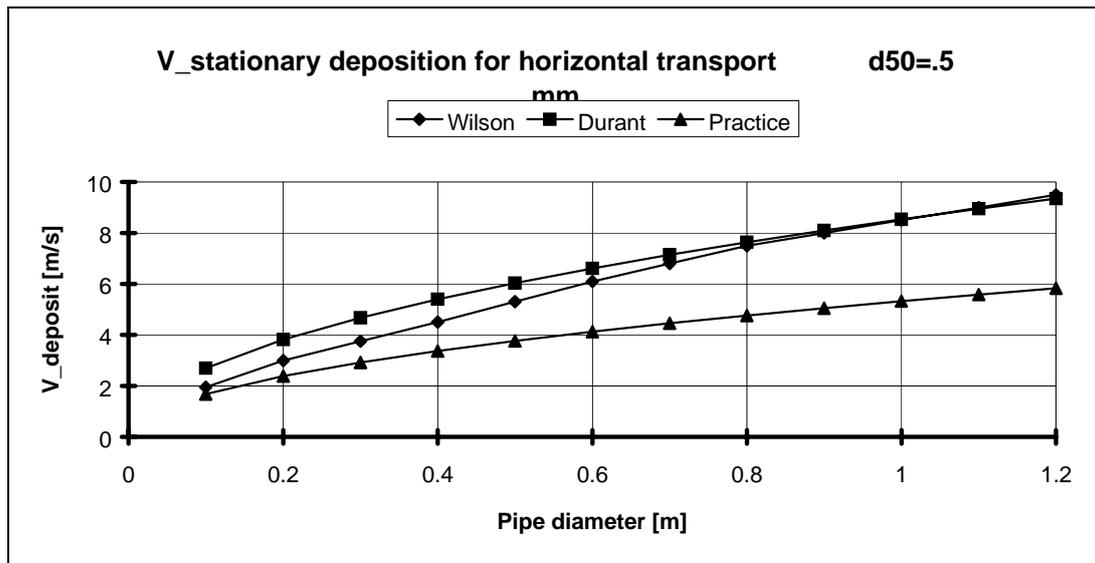


Figure 2-28

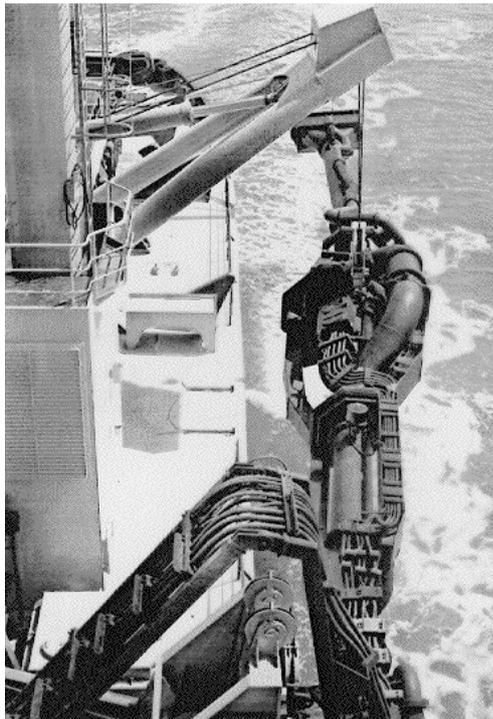


Figure 2-29 Dredge pump incorporate in the suction pipe

The use of suction pipe with a submerged pump (Figure 2.29) has a direct influence on the choice of the diameter of the suction pipe. Is this the case then it is possible to choose the suction pipe diameter a little smaller and so lighter and cheaper, against the disadvantage of a little additional pressure loss in the pipeline..

2.2.3.4 The pressure pipe diameter

The diameter of the pressure pipe should have a larger diameter than the suction pipe, because the factor 0.333 for the inclined transport. However often, depending of the value of the factor $F_{I,H}$, the pressure pipe diameter is chosen 50-100 mm smaller for costs reasons. Particular when the casted elbows and valves are used. The diameter of the pump ashore installation will generally be chosen smaller than the suction pipe. Normally the hopper is unloaded with considerable higher concentrations than loaded. This allow for a lower flow rate when discharge time equals the suction time.

2.2.3.5 The dredge pump

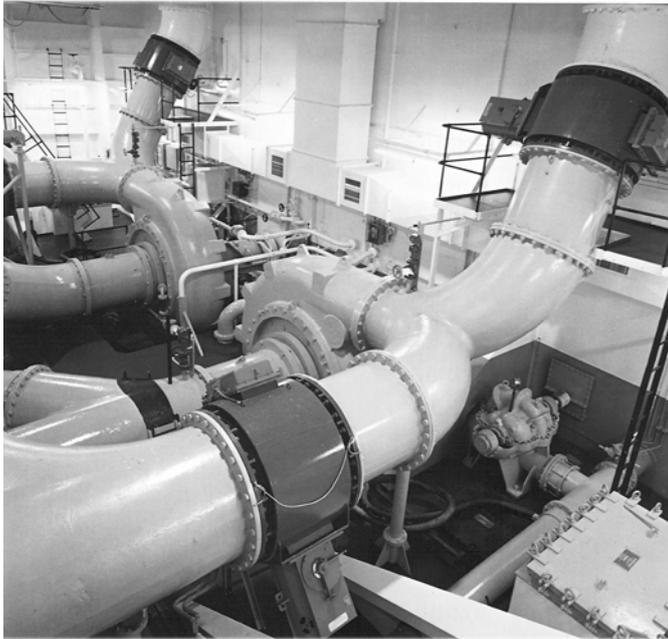


Figure 2-30 Pump room with 2 pumps

The main dimensions of the ship and the dredge installation are now known, so an estimate can be made to the required manometric head of the dredge pump for the different (un)loading conditions.

The required pump pressure during loading is determined by the static head from hart pump to the discharge in the hopper and the losses in the discharge line.

The manometric head is the sum of required pressure and the allowable vacuum at the suction side of the pump.

Because the impeller diameter is approximately known (minimum 2 times suction pipe diameter) and there is a relation between the required manometric pressure and the peripheral velocity of the pump impeller, also the specific pump speed is approximately known.

The dimensionless specific pump speed is defined as:

$$N_s = \frac{\Phi^{\frac{1}{2}}}{\Psi^{\frac{3}{4}}}$$

With:

$$\Phi \approx \frac{Q}{\omega \pi D b} = \text{dimensionless capacity}$$

$$\Psi = \frac{P}{\rho u^2} = \frac{P}{\rho \omega^2 r^2} = \text{dimensionless pressure}$$

In these is:

$$Q = \text{flow rate} \quad [m^3/s]$$

$$p = \text{pressure} \quad [Pa]$$

D = diameter pump impeller	[m]
b = width pump impeller	[m]
$r = \frac{1}{2}D$	[m]
ρ = density fluid	[kg/m ³]
ω = angular velocity pump impeller	[rad/s]

Filling in Φ and Ψ results in

$$N_s = \frac{\Phi^{\frac{1}{2}}}{\Psi^{\frac{3}{4}}} = \frac{\rho^{\frac{3}{4}} \omega \sqrt{Q}}{p^{\frac{3}{4}}} \cdot \sqrt{\frac{D}{4\pi b}} \quad (1)$$

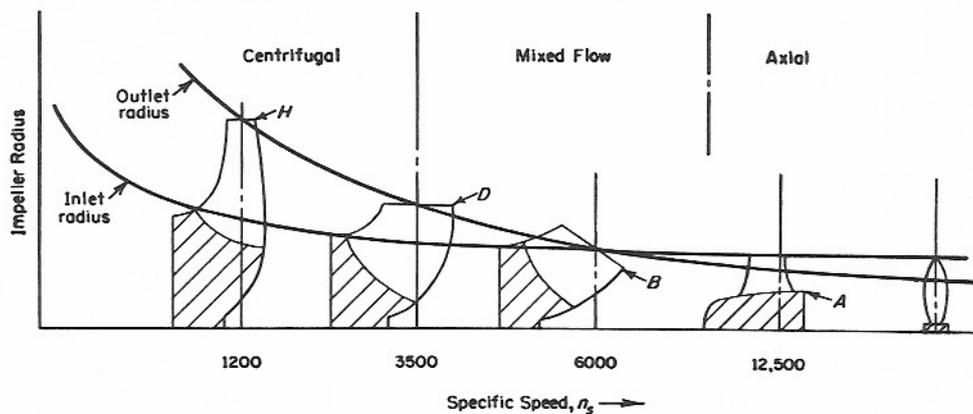


Figure 2-31

The specific speed is assessed to the maximum efficiency point and is a characteristic number to compare pumps with their dimensions like the b/D ratio, inlet and outlet diameter ratio D_i/D_u and impeller shapes (Figure 2.31). Equation (1) shows that for a constant number of revolutions (ω) the specific number of revolutions increases with an increasing flow rate and decreasing pressure. Since the pressure is proportional to the square of the peripheral velocity, the pressure will decrease at a constant number of revolutions with a decreasing diameter. A higher flow rate requires a larger diameter in the impeller, therefore a larger b/D ratio. Besides the b/D ratio especially a wider passage in the impeller has a large influence.

Figure 2.32 shows the relation between the dimensionless capacity and pressure as function of the number of revolutions for all types of hydraulic suction dredgers. Left in the chart are the standard centrifugal pumps and on the right the modern half-axial or mixed flow pumps, usually used as submerged pump in the suction pipe pump of trailing suction hopper dredgers and cutter suction dredgers. In general the dimensionless pressure for hopper pumps is slightly higher for the same specific flow rate than for the pressure pumps of cutter suction dredgers and suction dredgers.

From formula (1) it follows that when Q , p , and N_s are known, the pump speed can be determined, so that the pump and impeller type can also be chosen. (note: When the dredger will be equipped with a pump ashore installation, there will be two pump speeds.)

For relative small trailing suction hopper dredgers and suction depths a fixed pump speed for the dredging mode (suction) is often sufficient. When the difference between minimum and maximum dredging depth is large, a variable pump speed may be required.

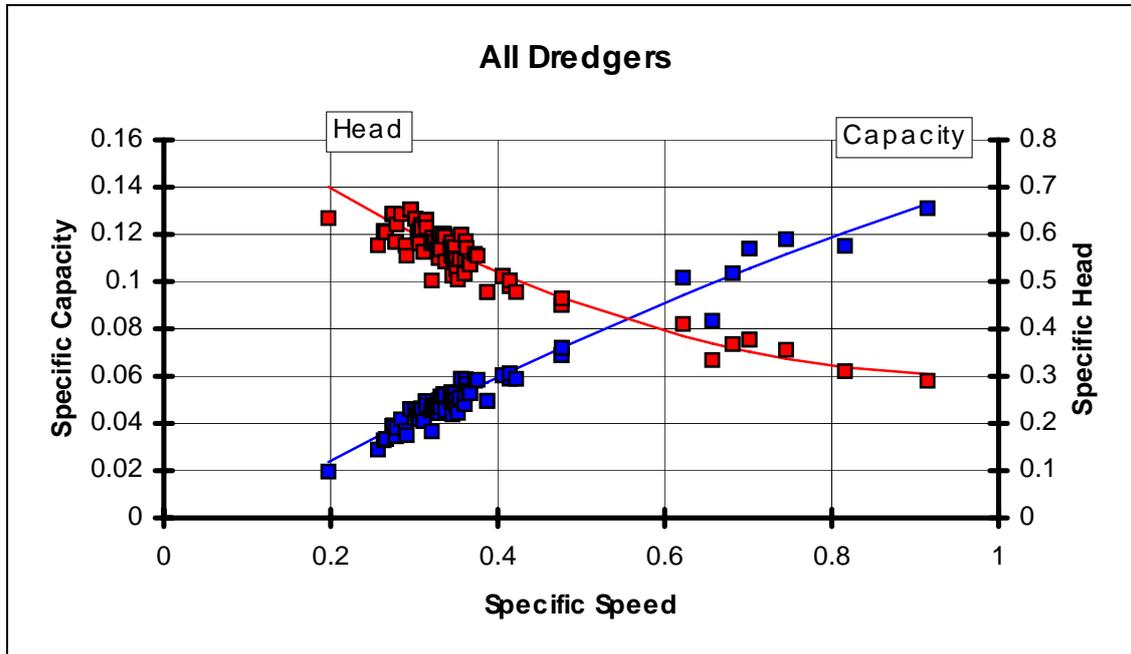


Figure 2-32

With increasing size and particular for increasing depth the question may rise if this can lead to large flow rate variations during the dredging process. Large flow rate variations often lead to water-hammer problems in the pipelines. If this risk exists than an adjustable pumpspeed is necessary.

There are more factors involved in the choice of a pump, such as:

- 3, 4 or 5 impeller vanes. Dependent on the required minimal opening area between the blades.
- Single- or double-walled pump (wear considerations).
- Inboard or submerged pump or both. If great suction depths are expected, it has to be considered if the installation of submerged pumps is more economical. The limit where this economical point is reached is closely connected with depth of the inboard pump below water level under service conditions, so roughly with the draught of the ship. This break point is therefore different for every ship.
- The operation of the pump during pumping ashore (if necessary).

When the dredger is provided with a pump ashore installation attention shall be given to the pumps working under both conditions. During pumping ashore it becomes more and more a custom that all available power of the main engines are used. This implies that the maximum pump speed when pumping ashore differs significantly from the pump speed during dredging. As a consequence the best efficiency point of the pump when pumping ashore shifts to a considerable higher flow rate than during dredging. This shift is in reality even larger because the pump ashore capacity is usually smaller than the flow rate during dredging (why?).

It has to be realized however that a pump working under conditions far above or below the best efficiency point, will wear faster. A good research of the position of the best efficiency points under the different service conditions is therefore necessary to obtain the optimal installation.

Also the required pump power for both modes can now be calculated. However, the maximum available pump power during pumping ashore is with a combined drive (one engine for pump + propulsion) determined by the required propulsion power.

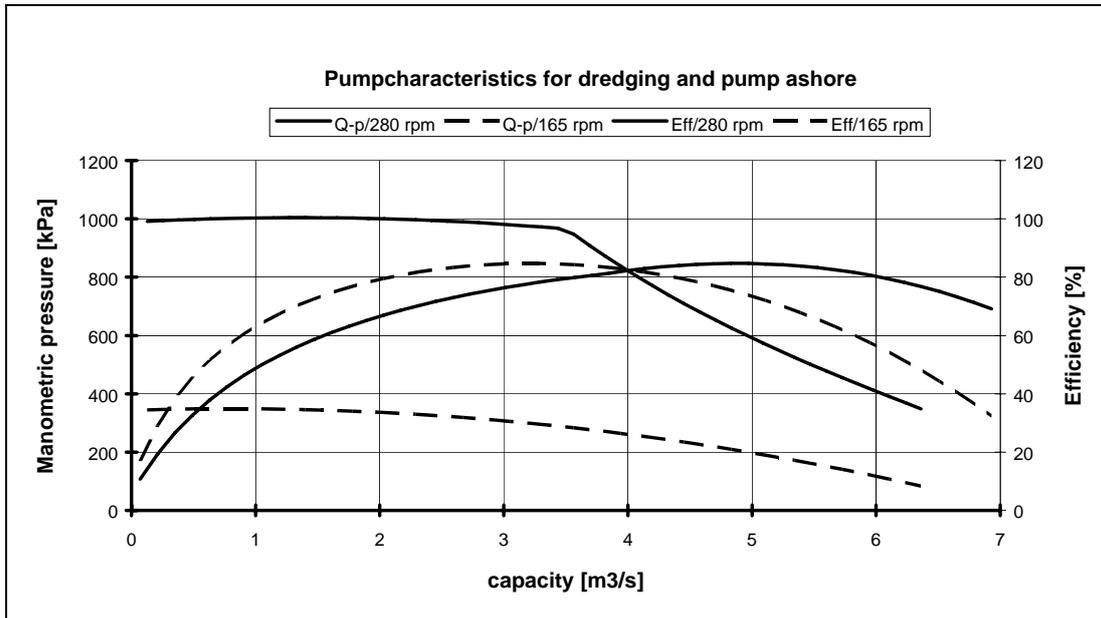


Figure 2-33

2.2.3.6 The dredge pump drive

Before choosing a drive the question should be answered whether continuous pump speed control is required or speed control by a gearbox is sufficient.

The following factors are involved:

- The expected range of the flow rate variation between the pumping of the water and of the slurry. This range is larger with an increasing suction depth, provided no cavitation takes place. Limitation of this variation can be necessary to reduce the risk of water-hammer. In that case a constant pump speed or a stepped control is insufficient.
- When a constant flow rate control is desired. The flow rate is regulated by a variation of the pump speed. An electric drive is necessary. A constant flow rate control by varying the number of revolutions is not suitable to prevent water-hammer (too slow).
- If the ship is equipped with a pump ashore installation and the propulsion power can be used totally or partly when pumping ashore. To use this additional power a higher pump speed than use in the dredging mode is required.

Dependent on these demands the sand pump can be driven directly by the main engine through a, if necessary, a stepped gearbox or directly by an electric engine through a generator. Of course there are several intermediate solutions that are treated in the chapter "Main arrangement".

2.2.3.7 The dragheads

Dragheads are designed to excavated the soil and mix it with water for hydraulic transport. Excavation can be done hydraulically or mechanically or combined. Hydraulic excavation is either by erosion of the dredge pump flow, by pressurized water jets or both

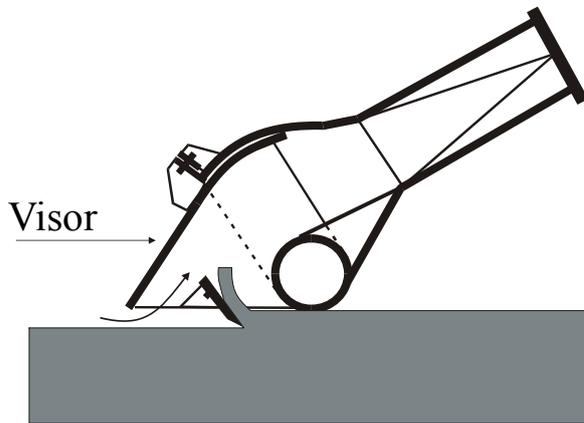


Figure 2-34 Draghead with blade

Pure mechanical excavation is mainly done in cohesive soils, such as clays and very soft rock. For that case teeth or blades are mounted in the draghead (Figure 2.34).

The width of the draghead is now dependent on the expected cutting forces in the particular soil in relation to the available cutting force from the propulsion. The length of the visor of the draghead should be chosen such the flow pattern for the transport of the excavated material suites the excavation process.



Figure 2-35 Draghead with jets (not working)

Modern dragheads have water jets assisted with knives or teeth.

A reasonable assumption is that the jet-production is linear with the total momentum flux of the jet system and independent of the trail speed. The momentum $I = \rho_w Qu$.

$$M_{\text{sand}} = \alpha \cdot I = \alpha \rho_w \cdot Qu = \alpha \rho_w \cdot Q \sqrt{\frac{2p_{\text{jet}}}{\rho_w}}$$

With:

- I = Momentum in N
- M_{sand} = Eroded sand mass in kg/s per jet
- p_{jet} = Jet pressure at the nozzle in Pa
- Q = Jet capacity in m³/s
- u = Jet velocity at the nozzle in m/s
- α = Coefficient depending on the particle size, jet pressure, jet capacity and trailspeed. A reasonable assumption for alpha is $\alpha=0.1$
- ρ_w = Water density in kg/m³.

When the nozzle are divided well over the width of the draghead the mass M should fulfill the relation:

$$\sum_{\text{all jets}} M_{\text{sand}} = B \cdot d \cdot v_{\text{trail}} \frac{\rho_{\text{situ}} - \rho_{\text{water}}}{\rho_{\text{particle}} - \rho_{\text{water}}} \rho_{\text{particle}}$$

B	=	Width draghead in m.
D	=	Eroded layer thickness in m
v_{trail}	=	Trailspeed in m/s
ρ_{situ}	=	Density soil in situ kg/m^3
ρ_{particle}	=	Particle density in kg/m^3

When the trailspeed is said to 1.5 m/s, which equals 3 knots and using the relation between pipe diameter and draghead width of Figure 2.36, d can be calculated.

In general the effective of the jet decreases somewhat with increasing pressure at constant momentum. This means that low pressure- high capacity jets are more effective than high pressure-low capacity jets. They use more specific energy too. On the other hand however, much jetwater dilutes the mixture density (Figure 2.128). So the designer has to search for the optimum solution between cost (power) en production

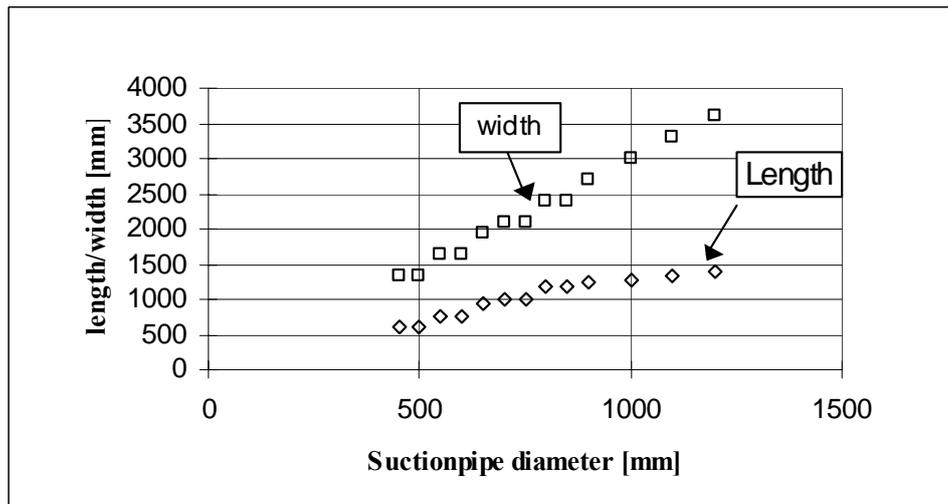


Figure 2-36 Dimensions Dutch draghead

2.2.3.8 The water pumps

Jet-water is used for loosening the soil within the dragheads, as well as to assist the process during discharging the load, either by dumping or by pumping ashore. The flow rate of the water pump is between 20 to 30 % of the sand pump flow rate and the pressure is usually between 5 and 15 bar. The required pressure can be calculated using the same basic formula's as mention in the forgoing chapter.

$$M_{\text{sand}} = C_{\text{vd}} Q_{\text{m}} \rho_{\text{sand}} = \alpha \rho_{\text{w}} Q_{\text{jet}} \sqrt{\frac{2p}{\rho_{\text{w}}}}$$

$$p = \frac{1}{2} \left[\frac{\rho_{\text{sand}}}{\sqrt{\rho_{\text{w}}}} \frac{C_{\text{vd}}}{\alpha} \frac{Q_{\text{m}}}{Q_{\text{jet}}} \right]^2$$

In general there is no requirement for speed control of the type of pump

2.2.3.9 The hopper

As mentioned before ships are built according certain L/B, B/T and B/H ratios. This also accounts for trailing suction hopper dredgers.

Some insight in the effect of these ratio's on the overflow losses is got from the Camps Diagram (Figure 2.132)

The removal Ratio R, the percentage of the incoming material that settles in the hopper is een function of:

$$R = f\left(\frac{S}{S_0}, \frac{S}{V_0}\right) = R = f\left(\frac{S(BL)}{Q}, \frac{S(BH)}{Q}\right)$$

The following conclusion from Figure 2.132 can now be drawn when keeping the hopper volume constant:

1. The width B is kept constant and L→2L and H→0.5H

1st term of the removal ratio shall increase and 2^e term shall decrease. This results in the conclusion:

More sedimentation at long shallow hoppers or less in short deep hoppers

2. The height H is kept constant and L→2L and B→0.5B

1st term of the removal ratio stays constant and 2^e term shall decrease. This results in:

A little less sedimentation at long small hoppers or little better sedimentation in short wide hoppers.

3. The length L is constant and H→2H and B→0.5B 1st term of the removal ratio shall decrease and 2^e term stays constant. This results in:

Less sedimentation in small deep hoppers or better sedimentation in wide shallow hoppers.

4. The height H and the width B are kept constant, while L→0.5L and Q→0.5Q

1st term of the removal ratio stays constant and 2^e term shall increase. This results in:

Central intake or a TSHD with 2 hoppers is a little better.

From the theory of the overflow losses (chapter 2.5.1.3) can be derived that long, shallow hoppers are favorable for the settlement process. Unfortunately such a shape leads to long relatively narrow ship with a limited depth that result in certain design problems for engine room en deckhouse. Therefore a compromise has to be found between the price and the performance.

When scaling-up the hopper shape to larger dimensions one should be aware for an undesirable increase of the overflow losses. After all for all new to build trailing suction hopper dredgers it is often demanded that the load time, independent of the size of the hopper, has to be 1 hour for a sand type with a d_{50} of 250 μm . This implies that the flow rate will be proportional to the volume of the hopper when the concentration is assumed constant.

Therefore the capacity scale is: $\eta_Q = (\eta_L)^3$

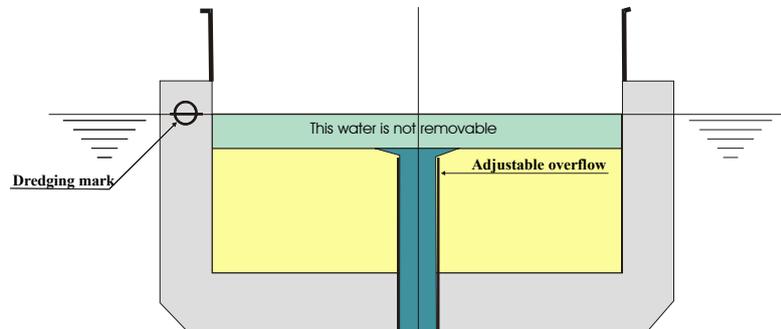
Both the terms $\frac{S(BL)}{Q}$ and $\frac{S(BH)}{Q}$ shall decrease and this implies that the overflow loss for

larger trailing suction hopper dredgers will be higher than for smaller trailing suction hopper dredgers, even if the hoppers are similar. Dependent on the magnitude of this increase this

could still be acceptable, since the cycle production can still be higher with higher overflow losses.

A design requirement directly related to the hopper shape is that the sand level at restricted loads needs to be higher than the sealevel.

Such a requirement is of importance in situations where it is not possible to dredge to the dredge mark because of the waterdepth. If the sea level is higher than the sand level, the water cannot flow out and the ship can't be loaded economically.



Constant Tonnage system

Figure 2-37

For modern ships this requirement can be satisfied for a 50-60% of the maximal load.

2.2.3.10 The discharge system

From the theory of the flow of bulk material from silos follows that a plane symmetrical flow will occur for discharge openings where length $L \geq 3B$ (width) and that this flow type, is preferred above an axial symmetrical flow. Unfortunately most discharge systems, except for the split hopper (Figure 2.38) don't satisfy this requirement, while the building of a split hopper suction dredgers is considerably more expensive than "single hull" ships.

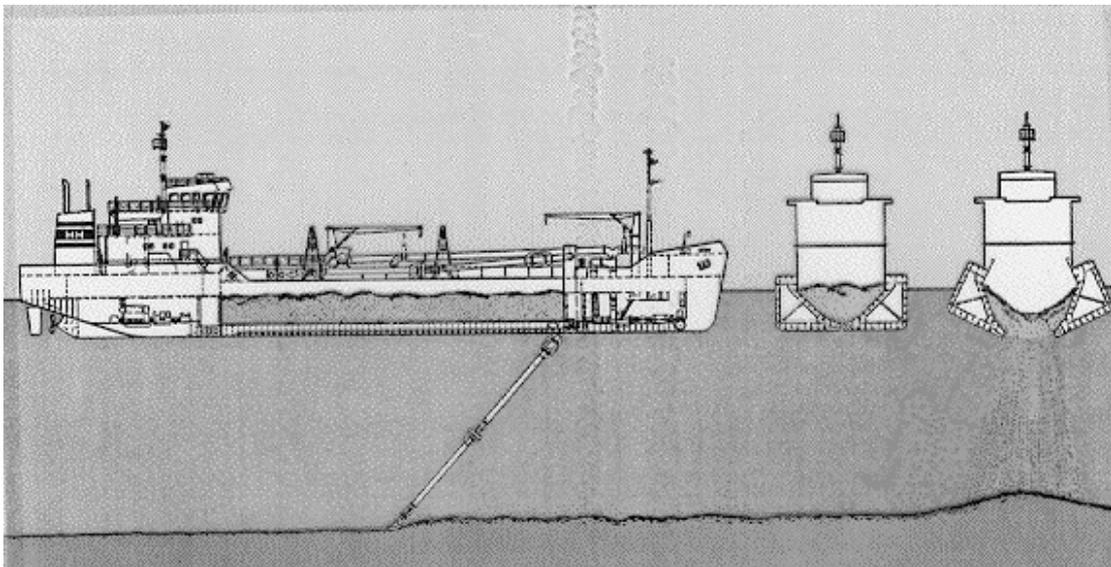


Figure 2-38 The split TSHD

As a rule of thumb the following ratios between the discharge opening and the well surface are used, dependent on the discharge material:

- for silt 10%
- for clay 50%
- average 30%

Instead of a large door or valve surface there are also systems that discharge the load with a limited amount of doors or valves by partly fluidizing or eroding the load. Experience showed that these systems function usually well for the fine sand types.

A design requirement for discharge system may be the necessity of dumping in shallow water. In this case than sliding doors or a splithopper are options. Also cone valves function well when discharging in shallow water. With a small opening they already provide a good discharge. If doors are used shallow dumping doors have to be considered

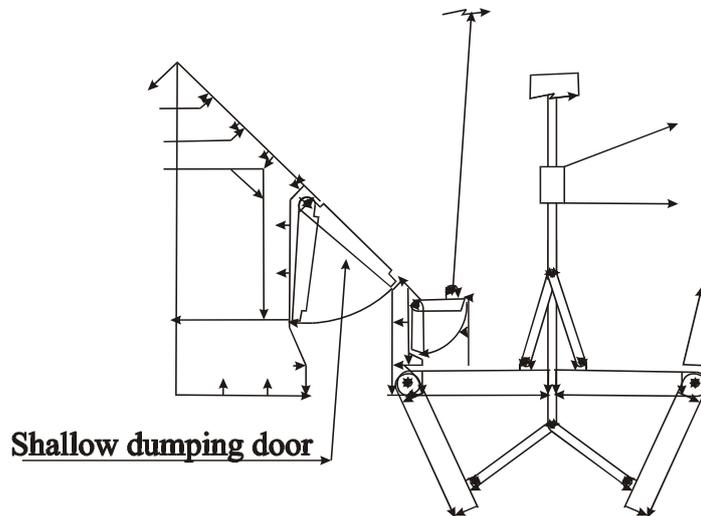


Figure 2-39

2.2.4 The propulsion power

Except for the propulsion there are also requirements for the maneuverability of the trailing suction hopper dredger. For this purpose extra bow thrusters are often used.

2.2.4.1 The propulsion power

Trailing suction hopper dredgers are real workships. They have a high block coefficient, no high ship velocities and they often sail in shallow waters, which make them "feel" the bottom. The velocities in knots do not exceed $1.4\sqrt{L}$ (Figure 2.40).

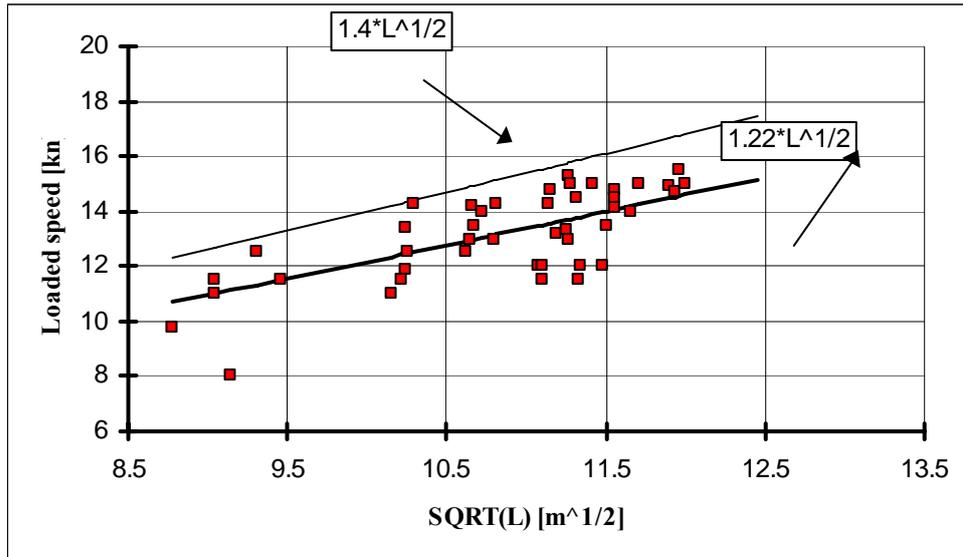


Figure 2-40 Maximum speed TSHD's

The ships resistance is composed of a number of components:

$$R_{total} = R_f (1 + k) + R_{app} + R_w + R_{TR} + R_A$$

with	R_{fl}	friction resistance according the ITTC-1957 formula	[N]
	$1+k$	shape factor for the hull	[-]
	R_w	wave resistance	[N]
	R_{app}	resistance as a result of the appendage	[N]
	R_b	resistance as a result of the additional pressure difference	[N]

Further is:

$$R_f = \frac{1}{2} \rho V^2 C_f S_{total}$$

with

$$C_f = \frac{0.075}{(\log_{10} R_n - 2)^2}$$

Determination of the resistance demands a lot of experience. The average sailing speed in knots for TSHD's is $1.22\sqrt{\text{Length}}$ ($0.63\sqrt{L}$ for v in m/s) Figure 2.40. That means that the wave resistance part is small and the total resistance can be estimated by a polynomial of the second order.

Nevertheless the ships resistance of a trailing suction hopper dredger is considerably higher under sailing conditions compared to normal ships with the same block coefficient. This is caused by the bottom valves or doors and the suction pipe guides in the hull.

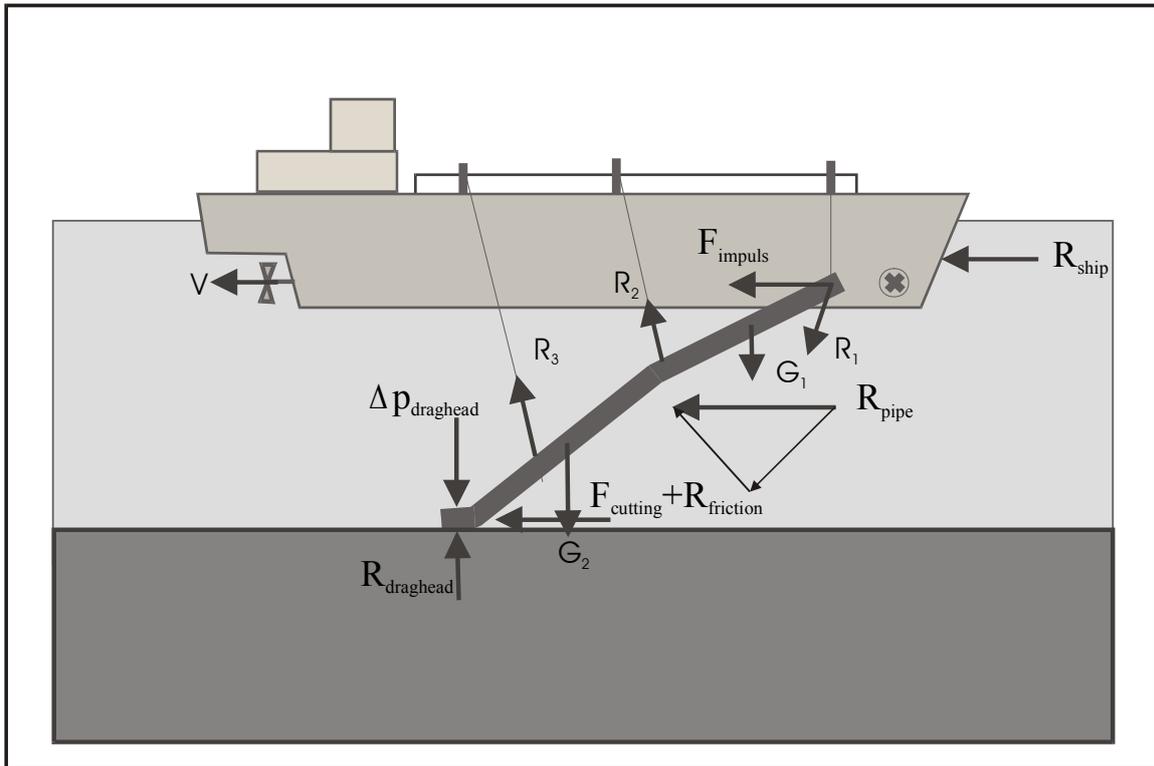


Figure 2-41 Forces working on a TSHD

The required propulsion power appears to be decisive under the trailing condition, in particular when a combined drive is used. For this condition requirements are set regarding the trail speed, expected counter current and effective cutting forces at the draghead.

For the trail speeds a normal value is 1.5 m/s with a counter current of 1 m/s. At these velocities the resistance of the hull, as could be expected, is little. The largest resistance arises from the dragging of the suction pipes over the seabed.

This suction pipe resistance is composed of several components:

The first, the hydro-visco components.

In the direction perpendicular of the pipe:

$$R_{\text{pipe} \perp} = C_D \frac{1}{2} \rho_w v \sin \beta \cdot |v \sin \beta| \cdot LD$$

In the direction parallel with the pipe:

$$R_{\text{pipe} \rightarrow} = C_L \frac{1}{2} \rho_w v \cos \beta \cdot |v \cos \beta| \cdot LD$$

In which:

C_D	= Drag coefficient	[-]
C_L	= Lift coefficient	[-]
D	= Pipe diameter	[m]
L	= Pipe length	[m]
$R_{\text{pipe} \perp}$	= Drag force	[N]

$R_{\text{pipe} \rightarrow}$	=	Lift force	[N]
v	=	Relative water velocity to the ship	[m/s]
β	=	Pipe angle	[°]
ρ_w	=	Density water	[kg/m ³]

The dimensionless coefficients C_D and C_L are apart from dependent on Reynolds number, also dependent on the appendages on the suction pipe. For a more accurate calculation it is better to divide the pipeline in different section with different projected areas. This has the advantage that the relative velocity of the water can be dependant of the waterdepth

Another force that the propulsion has to generate, which is often forgotten, is the force needed to accelerate the dredge mixture to the trail velocity of the ship, this momentum force.

$$F_{\text{Mom}} = Q \cdot \rho_{\text{mix}} \cdot v_{\text{trail}}$$

with:

F_{Mom}	=	Momentum force	[N]
Q	=	Pump capacity	[m ³ /s]
v_{trail}	=	Trail speed	[m/s]
ρ_{mix}	=	Density mixture	[kg/m ³]

The resistance of the draghead over the seabed.

This force is more difficult to determine, but it can be derived as follows:

During dredging erosion water shall enter the draghead at the backside and the sides. (See chapter 2.5.1.1.3) This pressure difference depends on the type of soil and the amount of jet-water used to loosen the soil (chapter 2.5.1.1). An average value for this pressure difference is 50 kPa. Multiplying the suction area of the draghead with the pressure difference gives the force that push the draghead to the seabed.

Additional to this is the weight of the draghead on the bottom, which can be determined with a simple equilibrium equation. The coefficient of friction of steel on wet sand is 0.3 to 0.5. Additionally it is known that the draghead "bulldozers". Therefore a coefficient of friction of at least 0.5 must be used.

Teeth or blades mounted in the draghead with intension to cut a significant part of the soil do increase the trail force significant. Effective trailing forces of 250 to 500 kN per pipe are common for the big dredgers

If the total resistance of the suction pipe is known than this can be roughly converted to other diameters using:

$$\frac{W_1}{W_2} = \left(\frac{D_1}{D_2} \right)^\alpha \quad \text{with } \alpha = 2.2 - 2.4$$

In conclusion the required effective trail force(s) are strongly dependent on the expected type of the dredging work and therefore to consider in detail during design.

The above consideration can be visually clarified in the resistance-propulsion power chart:

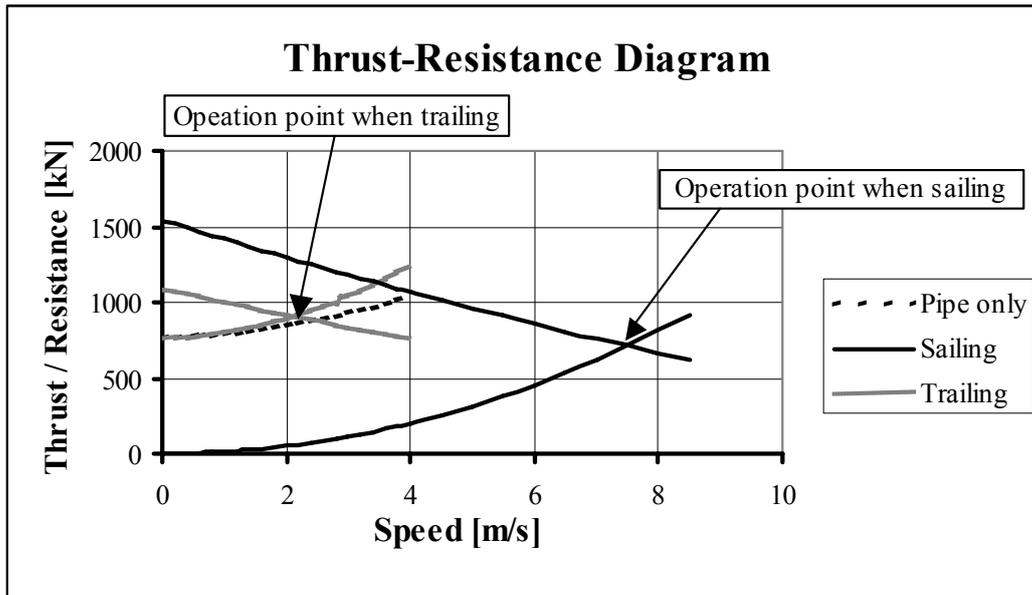


Figure 2-42

In Figure 2.42 the effective propulsion force (thrust), T_{sailing} (corrected for wake) as the ships resistance, R_{sailing} , are shown as a function of the ships speed. In the operating point "sailing" the supplied power is equal to the ships resistance. Under this condition the main engines are usually only driving the screws and the thrust curve is determined by the power of the main engines. This propulsion force curve can be described by a second-order polynomial:

$$T_{\text{sailing}} = a_0 + a_1 v_s + a_2 v_s^2$$

During dredging the main engines usually drive, besides the screws, also the sand-pump installation (sand- and water-pump) either directly or through a generator/electric motor set. This means that less propulsion is available for the propulsion in this mode. Because the propulsion force is proportional to the propulsion power as: $\frac{T^2}{P^3} = \text{constant}$, the propulsion force curve is approximated under dredging (trailing) conditions by:

$$T_{\text{trailing}} = a_0 \left(\frac{P_{\text{trailing}}}{P_{\text{sailing}}} \right)^{\frac{2}{3}} + a_1 v_s + a_2 v_s^2$$

The sum of the ships resistance (R_{ship}) and the suction pipe resistance (R_{pipe}) has to be equal with this propulsion force curve (operating point "trailing"). Usually this condition appears to be decisive for the to be installed power of the main engines. If no combined drive is used than the "sailing" condition is normative for the required propulsion power.

2.2.4.2 The bow thruster power

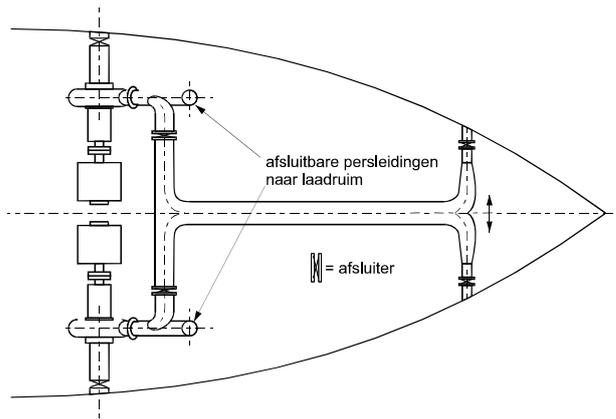


Figure 2-43 Bow jet

Maneuverability of THSD's has improved much compare to the past. In the sixties and the seventies the so-called bow jets (Figure 2.43) were used. These made it possible to generate a transverse force with the sand-pumps. But for practical reasons this was done only when the pump-room was positioned in the bow. The effectiveness of these jets is pretty good, certainly for 2 to 3 knots. The construction costs are only a fraction of those for a bow thruster.

However continuous use during dredging is not possible and so not economical.

Therefore this idea is abandoned and one or more bow thrusters are used. However bow thrusters have the disadvantage of hardly any transverse force above 3 knots. There are different types on the market.

A propeller mounted in a tunnel with a speed or pitch control, which means that the flow direction and capacity is control by the revolutions and speed direction or by changing the pitch of the propeller vanes. A axial flow pump by which the direction of the flow is control by valves and the capacity by the speed of the impeller.

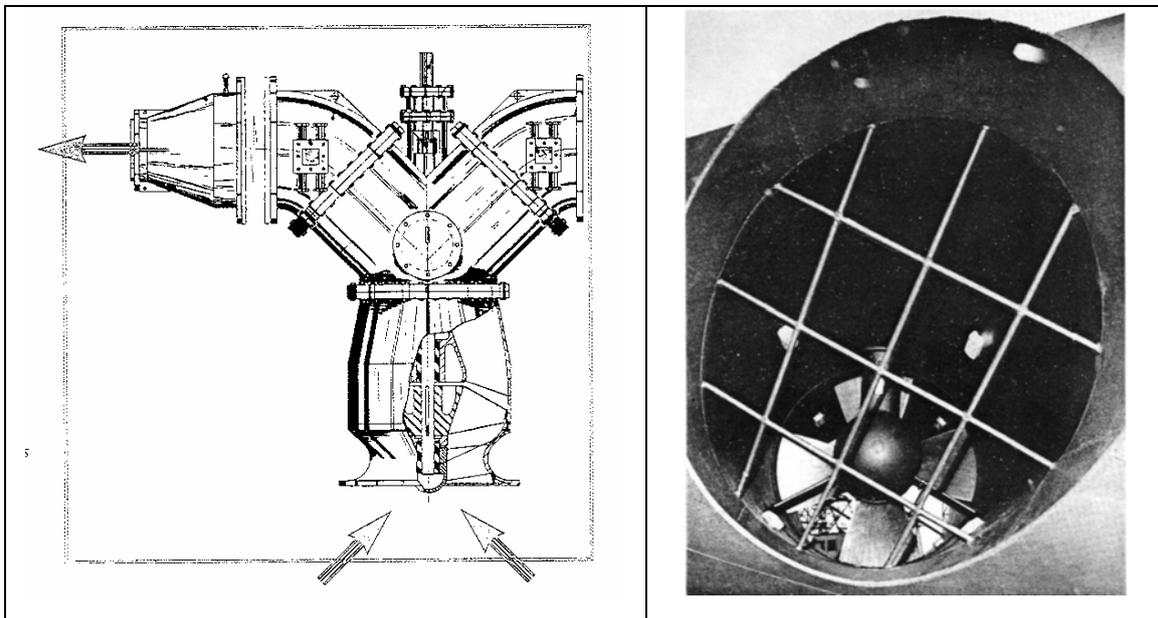


Figure 2-44 Thruster types

With the increase of the jet-pump power one could consider to use these, totally o partly, for the bow jets.

The required bow thruster power depends strongly on the expected type of work for which the trailing suction hopper dredger has to be designed.

2.2.5 Power balance

From the above mentioned it shows that a lot of power is installed in a trailing suction hopper dredger, that is:

- the dredge-pump power
- the jet-pump power
- the propulsion power
- the bow thruster power

and of course the power for the electrical circuit on board. After all the suction pipes have to be lowered and raised. The valves and other auxiliary equipment must operate, etc. Powers of 15000 kW or more are no exception. Therefore it makes sense to take a close look to the power balance. For instance, separate drives for the propulsion and the sand-pumps are not always necessary or desirable. Most of the time several objects can be combined. The following will show that this is strongly related to the suction pipe configuration.

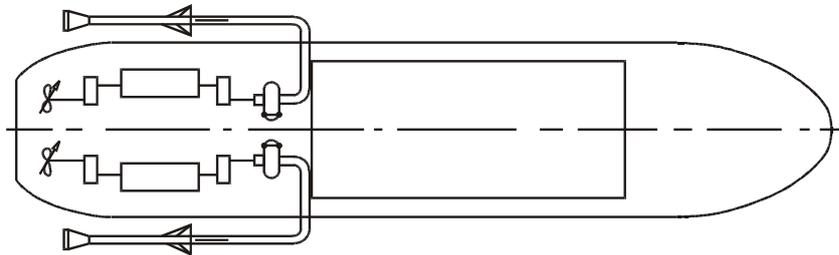


Figure 2-45 Direct drive

The most common combination is to drive both the propeller as well as the dredge-pump with one engine (Figure 2.45). The total installed power will not be much less than these units are separate as shown in Figure 2.46 but during sailing more power is available for a higher sailing speed and resulting in a higher production. If the units are driven directly, there will be no loss in generators, cables and electric motors. The speed control of the sand-pump is however poor. The engines run on constant speed, while adjustable propellers control the speed of the vessel, while the configuration of Figure 2.46 has fixed propellers (Why?).

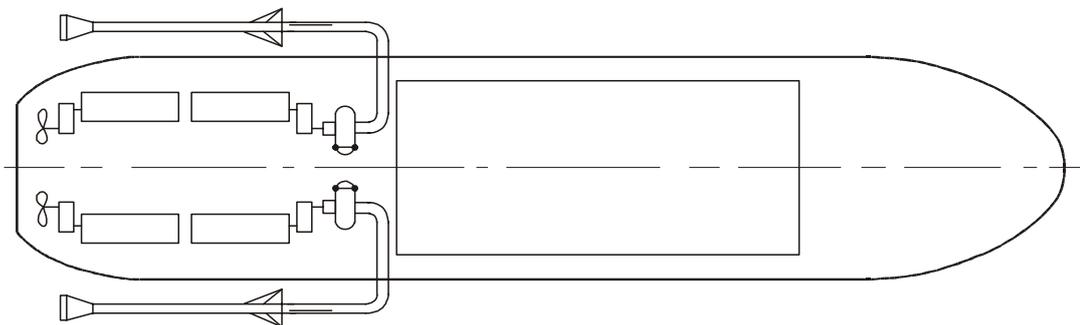


Figure 2-46 Separate propulsion and dredge pump engines

When the trailing suction hopper dredger needs pump ashore installation than generally an extra transmission is installed in the gear-box to use the total available power for this installation. The same engine supplies the jet-pump power usually. In that case the gear box is fitted with an extra axis. The only disadvantage for this arrangement is the limitation in the suction pipe length. Of course this is not totally black-and-white. Extending of the inboard

suction pipe offers the possibility to place a longer pipe on the deck, but this results in a lower production when dredging at large depths. Such a ship is put into service in 1992 and the concerned company (J.F.J. de Nul) took this decision intentionally.

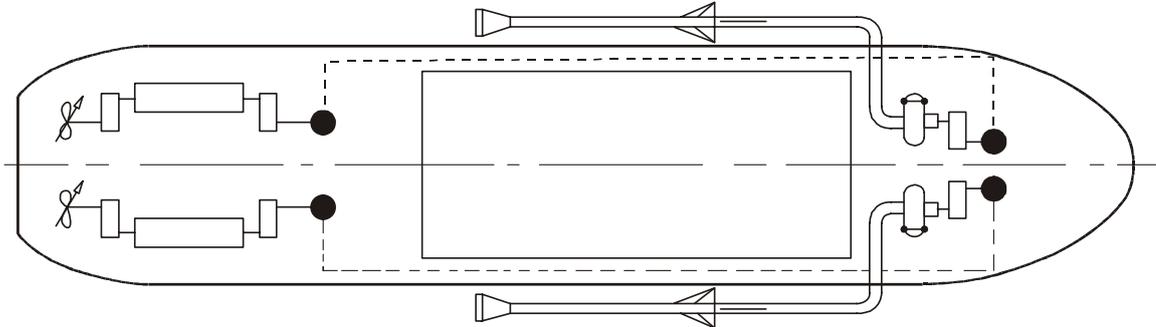


Figure 2-47 TSHD with dredge pumps in the fore ship

If limitation of the suction pipe length is not desired both powers can be combined with the arrangement of Figure 2.47. In the engine room the main engines drive the adjustable screw, but on the other side a generator is placed that supplies the dredge-pump placed in the fore ship with energy. This is attended by an energy loss of 10 to 15 % of the power required. So for a sand-pump power of 2000 kW times two, there is a loss of approximately 400 to 500 kW!

This also accounts for jet-pumps installed in the fore ship too. If the pump ashore installation needs the total power of the main engines this solution will require a considerable larger investment than the previous case. The speed control of the dredge pump can of course be well adjusted with an electrical drive.

Between these two solutions there are of course all kinds of variants possible, which have been built in the past too. (See chapter 2.26 Main Layout)

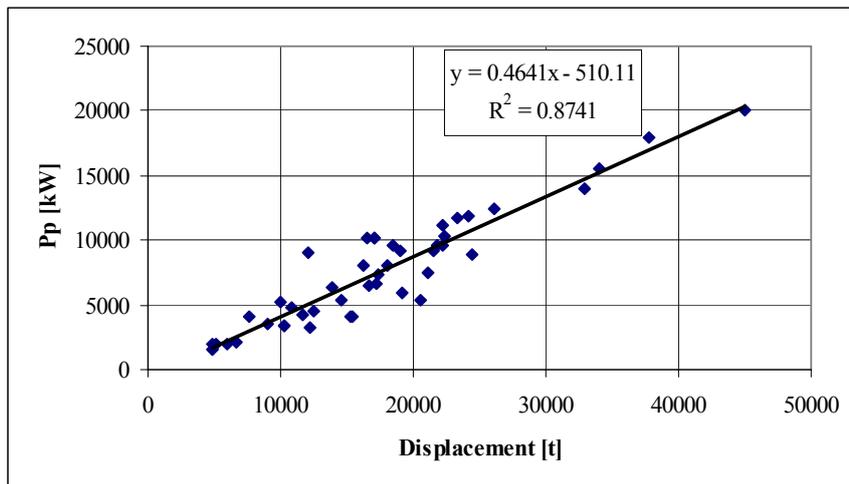


Figure 2-48 Propulsion power

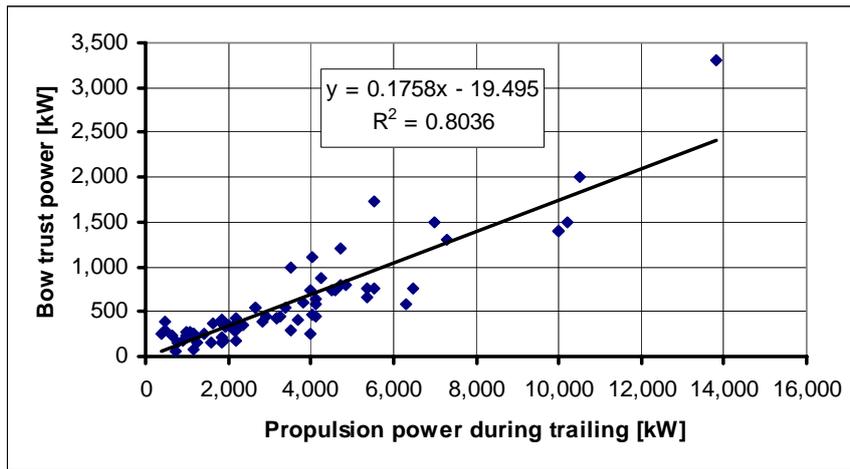


Figure 2-49 Bow thrust power

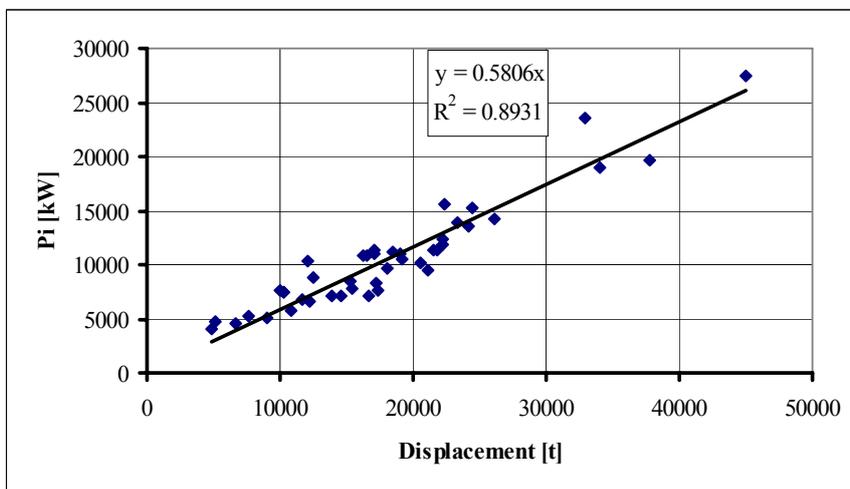


Figure 2-50 Total installed power

2.2.6 Main layout

Now the main dimensions of the ship and the dredging equipment are known, the layout of the ship has to be determined.

2.2.6.1 Single well ships

Most currently built trailing suction hopper dredgers are of the single cargo-hold type. The hopper, also called well, is positioned somewhat forward of the middle of the ship. This is also the case when the bridge is on the foreship. The engine-room is always positioned in the stern. The trailing suction hopper dredgers used by the dredging industry are usually equipped with two adjustable screws.

The position of the pump-room, a with watertight bulkheads sealed space in which the sand-pumps are located, also has a large influence on the layout of the trailing suction hopper dredger. The simplest and most efficient layout is the one where the pump-room is positioned just before the engine-room (Figure 2.45).

In this case the main engines drive both the adjustable screws as the sand-pumps. Adjustable screws are necessary in this case because if the sailing velocity of the trailing suction hopper dredger is controlled by varying the number of revolutions of the engine then also the production of the pump changes which can lead to production loss.

Since the sand-pumps on a trailing suction hopper dredger usually run on a fixed number of revolutions (variation of the suction depth has only a limited influence on the required head) the ships velocity can be easily adjusted by varying the pitch angle of the adjustable screws.

Of course adjustable screws are more expensive and vulnerable than fixed screws. If fixed screws are desired than the layout shown in Figure 2.46 is appropriate with different engines for the sailing and dredging.

An alternative for Figure 2.46 is Figure 2.51

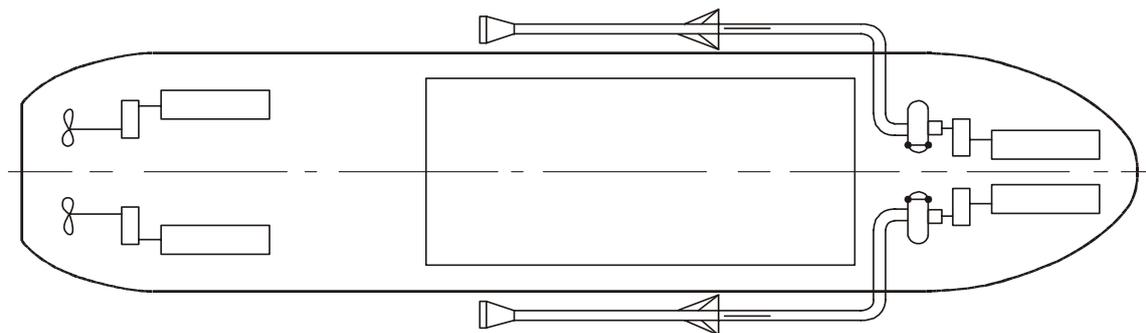


Figure 2-51

It will be clear that in the first solution the total installed power is better used. After all during sailing the full power of the engines is available for the propulsion. However these solutions are also seen with adjustable screws.

In both cases the limitation of the arrangement is the suction pipe length and therefore the suction depth. After all the suction pipes still need to be stored on board. If large dredging depths are also required (until ± 70 m) than the layout of Figure 2.47 and 2.52 are automatically

obtained. Figure 2.52 is called the All Electric Ship, an development of nowadays. All power needed is delivered by the main engines via high efficient generators and motors.

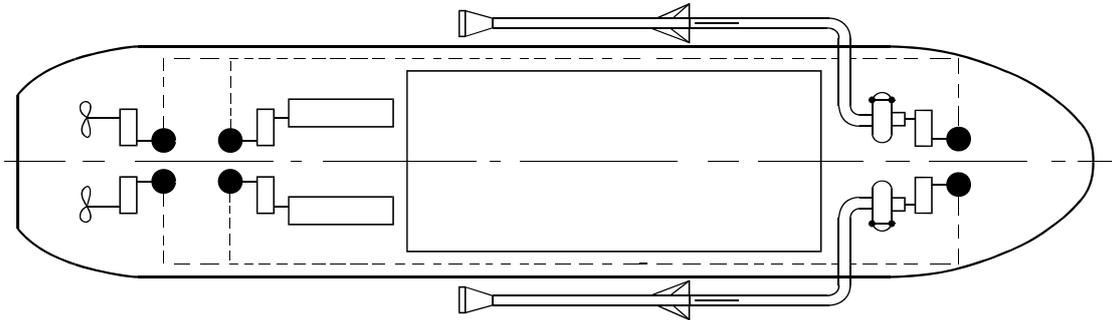


Figure 2-52 The all electric ship

Of course there are many combinations possible with these main layouts. The number of suction pipes may have some influence. Many smaller trailing suction hopper dredgers have only one suction pipe. Nevertheless these small trailing suction hopper dredgers are equipped with twin screws for two reasons:

1. The empty draught determines the maximum allowed propeller-diameter. Transferring a certain amount of power to one screw leads to a high revolutions, heavy loaded propeller with a relatively low efficiency.
2. A twin screw ship has a much higher maneuverability than a single screw ship

Nevertheless, special trailing suction hopper dredgers such as gravel dredgers, are equipped with a single screw (see special applications)

2.2.6.2 Twin Hopper Trailers

In the end of the sixties and starting seventies several trailing suction hopper dredgers were built with two separate hoppers. In these ships the engine-room and/or pump-room is positioned between the two hoppers. The main advantage of the twin hopper type is the smaller longitudinal ships bending moment that arises from the mid-ships connection of the engine-room and/or pump-room bulkheads.

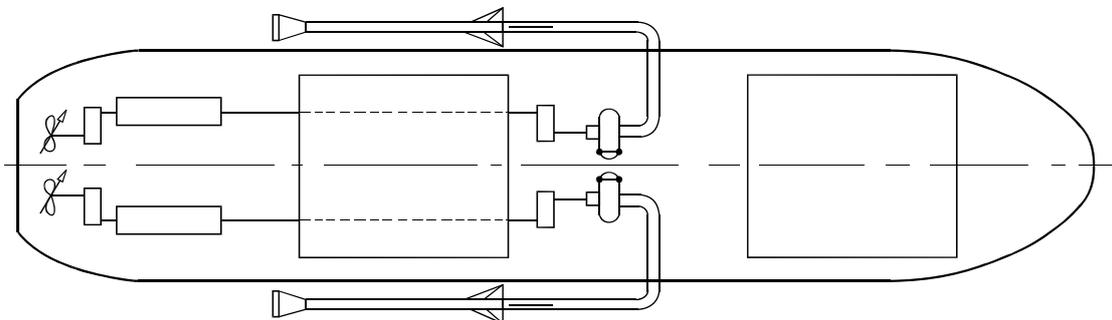


Figure 2-53

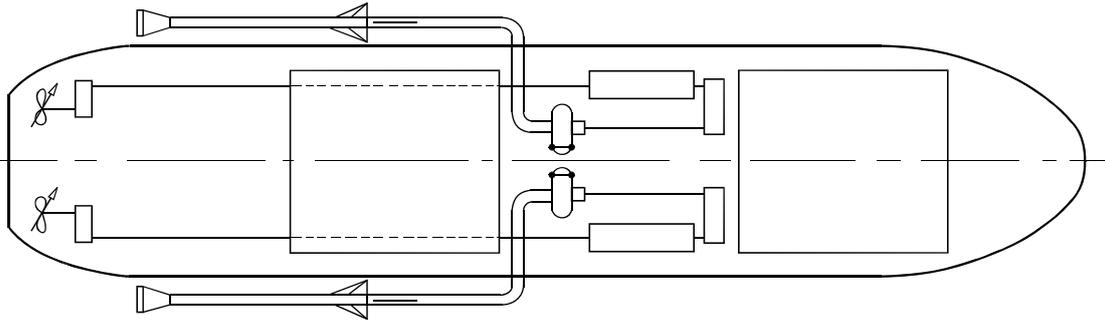


Figure 2-54

The disadvantage of such ships that on one hand the hopper ratios are unfavorable for the settling process and to the other hand the total capacity is divided over both hoppers which will improve the sedimentation process somewhat. Besides several extra valves are needed to trim the ship sufficiently. These layouts are shown in Figure 2.53 and Figure 2.54. The accommodation is also positioned amidships. In both cases the main engines drive propellers and dredge-pumps. Besides the longer pipes for large dredging depth can be installed. Of course an electrical driven dredging installation is possible too.

2.2.6.3 Single well ships with a submerged-pump

For larger suction depths, more than 50 m, the installation of a submerged-pump becomes economical. The submerged-pump, also called the suction pipe pump, can be driven electrical or hydraulically. The hydraulic drive exists on smaller trailing suction hopper dredgers.

On larger trailing suction hopper dredgers the pump and the electrical drive with bearings are accommodated in a compact compartment, directly mounted in the suction pipe. The number of revolutions of the electrical drive is chosen such that it corresponds with the required number of revolutions of the submerged-pump. This solution provides a compact and relative light construction.

The submerged-pump related possible layouts of the engine rooms and/or pump-rooms are shown in Figure 2.55.

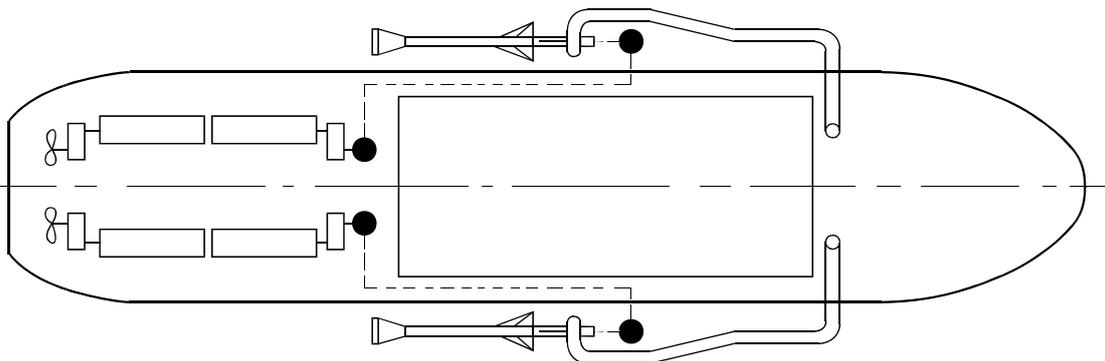


Figure 2-55 TSHD with inboard (direct driven) and submerged pumps

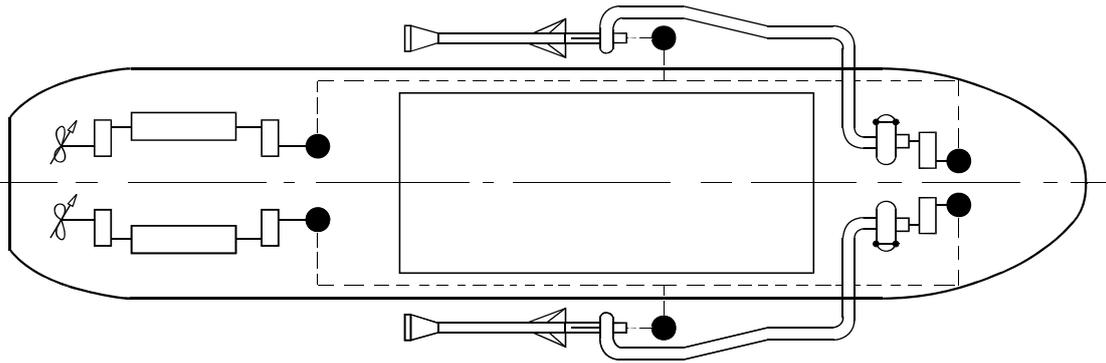


Figure 2-56 TSHD with inboard en submerged electrical driven pumps

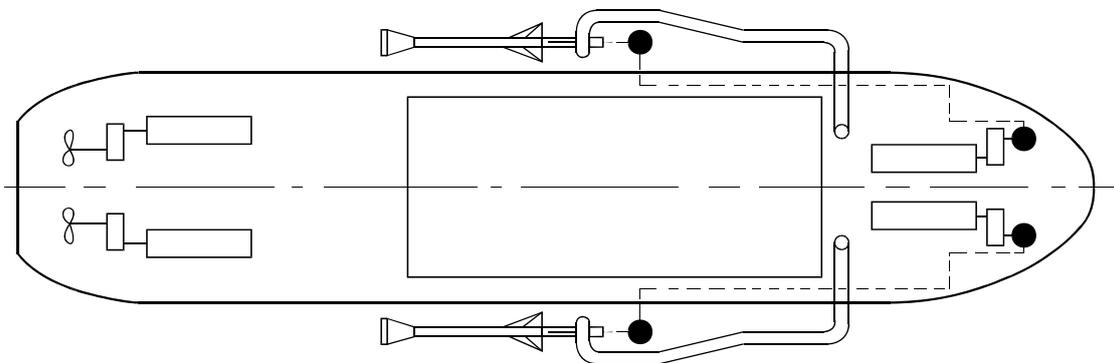


Figure 2-57 Electrical driven pumps and direct driven propulsion

For smaller, simpler trailing suction hopper dredgers and converted barges submerged pumps can be used to. For such ships the dredge installation is composed of modules (Figure 2.58). The drive unit of a dredge installation is now positioned on the fore-deck. The (existing) engine room is located in the stern. Therefore adjustable propellers are not necessary..

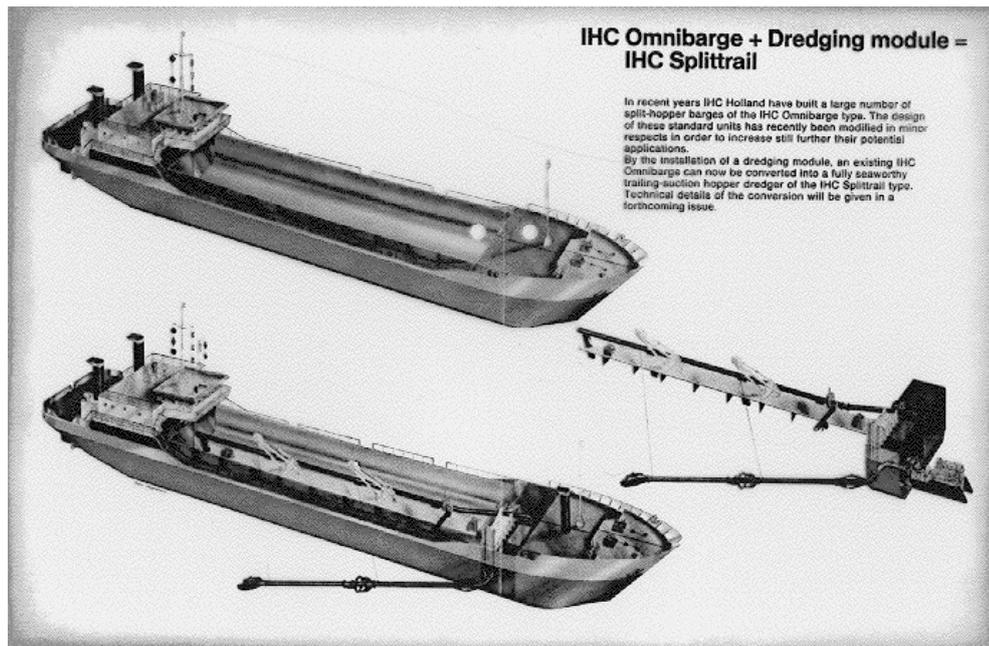


Figure 2-58 Pump module on barges

2.2.6.4 Split hopper suction dredgers

Split hopper suction dredgers can in principle also be divided as shown in Figure 2.59 and 2.60.

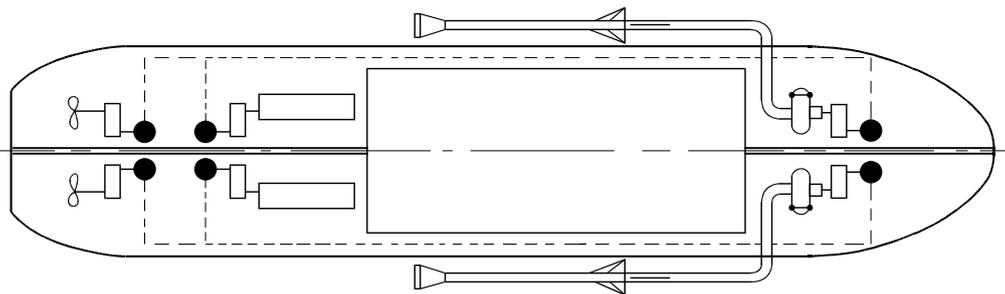


Figure 2-59

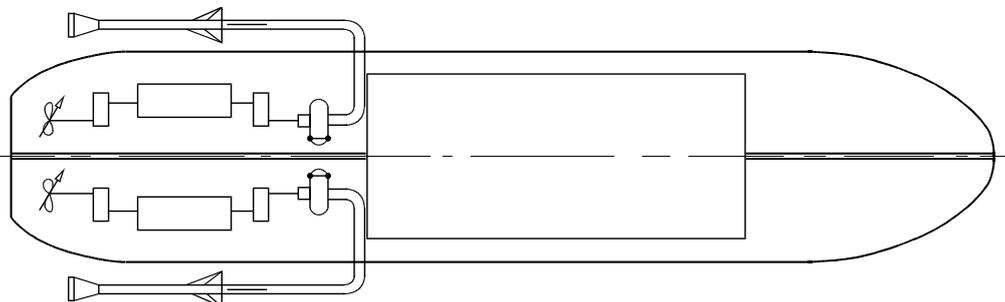


Figure 2-60

With the observation that both the engine-room and the pump-room are divided in the longitudinal direction (Figure 2.50 and 2.60).

The engine driver has to ascend to a height higher than sea level when he wants to go from starboard to portside.

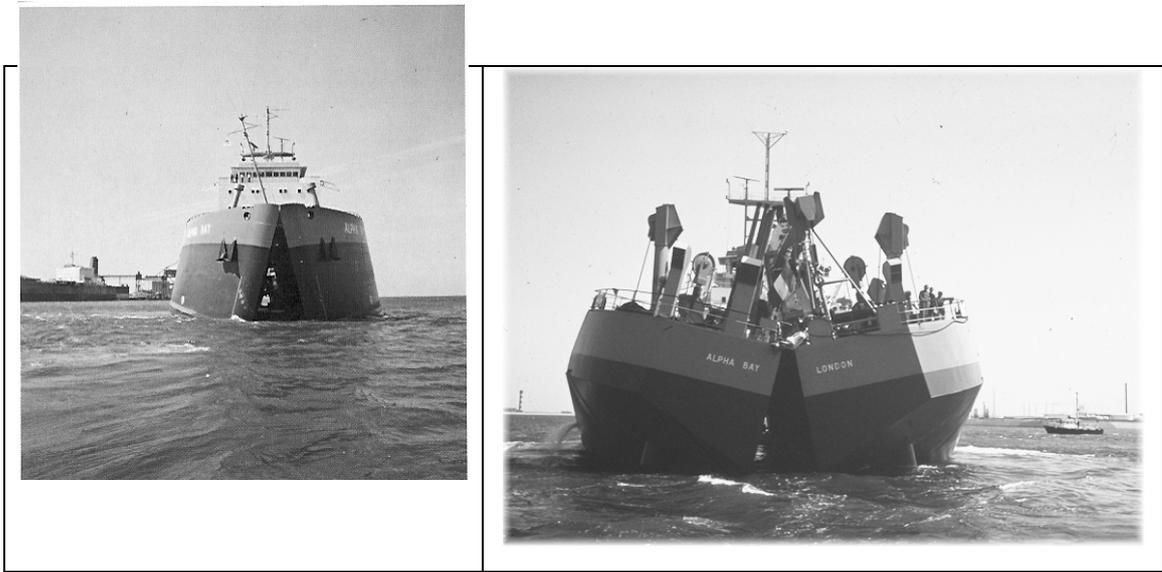


Figure 2-61 Split TSHD

2.2.6.5 The position of the pump-room

Positioning of the pumproom near the engine-room instead of in the fore-ship has the following advantages:

- the control and the maintenance of the pump installation can be done in a simple way by the engine-room personnel.
- with an empty ship the suction intake is submerged deeper than in the fore-ship, as a result of the trim.
- as a result of the shape of the stern the dragheads will move less frequently under the ship base, when working in shallow waters or on slopes.
- the direct drive of the sand pump by the main engine is considerable more efficient than the transport of energy from the stern to the fore-ship.
- the total propulsion power can be used easily for the pump ashore installation. With a fore-ship pumproom this requires considerable investments.

Of course there are also disadvantages:

- the main disadvantage of the pump-room near the engine-room in the stern is the limitation in the dredging depth of the suction pipe, something that has become more important in the last few years.
- the distribution of the weight is less ideal than with a pump-room in the front. For this reason the bridge is positioned on the bow nowadays.
- because the dragheads are nearer to the screws there is an increased chance for cables picked up to get entangled in the propellers.

2.3 Technical Construction

The technical construction of the trailing suction hopper dredger will be discussed in the flow direction of the dredging process.

2.3.1 The dredge installation

2.3.1.1 The dragheads

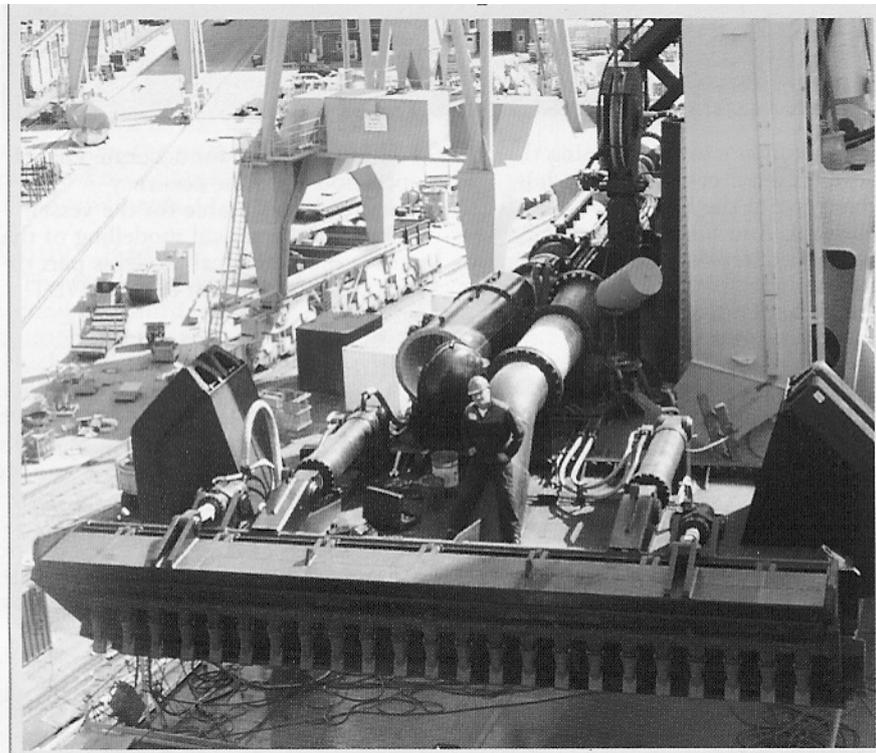


Figure 2-62 Modern draghead (Vasco da Gamma)

The draghead is the suction mouth of the trailing suction hopper dredger and is, with the sand-pump, one of the most important components of the dredge installation. Looking at the amount of patent applications on the area of dragheads the conclusion can be made that there is a lot of knowledge of the operation of this device. Unfortunately this is not the case, the last 5 years the remarkable progress made about the understanding of excavation process in the draghead.

Dragheads must be able to break up the coherence of varied soil types. The excavation process is done erosive, mechanical or by both methods.

Dragheads are designed to resist the forces, needed to loosen and suck up the soil. They also need to be strong enough to withstand collisions with unknown objects in the dredge area. This especially gives high demands on the reliability of the equipment mounted on the draghead to control the water supply and/or cutting blade depth.

In general draghead consist of a fixed part that is connected to the suction pipe, sometimes helmet mentioned and a one or two pivoting part(s), the visor, which is mounted in the fixed part. The last part is (self) adjusting to keep in fully touch with the seabed.

In the dredging industry different types of dragheads are used. The most known dragheads are:

- the Hollandse (Dutch) draghead, also called IHC draghead (Figure 2.63 and 2.65)
- the Californian draghead (Figure 2.64 and 2.66)

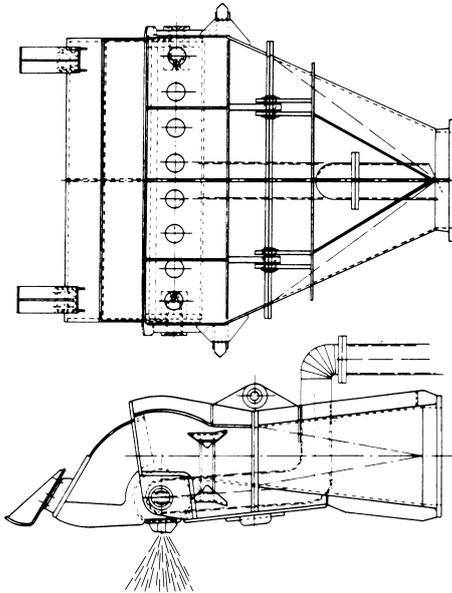


Figure 2-63 Dutch draghead

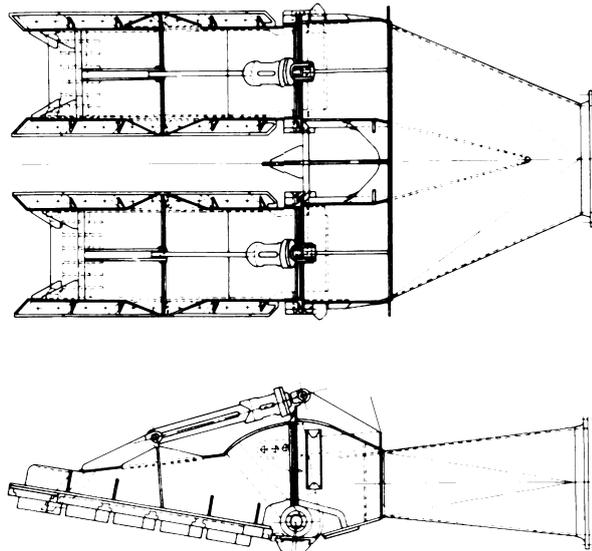


Figure 2-64 Californian draghead



Figure 2-65



Figure 2-66

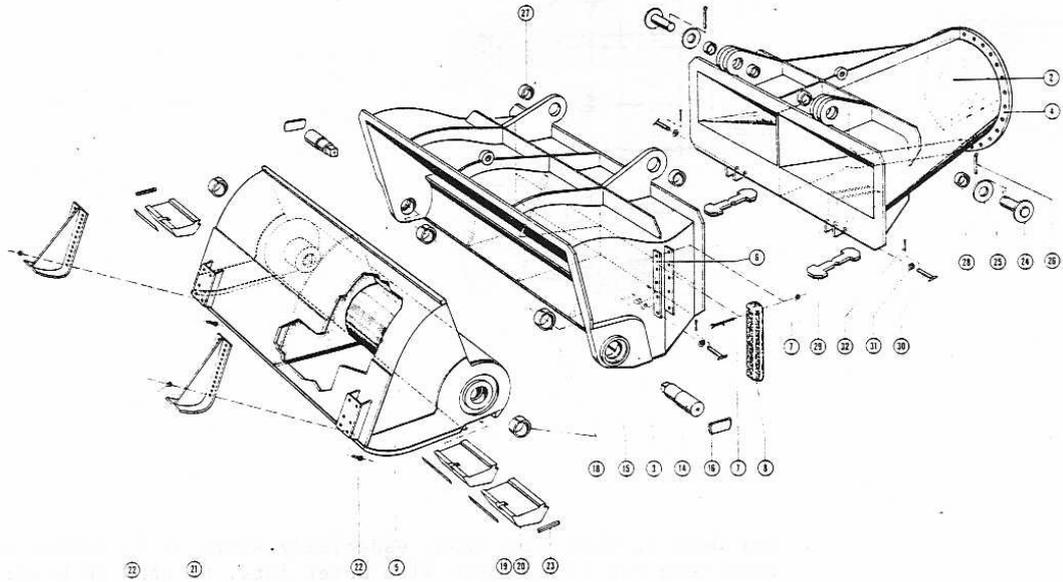
Both type are developed based on the principal of erosion generated by the dredge pump flow. Nowadays these dragheads can be equipped with water jets too (Fig 2.65 and 2.66)

In addition to the excavation of the soil, the jets are also important for the **forming of the mixture** in the draghead.

The dragheads rest on the seabed by means of replaceable, so-called, heel-pads of wear resistant material.

When dredging cohesive materials the dragheads are provided with blades or cutting teeth mounted in the visors. The position of the visor is fixed relative to the helmet corresponding with the average dredging depth. Sometimes this position is controlled by hydraulic cylinders.

When using tooth and/or blades one has to realize that different items can hook on to the draghead, causing high longitudinal forces in the suction pipe. This can be prevented by dividing the fixed part, **the helmet**, (Figure 2.67) in two parts, connected with a hinge on the top and breaker bolts at the bottom. The strength of the breaker bolts has to be slightly weaker than the weakest link of the several components of the suction pipe.



Parts List IHC drag head with divided fixed part	
Part No.	Description
1	Fixed part
2	Connector
3	Centrepiece
4	Welded flange
5	Visor
6	Strip for fender
7	Bolt and nut
8	Fender
9	Strip for fender
10	Bolt and nut
11	Fender
12	
13	
14	Shaft
15	Ring
16	Securing plate
17	
18	Bush
19	Heel wearing piece
20	Heel wearing piece
21	Shoe

Part No.	Description
22	Bolt and nut
23	Cover strip
24	Shaft
25	Securing plate
26	Split pin
27	Bush
28	Bush
29	Shear plate
30	Securing pin
31	Washer
32	Split pin
33	Stop
34	Bolt and nut
35	Strip for stop
36	
37	Screen with knife support
38	Stone trap
39	
40	
41	Knife piece
42	Bolt and nut
43	Circlip

Figure 2-67

However, if they are designed such that they fail regularly than soon the two parts are welded together with the danger that the next link fails.

Modern dragheads have one visor with jet nozzles over full width. At the backside of the visor replaceable teeth are fitted. The purposes of these teeth are to remove not eroded sand bands and to guide the flow in the direction of the suction pipe. Some of those dragheads do have movable water flaps to control the diluting water to the draghead. Visors can be adjusted either by bars or by hydraulic cylinders.

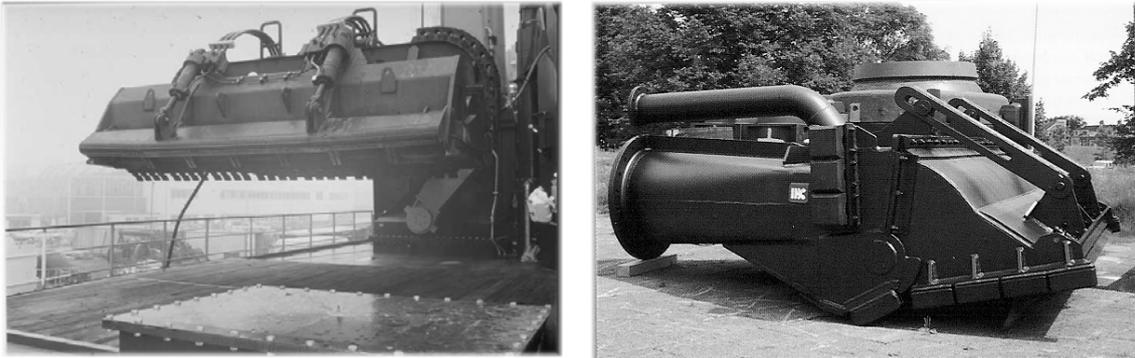


Figure 2-68 Modern dragheads

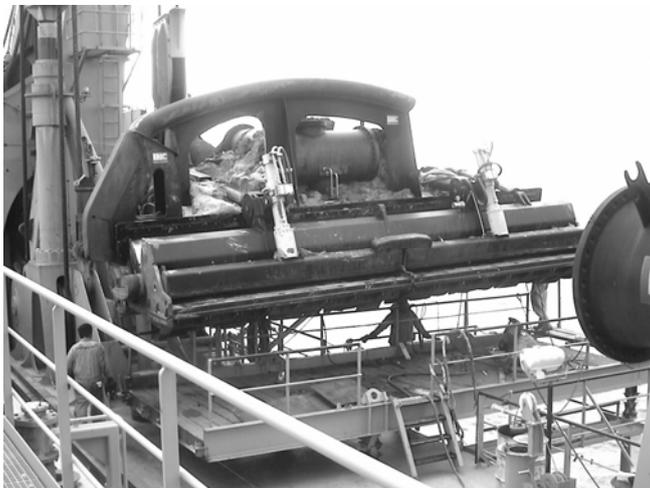


Figure 2-69 Fender for protection

Furthermore fenders are mounted on the draghead, to prevent damage caused by the bumping of the draghead against the hull. By mounting these fenders on both sides, the draghead can be used both on starboard and port.

Figure 2.69 fenders of the draghead of the One piper TSHD Volvox Terra Nova

The connection between the movable visors and the fixed helmet is usually sealed with a rubber strip. This prevents the entering of "strange" water and it decreases the wear caused by the sand picked up by this "strange" water.

2.3.1.1.1 Other types of dragheads

In the last 25 years a lot of experiments are performed with several types of dragheads, like:

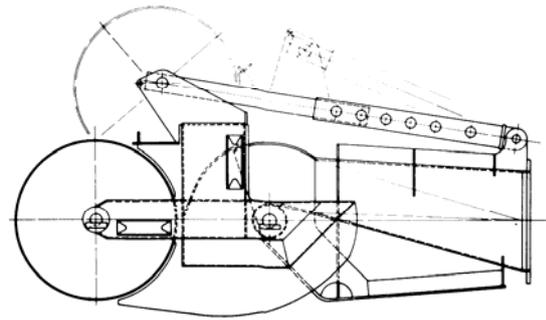
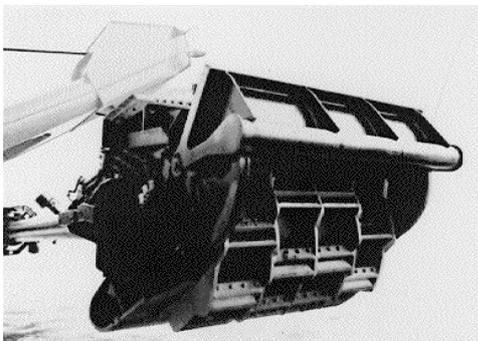


Figure 2-70 Silt draghead

The **silt head** (Figure 2.70). A draghead specially designed for dredging silt and soft clays. The silt is pushed in the draghead, while the propulsion delivers the required force.



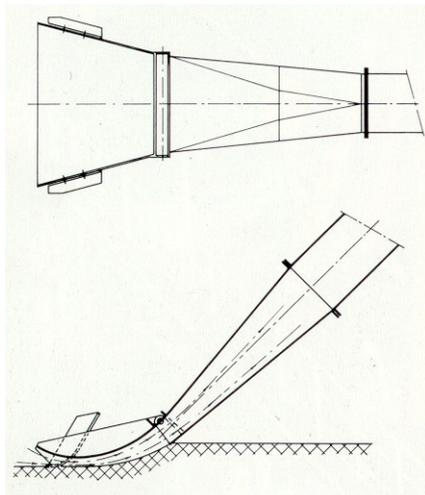
The **active draghead** (Figure 2.71)

A draghead with a hydraulic driven roller with cutting tools, able to cut firm clay or compact sand.

The disadvantage of this dragheads was the ability to pick up cables and wires

Figure 2-71 Active draghead

And



The **venturi head** (2.72). A draghead that would be hydraulically better shaped than the Hollandse and the Californian draghead and therefore would reach higher productions.

The advantage of this draghead was the high trailing force due to the pressure difference over the draghead

Figure 2-72 Venturi draghead

All these dragheads were not successful. Mostly the idea behind was good, but secondary reasons. like wear, sensitive for dirt, difficult to handle, etc. etc. Resulting in lower average productions than the earlier mentioned dragheads.

Aside from IHC there are also other companies that supplies dragheads. Usually these draghead are named after the company since they differ somehow from the standard dragheads. Examples are the "Van de Graaf-heads" and "VOSTA." heads

Furthermore every dredging company with self-respect has developed its own draghead, whether or not used.

2.3.1.2 The suction pipe

1. Suction bend, complete with arms.
2. Suction hose.
3. Intermediate pipe with arms.
4. Upper pipe section.
5. Intermediate pipe with arms.
6. Suction hose.
7. Universal joint.
8. Intermediate pipe with arms.
9. Turning gland.
10. Lower pipe section.
11. Draghead.

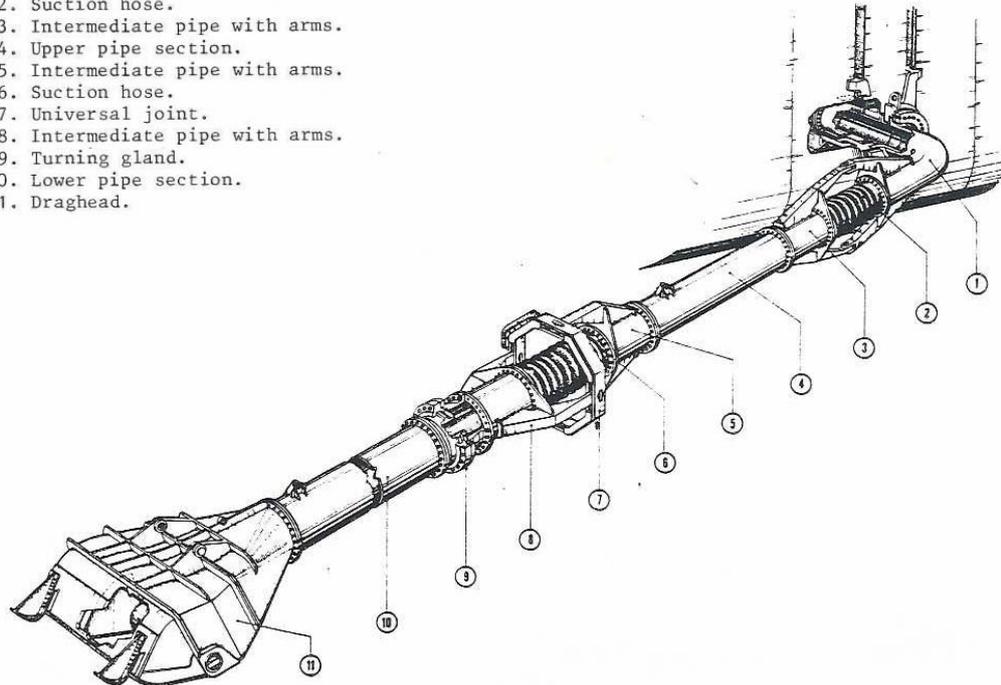


Figure 2-73 Suction pipe

The purpose of the suction pipe (Figure 2.73) is to make a connection between the seabed and the ship in order to make transport of dredge slurry possible. Because a fixed connection is not possible due to a varying water depth and the forces in size and direction, they have to comply with a number of important requirements:

- the dredging depth must be adjustable.
- there must be enough freedom of movement to maintain the connection with the seabed as good as possible.
- the bending moments due to the forces acting on the pipe should be kept as small as possible for reasons of strength and weight
- hit- and shock load resistant.
- a small pipeline resistance for the mixture flow.

The **trunnion slide** (Figure 2.74) that slides between the **hull guides** during the raising and lowering of the suction pipe, is fitted with tapered cams that push the trunnion slide against the hull when the suction pipe is in front of the suction intake.

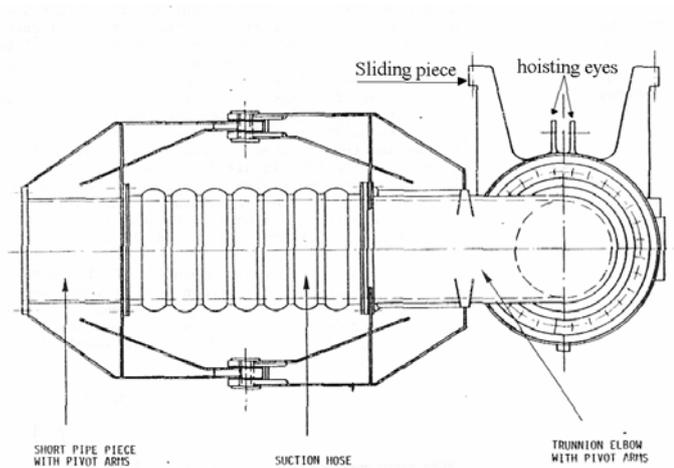


Figure 2-74 Trunnion slide with elbow

Mounted on this trunnion piece is a casted **elbow**, which can rotate around a horizontal axis, perpendicular to the hull. This hinge construction allows the suction pipe to be lowered to the desired depth. The elbow has two **arms**, positioned in the vertical plane of the suction pipe. On these arms, the upper or short piece pipe is mounted with hinges. This upper hinge makes the bending moments small, for example for the case

where the ship is swayed aside by the current.

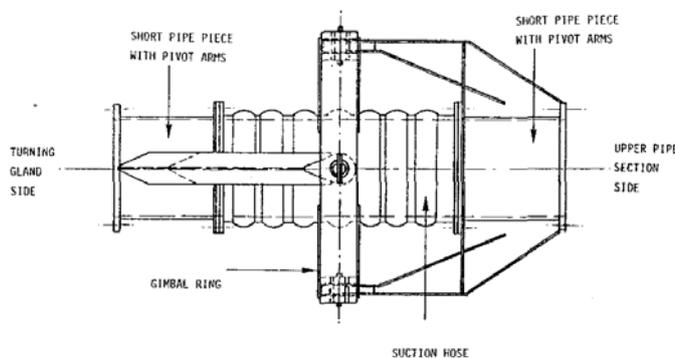


Figure 2-75 Universal joint

Between the elbow and the upper pipe a rubber **suction hose** is mounted that can move 40° to both sides. Steel rings are vulcanized in this suction sack to prevent a collapse of the suction sack by the subpressure as a result of the suction. The upper pipe is connected with the **lower pipe** by the **gimbal** (Figure 2.75) and a second

suction sack. This gimbal allows the two pipes to move independently, which is necessary in heavy weather and/or an irregular sea bottom.

A **turning gland** (Figure 2.76) is mounted, usually directly behind the gimbal, in the lower pipe. This allows the lower pipe to rotate around its longitudinal axis, so that the draghead can also follow the bottom profile in the transverse direction.

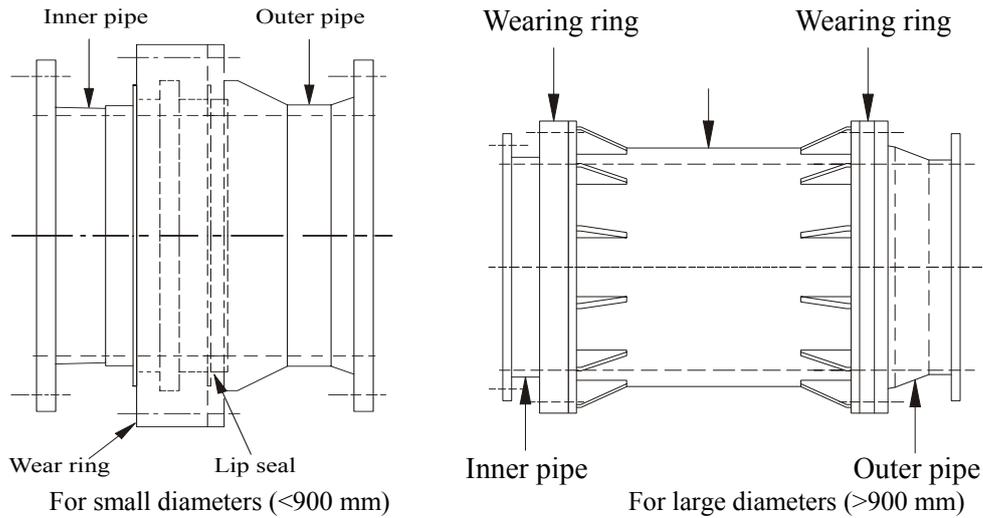


Figure 2-76 Turning glands

If the draghead is fitted for jet-water, a **jet-water pipeline** is mounted along the suction pipe (Figure 2.77).

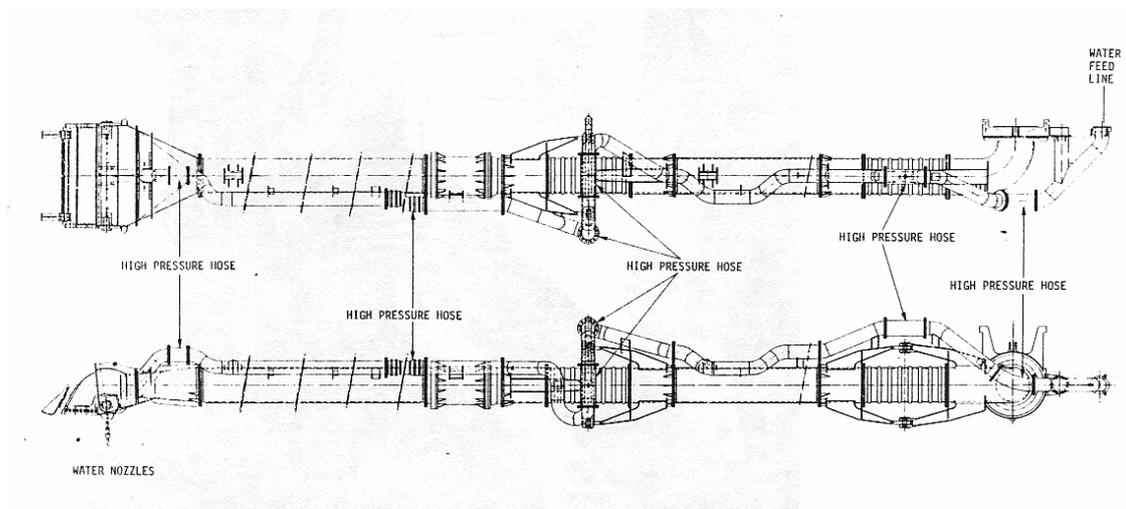


Figure 2-77 Suction pipe with a jet water pipe



Figure 2-78 Jet pipeline passing the universal joint

Because this pipeline also needs to follow all suction pipe motions, a lot of pressure hoses and elbows are needed, causing additional pressure losses in the jet-pipeline. The connection of the suction pipe with the ship becomes now more complicated.

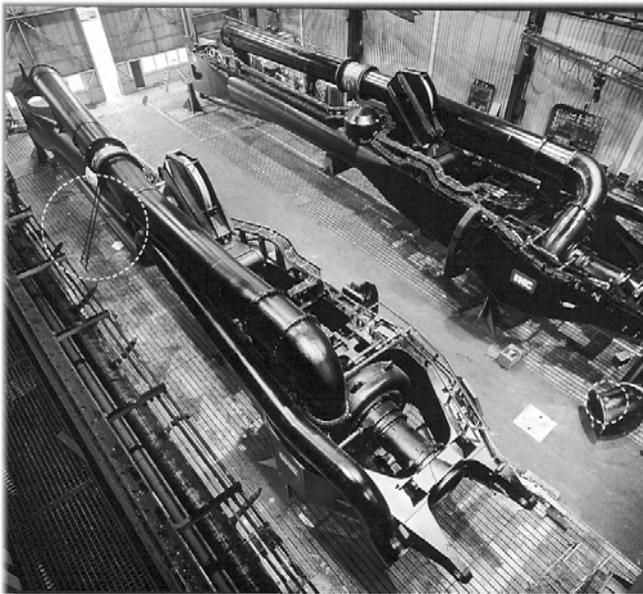


Figure 2-79 Submerged sand pump frame

It becomes even more complicated when a submerged pump is mounted together with the suction pipe (Figure 2.79). Except the pipelines, a lot of cables for power supply and to control the pump speed are necessary. For the powerful pumps a special frame is necessary to carry the loads.

2.3.1.3 The suction pipe gantries

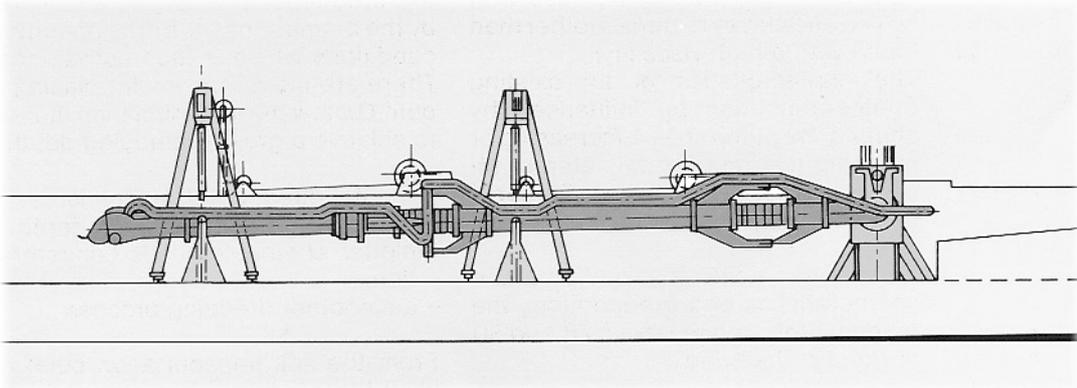


Figure 2-80 Suction pipe gantries

The three **suction pipe gantries** serve to move the suction pipe either inboard or outboard.

The **draghead gantry** and the **middle gantry** are carried out mostly as an A-frame, connected with the main deck by a hinge-construction (Figure 2.81 and 2.82). A hydraulic cylinder or the hoisting wires controls the motion when moving the suction pipe in- or outboard.

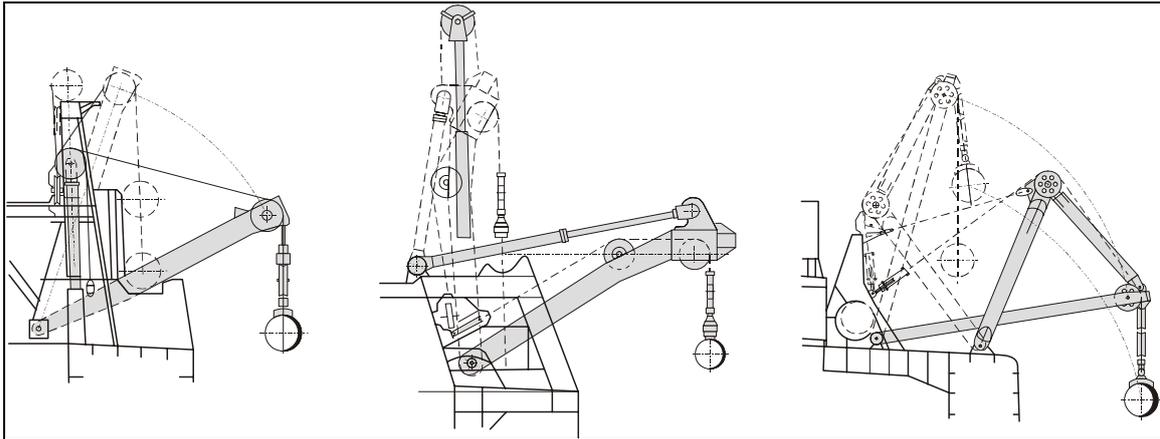


Figure 2-81 3 different types of suction pipe gantries

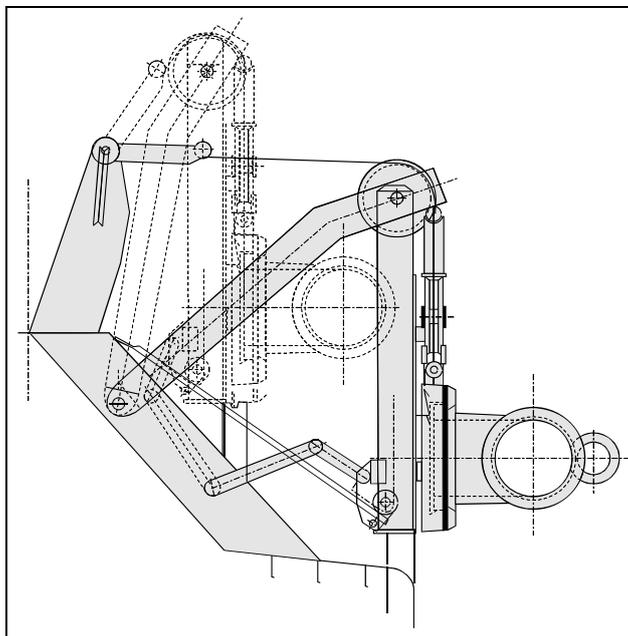


Figure 2-82 Suction pipe elbow gantry

The **suction elbow gantry** consists of a fixed and a moveable part. The fixed part is welded to the main deck and is fitted with tracks for the wheels of the moveable part. (Figure 2.82). When the moveable part has reached the lowest [position than the trunnion slide can be lowered into the guides in the hull

2.3.1.4 The swell compensator

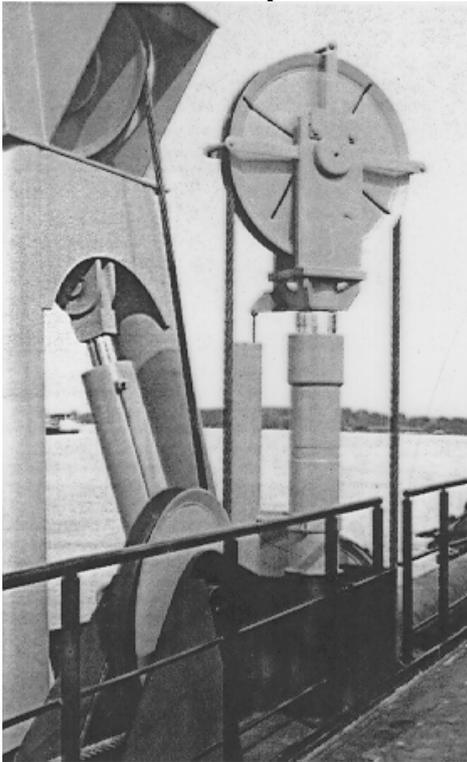


Figure 2-83 Swell compensator

The **swell compensator** has contributed to the success of the trailing suction hopper dredger too. The most important goal of the swell compensator is to maintain the contact between the seabed and the ship, due to either both ship motions or the irregularities of the bottom contour. The swell compensator is positioned in the hoist-cable system of the draghead winch gantry. The swell compensator prevents the uncontrolled slackening and re-tensioning of the hoist cables. (Figure 2.83):

Furthermore it maintains almost a constant pressure of the draghead on the seabed. A swell compensator system consists of the following components:

An hydraulic cylinder, of which the head is fitted with one or two pulleys that guide the hoist cable of the draghead.

One or more pressure vessels, of which the lower part is filled with oil and the upper part with air.

A oil pump and reservoir.

An air compressor.

A pipeline system that connects the hydraulic or pneumatic components.

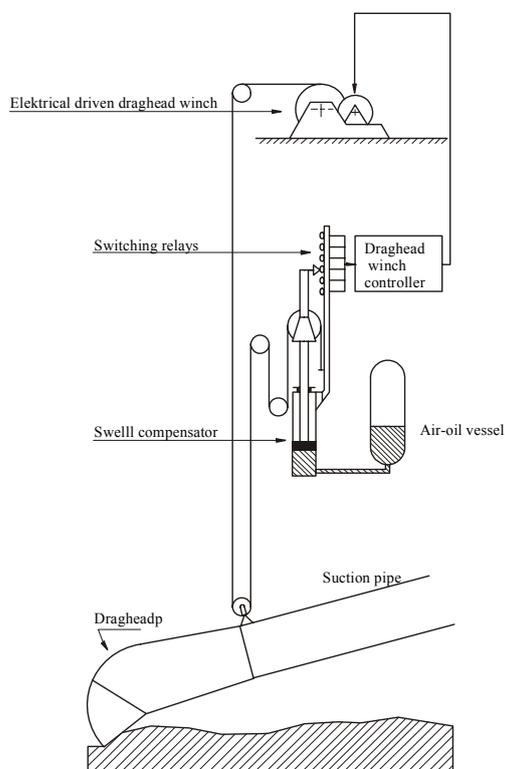


Figure 2-84 Swell compensator with draghead winch controller

During an ascending motion of the ship the piston rod of the compensator is pushed downward as a result of the increasing force in the cable. The plunger then compresses the air in the pressure vessel. During the following descending motion of the ship the piston is pushed out again as a result of the increased pressure in the pressure vessels. This assures a tight cable at all times.

The average pressure in the pressure vessels is determined by the weight with which the draghead may rest on the bottom, or better: how much the swell compensator has to compensate this weight. It will be clear that the compensation in silt will be higher than in sand. In table 1 values are given as a guideline by IHC for a certain configuration.

Table 1.

Suction depth			Compensation			
			Mud		Sand	
			80%	50%	50%	20%
25 m	Draghead weight on bottom	kg	1800	4500	4500	7200
	Fill air pressure	bar	15.0	15.0	8.0	8.0
	P in	bar	26.2	17.9	18.6	9.8
	P midstroke	bar	24.7	17.1	17.1	9.4
	P out	bar	23.3	16.4	15.8	9.0
17.5 m	Draghead weight on bottom	kg	2080	5200	5200	8320
	Fill air pressure	bar	15.0	15.0	8.0	8.0
	P in	bar	30.0	20.1	21.0	10.8
	P midstroke	bar	27.9	19.1	19.1	10.3
	P out	bar	26.1	18.2	17.5	9.8
10 m	Draghead weight on bottom	kg	2190	5475	5475	8760
	Fill air pressure	bar	15.0	15.0	8.0	8.0
	P in	bar	31.4	20.8	21.8	11.2
	P midstroke	bar	29.1	19.8	19.8	10.6
	P out	bar	27.1	18.9	18.1	10.1

2.3.1.5 The suction pipe winches

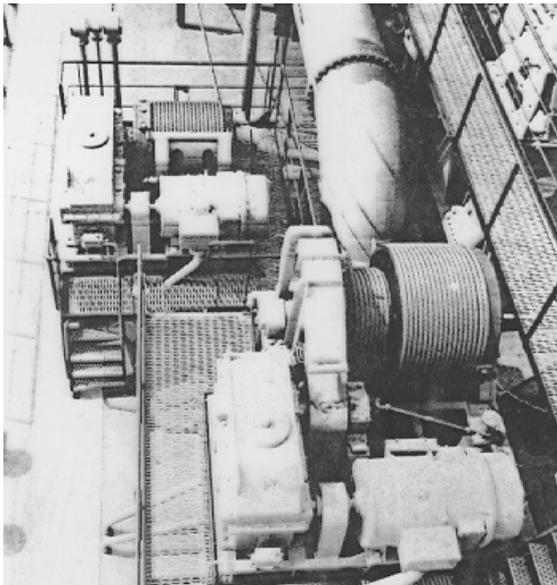


Figure 2-85 Suction pipe winch

Suction pipe winches have a grooved winding drum, with a length and /or diameter such that there are 5 windings left on the drum (Figure 2.85) when the suction pipe is in its lowest position. When the suction pipe is out of the water. The load of the winches becomes heavier. To overcome this problem the wire is transport to a drum with a smaller diameter, which results in a lower torque for the winch drive.

The winch drives is either electrical or hydraulically.

2.3.1.6 The dredge pump

The dredge pump is the heart of the trailing suction hopper dredger.

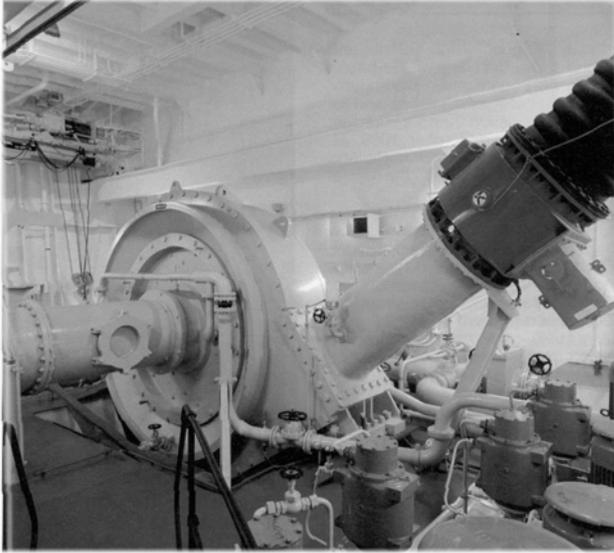


Figure 2-86 Dredge pump

The position of the **dredge or sand pump** has to meet certain requirements, certainly for the case without a suction pipe pump:

1. The inboard placed dredge pump must be installed as low as possible. The deeper the pump is under the water level, the higher the concentration of the mixture can be.
2. The resistance of the pipeline must as low as possible. So short suction pipes, wide bends and no constrictions.
3. The direction of rotation of the pump has to comply with the rotation direction of the mixture caused by the bends in the piping system.

The second requirement cannot always be met because of demands for maintenance or the accessibility for inspection or removal of debris.

There are also some practical objections concerning the third requirement. To comply with it the direction of rotation of the starboard and port pumps has to be opposite. This means more different spare parts like pump casings, impellers etc.

Speed control of the dredge pumps is highly dependent on the type of drive. If the main engine directly drives the sand pump then speed regulation is not possible or only by stepwise control using a gearbox. Is the dredge pump driven by a separate diesel engine then speed control is possible, but the best control is obtained by an electric drive. It has to be mentioned that currently new developments in variable transmissions come available for diesel engine driven pumps.

In most cases the requirements regarding the cavitation properties of the dredge pump are more important than the pressure properties. After all, even if the trailing suction hopper dredger has a pump ashore system, operations in dredging mode are considerably more frequent than the pump ashore mode.

Both **single walled** and **double walled** pumps (Figures 2.87 and 2.88) are used in trailing suction hopper dredgers, dependent on the view and strategy of the dredging company. Double walled pumps have a separate inner pump casing that can be worn out without necessary repairs. This is achieved by pressure compensation. The pressure in a running pump is equal inside and outside the inner pump casing. To do this the space between inner and outer pump casing is filled with water and pressurized. Besides the advantage of a longer lifetime for the inner pump, this type of pumps gives a higher security in case of explosions.

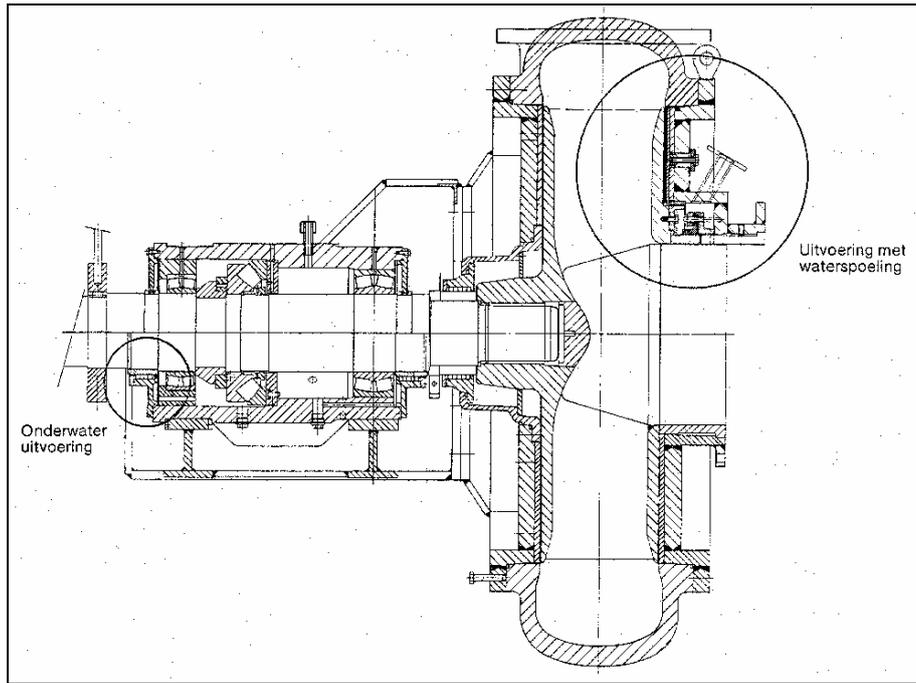


Figure 2-87 Single wall dredge pump

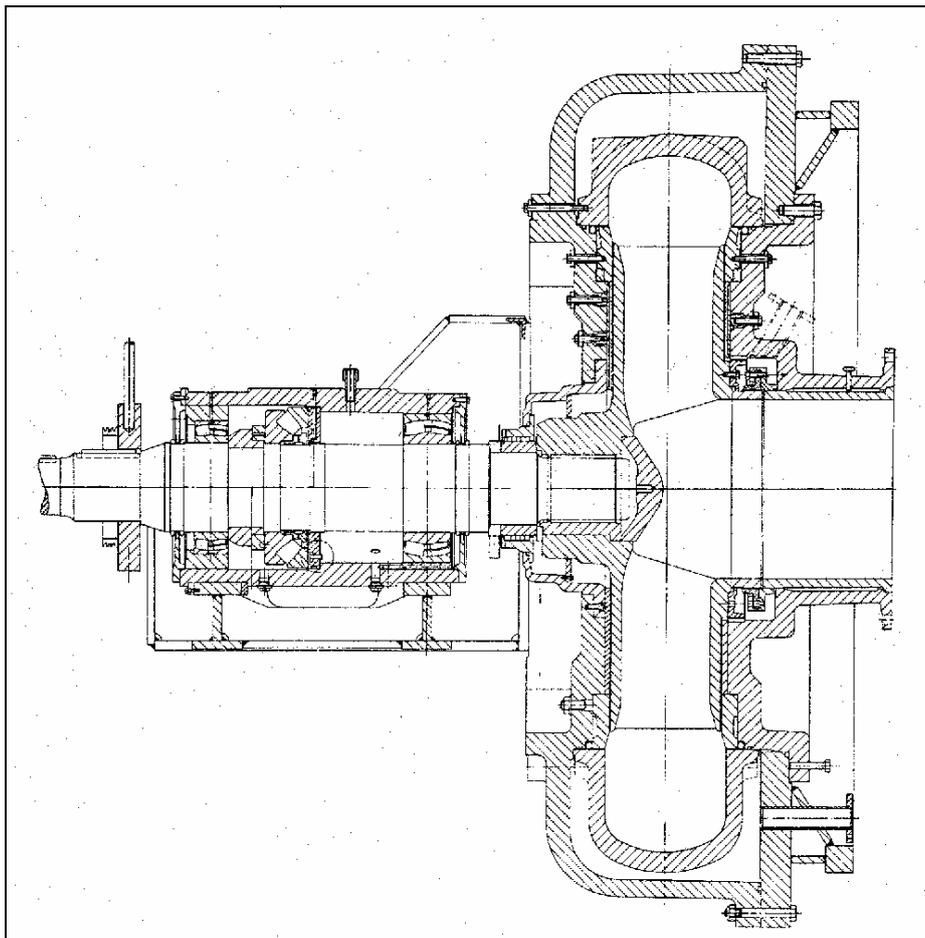


Figure 2-88 Double wall dredge pump

2.3.1.7 The jet-water pump

The **water- or jet pumps** are usually also positioned in the pump room. If these pumps are implemented with "clean-water pumps" than attention has to be paid to the position of the water inlet. After all contaminated water causes extra wear. Because the water surrounding the trailing suction hopper dredger is usually very muddy due to the overflow water, nowadays dredge pumps or weir resistant water pumps are used jet pumps.

2.3.1.8 The discharge pipeline

The discharge pipeline connects the dredge pump and the hopper loading system, or the dredge pump and the shore pipeline. Every trailing suction hopper dredger has the possibility to discharge the dredge mixture directly. Previously this was done above the waterlevel, but with increasing environmental protection demands, the so-called **poor mixture installation** (Figure 2.89) is connected with an always submerged pipe-end.

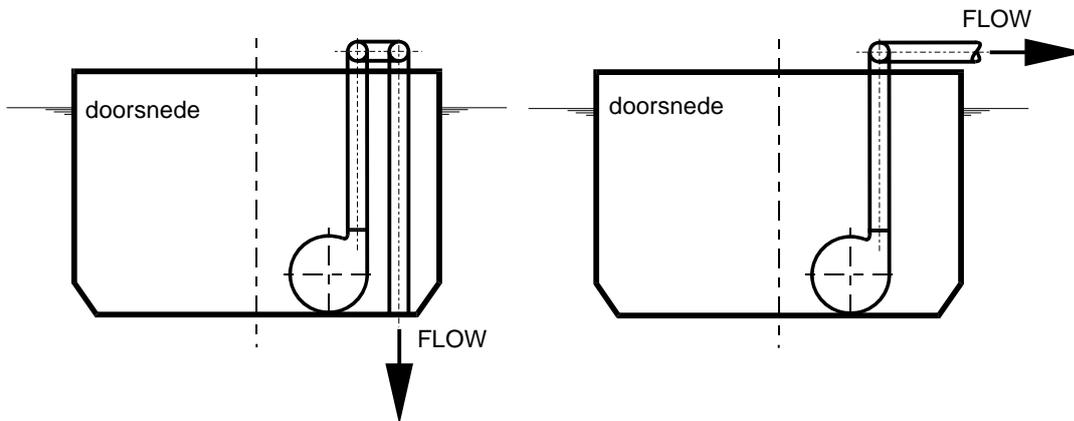


Figure 2-89 Poor mixture overboard systems



Trailing suction hopper dredgers with one suction pipe do have one delivery pipe constructed over the middle of the hopper and connected the discharge side of the dredge pump. Trailing suction hopper dredgers with two suction pipes can also have one central delivery pipe (Figure 2.90) on which the discharges of both dredgepumps are connected, or two separate delivery pipes (Figure 2.91).

Figure 2-90 TSHD with one delivery line



In this last configuration it must be possible to use both delivery pipes with both pumps. When one of the suction pipes cuts of, whatever the cause may be, the ship still must be loaded equally athwart-ships to prevent listing. This requires more valves than for one central loading gully.

Figure 2-91 TSHD with 2 delivery pipes

A similar complexity of the piping system arises also when shore pumping must be possible over starboard, port and over the bow. In a shore pumping installation the pressure pipe usually ends in a ball on which the shore piping can be connected. The bends are usually from cast steel for maintenance reasons.

On every trailing suction hopper dredger it must be possible, whether or not it is equipped with a shore pump installation, to suck the water from the well. If poor settling material is sucked than it is strongly recommended to discard the water that is left in the well when dumping, before suction to prevent dilution of the sucked up mixture.

2.3.2 The hopper

2.3.2.1 The loading system

The goal of the loading system is to dump the sucked up sand-water mixture as quiet and even as possible in the well. Three systems can be distinguished:

- the diffuser system (Figure 2.92).
- the central loading system (Figure 2.93).
- the deep loading system (Figure 2.94).

All with several variants on which many have explored their creativity.

In the diffuser system an open diffuser is positioned at the end of the delivery pipe, which discharges just under the highest overflow level. With such a system a good width distribution can be achieved. A disadvantage of the open diffuser is the reasonable amount of air that is taken in, which can obstruct the settling. Therefore closed diffusers are used sometimes that always discharge under the overflow level. The system is maintenance friendly of the system, compare to deep loading systems

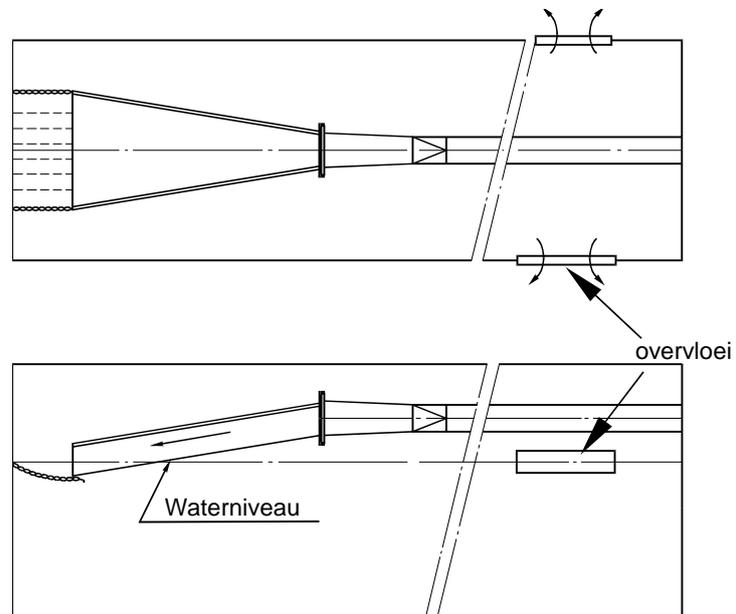


Figure 2-92 Diffuser system

Via closed diffuser the mixture is dumped through a distribution box in the middle of the hopper. The mixture flows to both sides of the hopper, where adjustable overflows are fitted. Theoretically the hopper load remains equal, if the flow remains 2D. The turbulence degree will decrease due to the distribution of the flow rate to two sides. An additional advantage of this system is that due to the overflows on both sides of the hopper the ship can achieve even keel more easily.

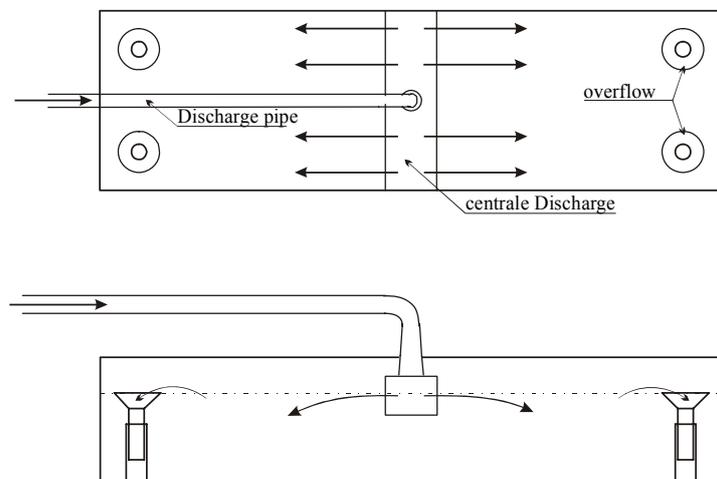


Figure 2-93 Central loading system

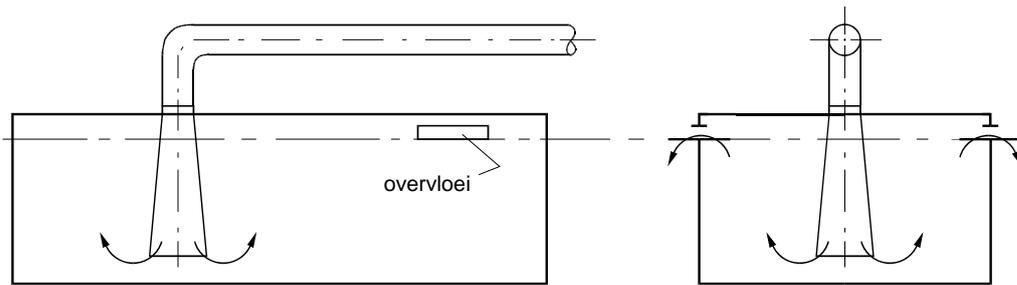


Figure 2-94 deep loading system

In a deep loading system the mixture is discharged deep in the well, whether or not with a vertical diffuser. The advantage of such a system is the energy reduction that is achieved as a result of the contact of the mixture with the already settled material. Another advantage mentioned the energy profit as a result of the siphon effect. In principle this is true, but there are quite a number of trailing suction hopper dredgers with a deep loading system for which it doesn't count because the delivery pipe is not airtight. Fitting of a simple kind of heavy loading or distribution valves in the delivery line causes this. These valves are necessary dredging coarse sand coarse or gravel. Then the settlement is that good that when these valves are not fitted the material settles immediately at the inlet and it becomes impossible to fill the hopper evenly (Figure 2.95). This results in a uneven trim vessel with water on their load

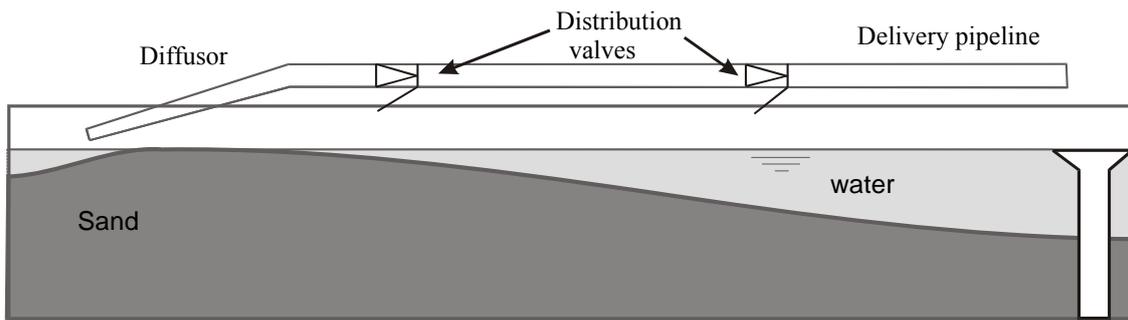


Figure 2-95 Distribution valves in the delivery pipeline, necessary for coarse material

Apart from that the take-in of air largely reduces the advantage of the deep loading system. Another disadvantage is that it is very hard to discharge the mixture evenly distributed over the width of the hopper. This causes jets with turbulence production with as a result possible disturbance of the already settled material.

A combination of the diffuser system and the deep load system is the diffuser box, which is placed half way the hopper height

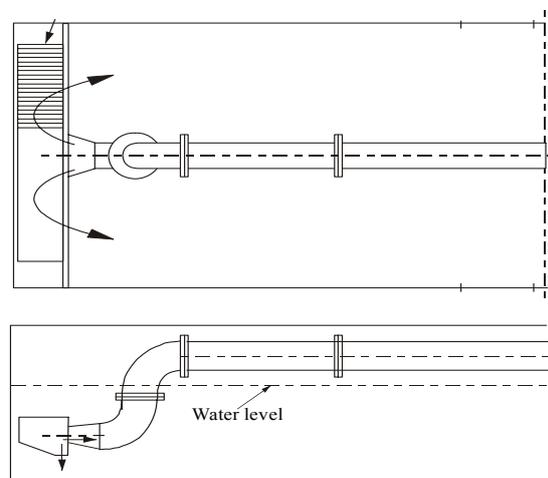


Figure 2-96 Box diffuser system

2.3.2.2 The well shape

The well shape has to comply with the following requirements:

- the least as possible obstructions in the well to keep the turbulence degree as low as possible in connection with the settling.
- as straight as possible side walls, preferably angling inward to improve the discharge of the load.
- easy accessible for maintenance.
- sand level above outside water level at least when the ship is in maximal draught, but preferable also at restricted draught (50-60% of maximum pay load).

The goal of the well or hopper is that the dredged material settles while the surplus water leaves the hopper through the overflow.

These **overflow losses** are largely dependent on the parameter $Q/(L*B)/w$ and less on $Q/(B*H)/w$. The first parameter is the ratio between the time a particle needs to settle and the time it stays in the hopper. The second parameter is the ratio between the horizontal velocity in the well and settle velocity of the particle and is a measure for the turbulence degree in the hopper. For a good settling a long narrow and shallow hopper shape is therefore favorable.

A danger is however that no equal distributed load over the length of the hopper can be obtained which results in a need for distribution valves in the delivery pipe. These valves decrease the settle length the final result can become worse. Besides, long small ships with a limited depth results in small engine room(s). A compromise between price and performance has to be found.

In the years past the obstructions in the hopper became less and less, as can be seen in the following cross sections (Figure 2.97):

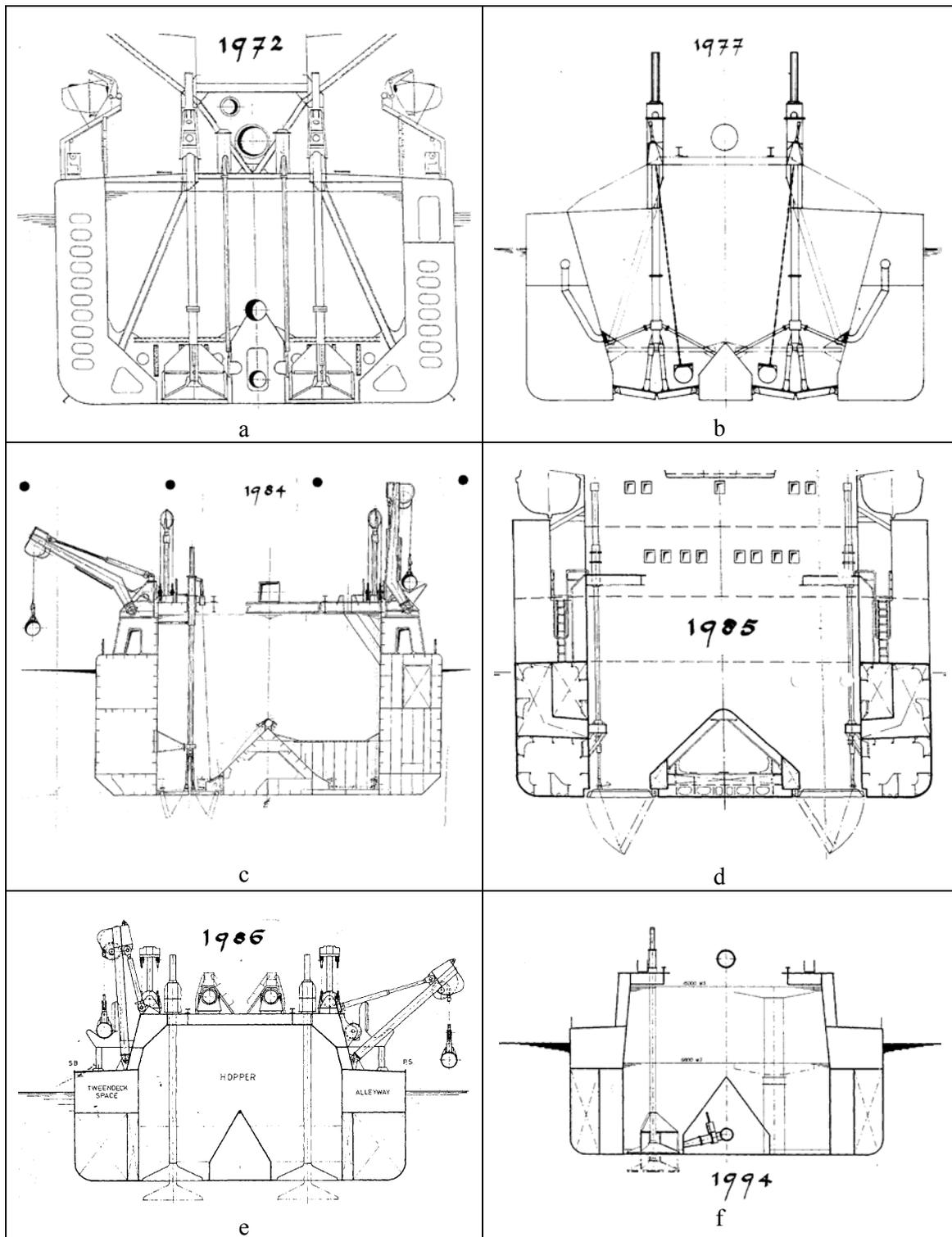
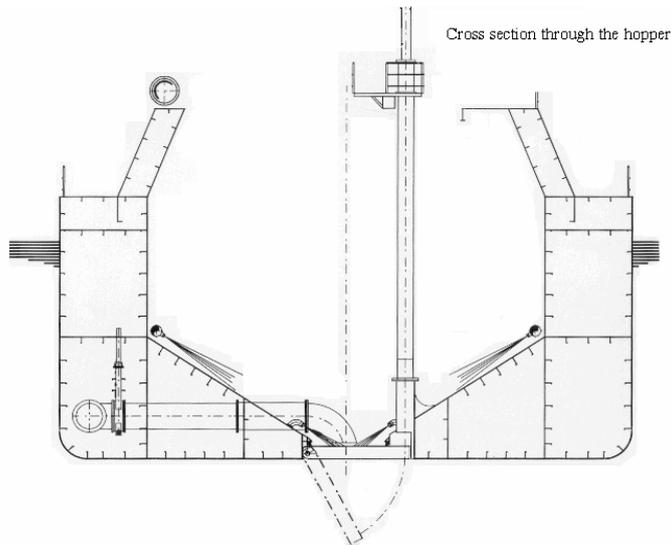


Figure 2-97 Different hopper cross sections



The last years hoppers with a V-shape become more and more popular

Figure 2-98 V-shape cross section

A well-shaped hopper (Figure 2.99) without any obstacle is formed by the **split hopper suction dredger**. There are no bars or obstacles, because the ship has no doors or valves but splits in two parts. The largest split hopper suction dredger built, has a deadweight of 7000 ton.

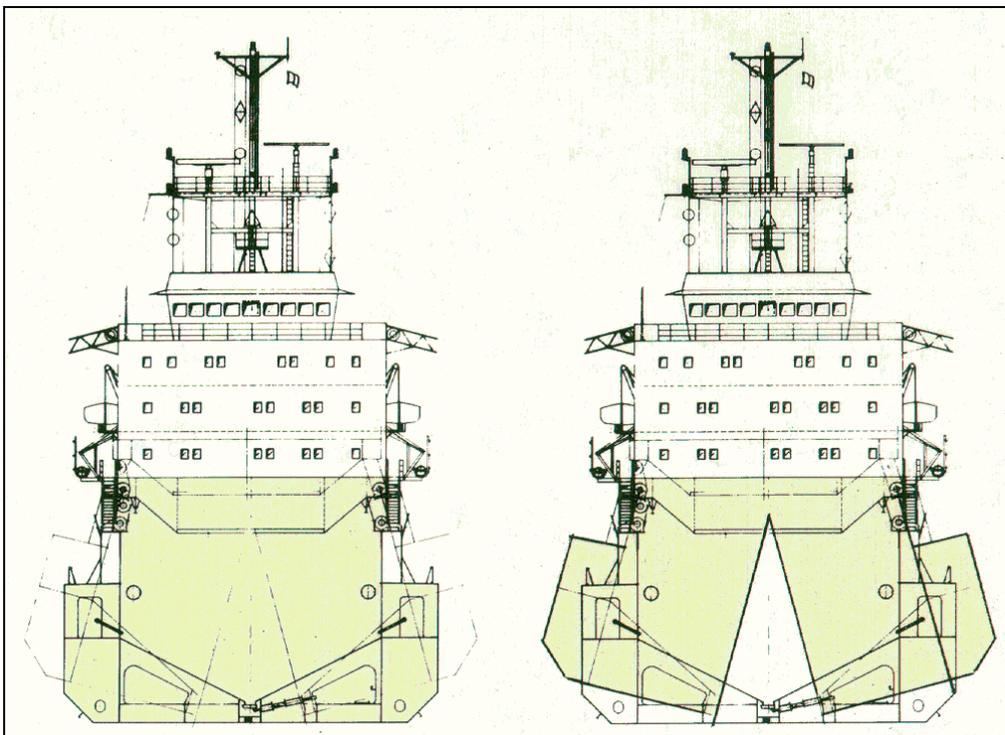


Figure 2-99 Split TSHD

The installation of pump ashore systems, as well as the requirement for easy maintenance have caused that, in general, closed hoppers hardly build, although they have certain advantages. (Figure 2.97e)

- In heavy seas rolling and pitching of the ship with a open hopper causes water movements and splashing over the deck of the mixture. A ship having with a closed hopper and a small overpressure, the water displacements during the rolling and pitching will be much less, which improves settling.
- The free space on the deck of a closed hopper is also seen as an advantage. Especially during mobilization, the trip from one job to another, when all kinds of equipment can be stored on the deck. During dredging these have to be removed to increase the deadweight of the ship.

2.3.2.3 The overflow type

At present almost all trailing suction hopper dredgers are built with a continuous adjustable overflow (Figure 2.101 & 2.102). Besides that most trailing suction hopper dredgers are of the so-called **Constant Tonnage System**, which requires a continuous adjustable overflow system.

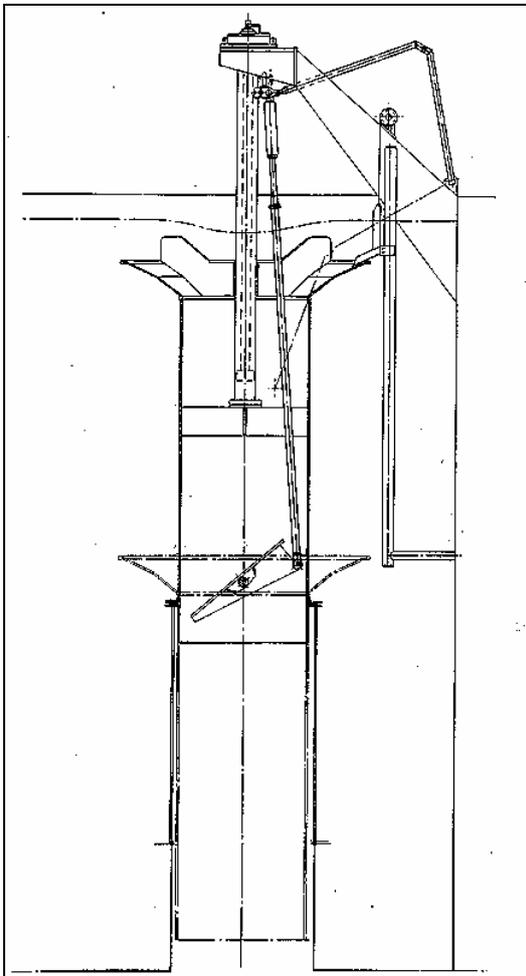


Figure 2-100 Overflow with environmental valve



Figure 2-101 Adjustable overflow over the full width of the hopper



Figure 2-102 Standard adjustable overflow.

There are however differences in the shape and place of the overflow to in order to increase the effective settling length (Figure 2.103 and 2.)

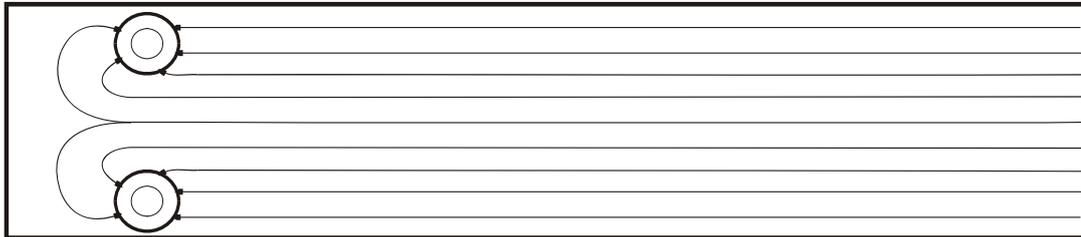


Figure 2-103 Flow of two round overflows on the side

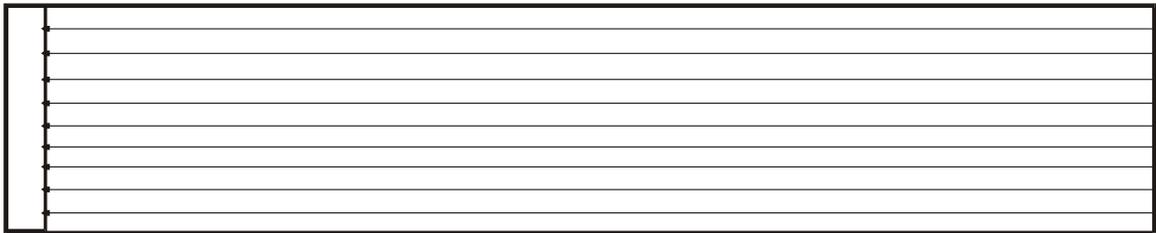


Figure 2-104 Flow of straight overflow at the end

A requirement that gets increasing attention is the environmental friendly overflow. Environmentalists do not liked a “beautiful” silt-jet behind the dredger. Dredging is often associated with polluted silt, so everything visible behind the dredger is “polluted”. A method to reduce the visibility of the overflow losses is to prevent the intake of air by the flow. This means that the overflow has to work as a non-free fall spillway instead of a free fall spillway. This can be done by building a so-called environmental valve (Figure 2.100) in the overflow. However, it is of course much better to design the overflow such that it works as an imperfect weir. This leads to a higher head (the height of the fluid surface above the upper side of the overflow).

2.3.2.4 The discharge system

As said earlier, discharging the load can be done in two completely different ways, either by dumping or by pumping.

2.3.2.4.1 Dumping systems

The goal of the dumping system is to discharge as quick as possible the material dredged with great effort.

All kind of systems are available. Expensive **conical valves** (Figure 2.105a), simple **bottom doors** (Figure 2.105b), horizontal **sliding doors or valves** (Figure 2.105c) or a ship that splits totally in two halves (fig 2.105d). There are also several exotic systems (fig 2.105f to 2.105h) all with their specific advantages and disadvantages. The **lijster valve** (Figure 2.105f) is very

expensive and takes a significant loss of hopper space. **Recesses valves** (Figure 2.105g) influence the stability unfavorably and necessitate a larger hull.

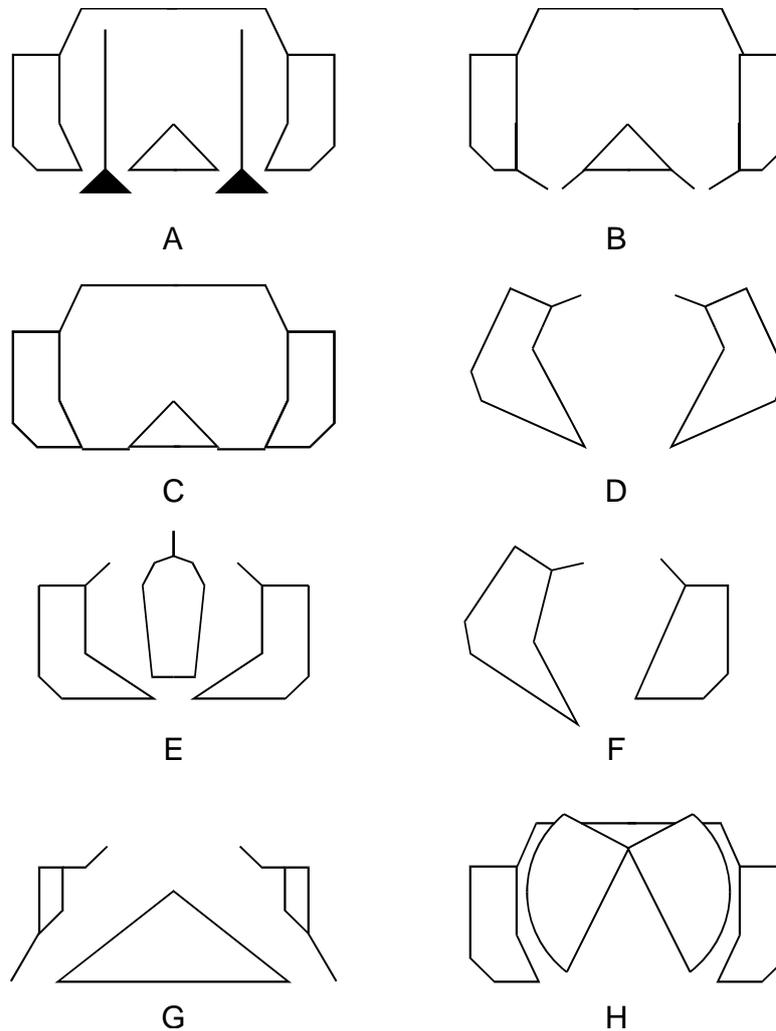
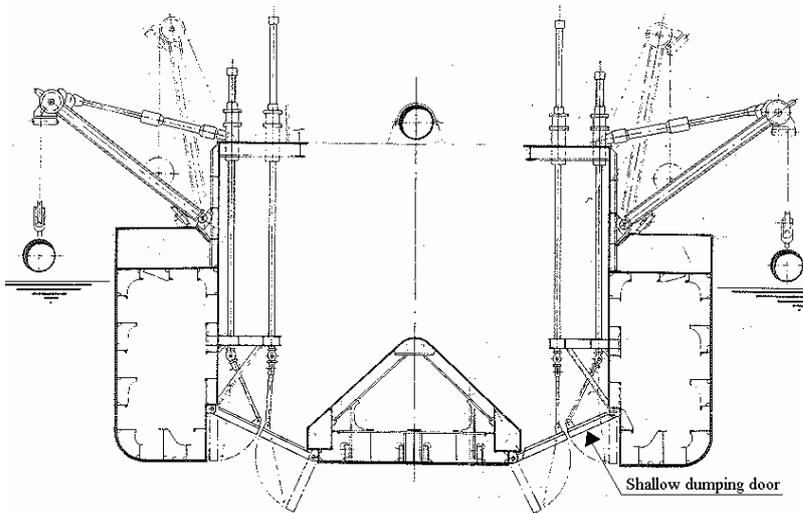


Figure 2-105 Different discharge systems

Requirements for his dumping systems

- First of all the ship has to be able to discharge the load in a short time, as completely as possible (so without any load left) and for all types of soil. This means that the discharge area has to be large enough. Dependent on the dredged material the discharge-area ratio (the ratio total discharge-area/ horizontal hopper area) increases from 10% for slurries to 50% for the cohesive soil types. For general useable ships this will be about 30% of the hopper area. As already mentioned in chapter 2.2.3.10 the discharge is better as the out-flow behaves like a plane symmetrical flow. The length/width ratio of the discharge opening has to apply to $L \geq 3B$.
- Furthermore as few as possible protruding parts are allowed in the hopper, they can cause bridging of the material. Additionally they have the disadvantage of forming an obstruction for the settling too.
- An proper sealing under all circumstances. This demand increases in importance when (polluted) silt is dredged.

- Little or no influence on the ships resistance.
- Maintenance friendly. Places where wear can occur have to be easily accessible.
- Possibility for discharging the load in shallow waters and grounded ship.



Regarding the first requirement the doors have the advantage over the others and for the last four demands the conical valve or the split-hopper. Dumping in shallow water can also be achieved with so-called shallow dumping doors. (Figure 2.106).

Figure 2-106 Shallow dumping doors

The operation of the dumping system is mainly done by a hydraulic system. For the doors and the valves the cylinders are positioned vertical. The doors or valves in this system can be operated in groups, usually three. In every group the hydraulic system controls both the starboard and the corresponding port cylinder.

For the horizontal sliding bottom valves two cylinders positioned in the longitudinal direction of the ship activate those. Both cylinders move simultaneously, so all doors are open at the same moment.

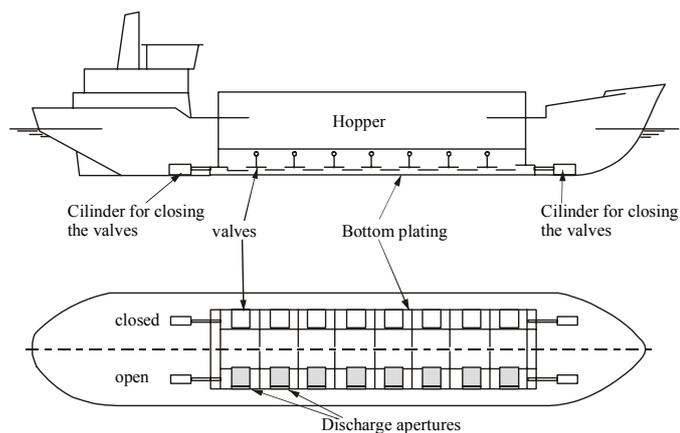


Figure 2-107 Sliding bottom valves

The split hopper dredger has a hopper without obstacles and in opened position one large discharge opening (plane symmetrical flow) and therefore a high discharge velocity, especially

useful to dump submerged dams. The split hopper dredger can under grounded conditions discharge well. The frequently mentioned advantage of well discharging cohesive soils is disappointing in practice. Usually the bottom plates in the hopper, even in opened position are insufficiently steep to be assured of a good discharge (Figure 2.108).



Figure 2-108 Split hopper dredger

For a split hopper dredger dumping is done by the splitting of the ship in the longitudinal direction. The two halves are connected with hydraulic cylinders and hinges. Of course the deckhouse and the accommodation remain upright during the splitting, because it is connected with the deck by hinges and hinge rods.

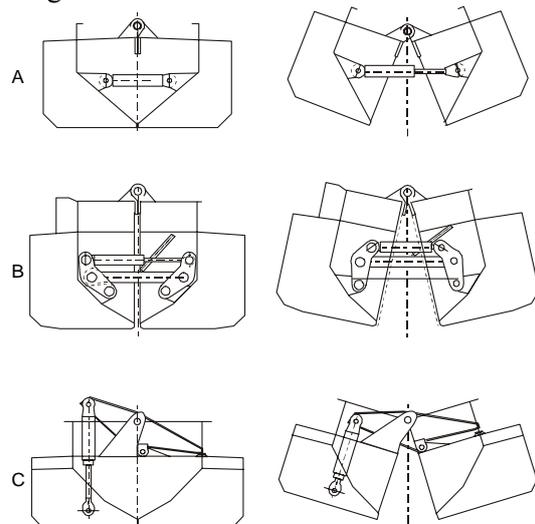


Figure 2-109 Different mechanism

2.3.2.4.2 The pump ashore system

Except for direct discharge or dumping, it can be desirable for certain works to pump straight to shore, not only for technical reasons but sometimes also for financial reasons. In principal direct discharge and re-handling with a cutter suction dredger is cheaper, however several important financial conditions have to be met:

- The work must have a sufficient size to earn back the mobilization costs of an extra cutter suction dredger.
- This also counts for the re-handling pit, from which the cutter suction dredger pumps the dumped sand to the reclamation area. This can be positive if such a dump can be situated within the work.

If the work is done with more trailing suction hopper dredgers it is in many cases beneficial to discharge directly and re-handle the sand. Because, even having two identical trailing suction hopper dredgers on the job, the stochastic behavior of the dredging process causes that at a certain time that the two ships arrive at the same time at the connection point for pumping ashore, causing waiting for one of the dredgers.

But there are also works where the direct pumping ashore or so-called rainbowing has large advantages. For example works at sea like beach nourishments. For that goal small trailing suction hopper dredgers are equipped with pump ashore equipment. There are also jobs without space for a re-handling pit.



Figure 2-110 Rainbowing

Besides, there are jobs requiring controlled dumping of their load at a certain depth and in a relative small area. Then the material is pumped back through the suction pipe. This has been the case at the Oosterschelde works and is done too when covering pipelines.

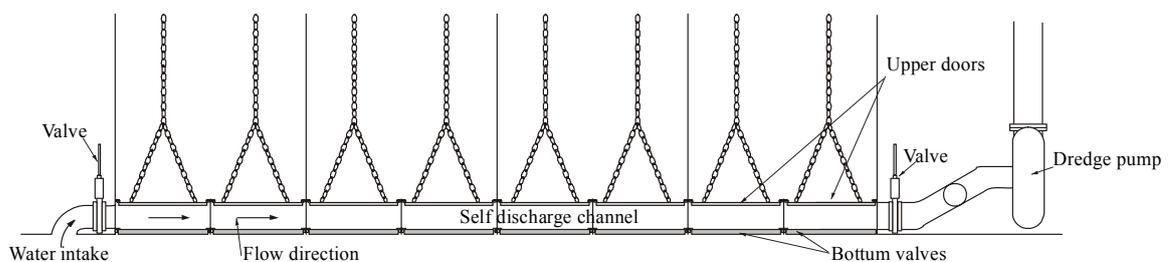


Figure 2-111 Longitude cross section pump ashore system

The decision to equip a ship with a pump ashore system is not taken just before the work needs it. Except for the fact that the preparation and the fitting time can be more than half a year, it is also much more expensive than when it is fitted directly during the construction of the ship. Ships initially not fitted with a pump ashore system don't have mostly today. Nowadays the European dredging contractor usually chooses for a pump ashore system.

A pump ashore discharge system consists of one or two suction channels, situated at both sides of the center-keelson (Figure 2.112 under) or a pipe centrally placed in the center-keelson (Figure 2.112 upper). In the first case the top of this suction or self-emptying channel is fitted with so-called top-doors, by which the sand can be supplied into the channel. Transport water is mostly supplied in two ways, first through the channel, which is connected in some way with

the outside water and second by the jetpumps that fluidize or erode the sand in the surrounding where the sand has to enter the channel.

The mixtures pumped ashore with a well-designed installation do have very high densities. For example 7500 m³/h in a 800 mm pipe. Of course this is also dependent on the type of sand.

The rest load, the load that cannot or hardly be removed, is a measure for the design of the shore pump discharge system. For the mono-hull ships it may not be more than 5% of the total load.

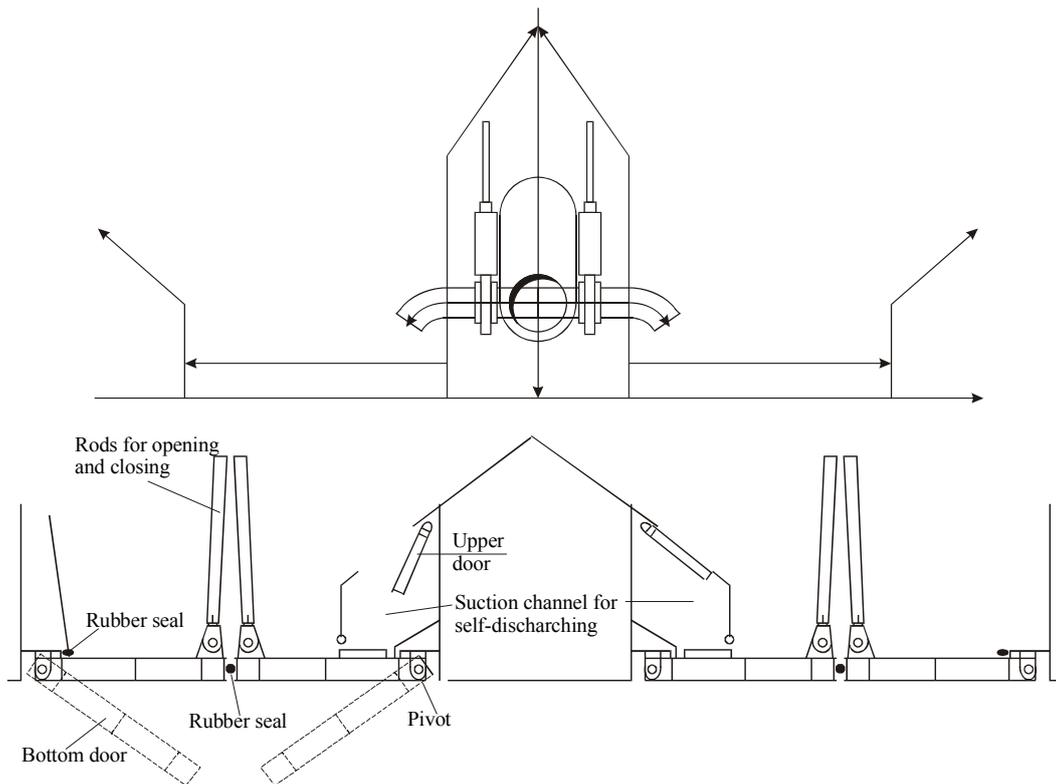


Figure 2-112 central discharge pipe line (above) and channels on both sides of the keelson (under)

In split hopper dredger the self discharge channel (Figure 2.113) is situated exactly in the middle, between the connection of the two halves.

For split hopper dredgers this rest load is zero, except for cohesive materials.

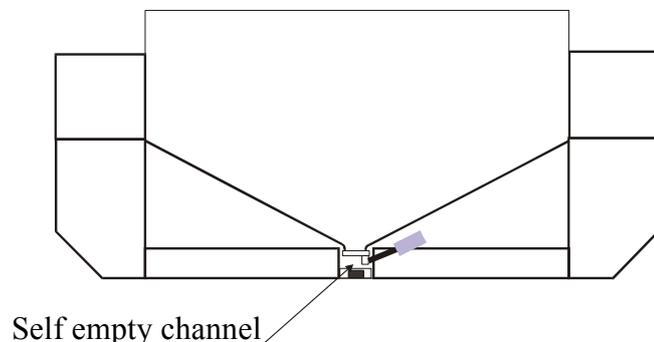


Figure 2-113

Except for trailing suction hopper dredgers having besides bottom doors or valves, a pump ashore discharge system, there are also trailing suction hopper dredgers without a bottom discharge system, but with a self-discharge system. This is usually seen on aggregates hopper dredgers. The self-discharging happens mechanically, either with a dredging wheel (Figure 2.114) or with a clamshell that grabs over the full width of the hopper.



Figure 2-114 Dredging wheel unloader (Left) and clamshell unloader (right)

2.3.3 The propulsion

Trailing suction hopper dredgers in general two controllable pitch propellers. (see also chapter 2.25) Only in the sea mining industry trailing suction hopper dredgers with only one screw can be found, whether or not controllable pitch. The advantage of controllable pitch propellers has to do with the method of operation of the ships. On one hand the ship needs enough propulsion power at relative slow speed of 2 to 3 knots to drag the suction pipes over the seabed. On the other side the sailing speed from and to the borrow area should be as high as possible, normally between 12 and 15 knots. To fulfill both requirements the propellers are placed within nozzles. Additionally the concept of double and adjustable screws strongly improves the maneuverability.

A trailing suction hopper dredger needs surely good maneuverability. For instance dredging along a quay wall with a ship with a length of 100m or more on a distance of less than 10m. When maintaining harbors trailer dredgers always moves in shipping lanes. This in contrast with merchant shipping stays in the harbor as short as possible. The maneuverability has strongly improved over the years. Not only by installing more powerful bow thrusters and in some cases even aft thrusters, but also by (special) rudders with large angles

2.3.4 The maneuverability

The trailing speed of trailing suction hopper dredger dredges is 2 to 3 knots (1 to 1.5 m/s). At this velocity the maneuverability needs to be high. After all the higher the maneuverability the less the over-dredging (outside the tolerances) and the less a chance on collisions there will be. Therefore most trailing suction hopper dredgers are equipped with double propellers and one or more bow thrusters. If Dynamic Positioning/Tracking (DP/DT) is stern thrusters are sometimes installed too. To maneuver the following options are available on a trailing suction hopper dredger:

- Just rudders
- Just the adjustable screws
- Just the bow screw and/or stern screw

- A combination of these

Which possibility will be used depends strongly on the direction in which the ship has to sail and the effectiveness of the various options under certain circumstances. The thrusters are only effective for very slow forward velocities. Above 2 to 3 knots the effect is mostly gone, the combination of propellers and the rudders are in that case a better option. However, the maneuverability is also strongly dependent on the center position of the rudders in relation to the propellers. On trailing suction hopper dredgers these are usually positioned more inboard in relation to the direction of the propeller shafts to be able to exchange the propellers without removing the rudders. Turning with one propeller forward (port) and one backward (starboard) with both rudders fully starboard is now less effective than the starboard propeller full ahead. After all in the first case the port propeller will hardly exert any force on the rudder.

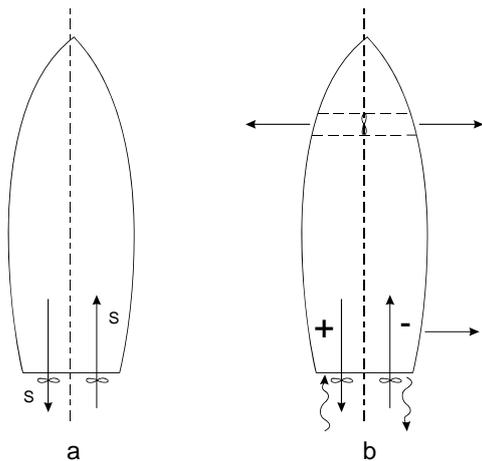


Figure 2-115 Opetration of adjustable screws

Is a transverse movement desired and the ship is equipped with both a bow and stern thruster than it is logical to use these. If there is no stern thruster available the transverse movement can be generated by rotating the adjustable screws opposite (Figure 2.115).

Also the effects of flow during dredging have to be compensated either by the bow thruster or delivering more power to one of the propellers than the other.

2.4 Strength and stability

2.4.1 Strength

Every sea-going vessel longer than 24 m, and therefore also a trailing suction hopper dredger must have a **load line** assigned according international agreed rules. The **free board** is the distance from the load line to the top of the main deck. The size of this free board is indicated on the vessel both on port and starboard by the **Plimsoll-line**¹ (Figure 2.116) (Samuel Plimston let the English Parliament approve an act in 1876 that had to prevent the overloading of ships).

This line indicates, except for the allowed loading level in several different waters, also the initials of the registering agency of the ship.

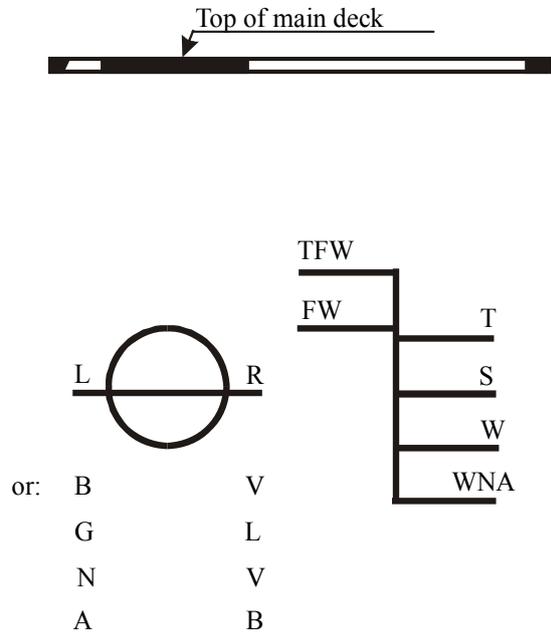


Figure 2-116 Plimsoll mark

Every seaship loaded to the **International Free Board Line**, has to comply with certain demands for strength. In principal there are two demands:

1. demands of strength concerning the loading of the ship until the allowed draught on flat water.
2. demands of strength concerning the wave forces on the ship

For this last condition a distinction is made of the working areas of the ship. The so-called **classification**:

1. Deep sea (haute mer). Is assigned to ships capable for transoceanic navigation.
2. Great coasting trade (grand cobotage). Assigned to ships deemed suitable to perform deep sea voyages but not transoceanic navigation.
3. Small coasting trade (petit cabotage). Assigned to ships that may not sail further from the coast than a distance from the coast that they can reach a safe harbor or mooring place within six hours.
4. sheltered waters (eaux arbitrées). This class is assigned to ships that are allowed to sail, usually under good circumstances, at most at a small distance from the coast (mostly less than 15 miles).

Above mentioned classification, of the **Bureau Veritas**, is international acknowledged, as well as those of other classification bureaus (Lloyd's Register, Germanische Lloyd, Norske Veritas, American Bureau of Shipping and others).

In the dredging industry there is a by local authorities allowed draught, known as the dredging mark. That is the allowed draught that is usually set in the middle between the international free

board and the top of the main deck of the ship. The ship must of course be able to carry the loads that can arise under such circumstances.

Trailing suction hopper dredgers that are loaded to the dredging mark are not allowed to make international trips.

Except for classifications there are also notations that are related to the rules for building specialty ships. Both the trailing suction hopper dredger as the stationary suction dredgers are assigned to those rules.

2.4.2 Stability

Except demands regarding the strength, a ship has to comply too with a minimum stability. For sea-going ships the international demands apply, dependent on the type of the ship. For trailing suction hopper dredger in principal the same rules apply as for sea-going cargo vessels.

Definition: **Stability is the ability of a floating construction (ship) to return to its original equilibrium position when it is disturbed from its equilibrium position by external effects.**

The stability of a ship is determined by a lot of factors, like the shape, the weight, the weight distribution and particular for a trailing suction hopper dredger all so-called free liquid surfaces in relation with the "wet surface". Wind, waves, movement of the cargo, movements of liquid cargo, sharp turns, etc can cause forces or moments that can bring the ship out of equilibrium.

When a ship tilts, the position of the mass center of gravity doesn't change as long as the cargo doesn't move. This is in contradiction with the center of buoyancy that shifts to the side to which the ship tilts (Figure 2.117).

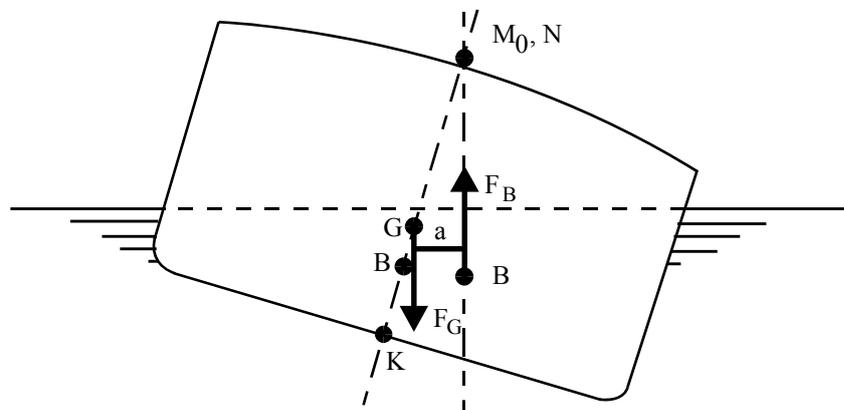


Figure 2-117 Recovering moment

The upward force remains, of course, the same but opposite to the weight, but their worklines are now shifted apart over a distance a . They form a moment that tries to bring the ship back in equilibrium. This moment is called the **static stability**. The work-line of the upward force cuts the symmetry plane in a point that is called the **meta-center M**. For small angles of heel ($<6^\circ$) this point can be considered as fixed (initial meta-center). The distance between the center of gravity and the meta-center is also called **meta-center height MG**. For larger angles of heel the meta-center is dependent on the angle of heel (false meta-center).

From the Figure 2.117 can be directly derived that :

- The arm of the static stability is equal to $MG \cdot \sin \varphi$.
- There is only an equilibrium recovering moment when the meta-center is above the center of gravity of the ship.

If the arm of the static stability is set out as a function of the angle of heel φ than a curve is obtained that looks globally like Figure 2.118.

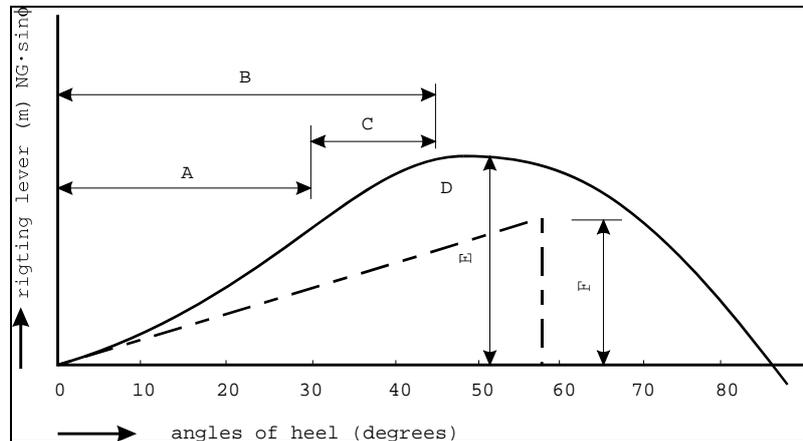


Figure 2-118

Every ship has to comply with the minimum stability curve (Figure 2.119).

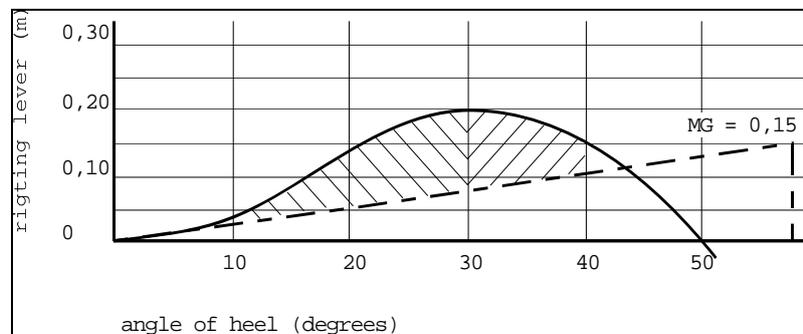


Figure 2-119

This is determined with the following requirements:

- The surface under the curve to a angle of heel of 30° has to be at least 0.055 radial.
- The surface under the curve to a angle of heel of x° has to be at least 0.09 radial.
- The surface under the curve between the angles of heel of 30° to x° has to be at least 0.03 radial.
- The arm of the static stability has to be at least 0.2 m.
- The initial meta-center height has to be at least 0.15 m.

In the above mentioned requirements x° is equal to 40° or a smaller angle that is indicated by openings in the hull or deckhouse that cannot be closed watertight. With the above mentioned

stability curves it has been assumed that the mass center of gravity does not shift but remains in the symmetry plane.

If a fuel or water tank is not completely filled, the fluid will try to maintain a horizontal level independent of the tilt of the ship. This so-called **free water surface** is the cause, however, of a shift of the mass center of gravity outside of the symmetry plane. As a result the arm of the raising couple becomes smaller. It is clear that the effects of a free liquid surface in all possible storage tanks have to be taken into account in a stability calculation.

The free liquid surface is not only important for the tanks of common ships, but particularly important for ships with a relative large free liquid surface like a trailing suction hopper dredger.

2.5 The dredging process

As already described in paragraph 2.1.4, the dredging process of a trailing suction hopper dredger consists of the cycle of dredging, sailing to the discharge area, discharging and sailing back to the dredging area. Every part of this cycle contributes more or less to the production. So the less malfunctions occur in the separate processes the higher is the cycle production. In the following chapter these cycle parts and the connected dredging processes are discussed.

2.5.1 The loading process

The loading process can be divided in **excavation, the transport and the deposit** of the material in the hopper.

2.5.1.1 The excavation

Though other working methods exist, in principal the trailing suction hopper dredger deepens a large area entirely. The different layers of soil are removed horizontally. This in contrast to the cutter suction dredger and surely the suction dredger, that first deepens locally and then slowly expand horizontally. This working method has consequences for the determination of the material to be removed. Usually the horizontal variation, for instance the grain size or the chance of soil type, is considerably less than the vertical variation. This also implies that the mixture of the several layers is considerably less, which gives less meaning to an average material in the dredging area.

The trailing suction hopper dredger can in principal be deployed in nearly all soil types. Only the efficiency is strongly dependent on the soil type and the power and means to break up the coherency of the soil type.

When excavating with dragheads the soil type is very important. In the excavating process the following materials can be distinguished:

- Liquid soil types (silt and soft clay).
- Cohesive soil types (firm clay, soft rock).
- Non-cohesive soil types (sand and gravel).

2.5.1.1.1 *Excavating of liquid soil types*

When dredging silt or soft clay the Attenbergs limits (plasticity-index and the liquid-index) are important. The first index determines if the soil type behaves clayey or sandy. For a plasticity-index < 7 the material behaves sandy. The second index determines if the material behaves like

a fluid and thus easy to dredge or firm and has to be cut. A soil type behaves like a fluid when the water content is close to the liquid limit.

For a fluid-like behavior the liquid-index must be like: $\frac{w - w_p}{I_p} > 0.9$

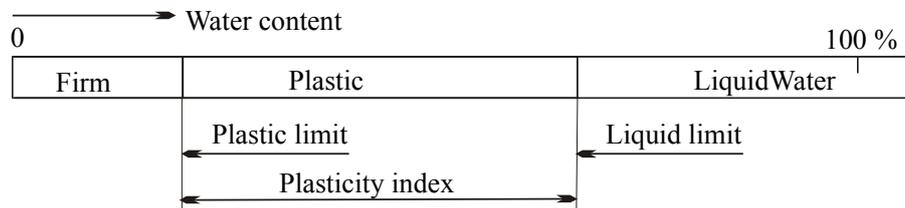


Figure 2-120

When dredging a liquid-like soil the volumetric concentration, $C_v = \frac{\rho_{mixture} - \rho_{water}}{\rho_{situ} - \rho_{water}}$, is almost independent of the in situ density. Also the dimensions and type of the draghead have hardly matters. This means that the fill rate also is almost constant. For virginal fluid silt this is around 70 to 75 %. Then the ship is loaded **"until overflow"**. The nett suction time is totally determined by the rheological behavior of the silt.

If there is a lot of contamination, like stones, wires, old bikes, etc. in the silt or if the length of the dredging area is small, requiring frequently turning, the fill rate will reasonably decrease. When debris clogs the draghead, the dredge-master will dilute the mixture. Besides that regular stops for removing the debris in the draghead as well the restarts of the process, dilutes the mixture too. Fill rates of 40 % or less are easily reached. When the silt gets a more consistent behavior, thus a lower liquid-index, the fill rate to the overflow decreases. But because the silt is more consistent it will behave less like a homogeneous fluid and more like a mixture of pieces silt/clay in a heavy transport fluid. The loading after the overflow is reached, with a lot of overflow losses, becomes interesting again; therefore the fill rate can still be reasonable. However the suction time will increase.

In silt, as a result of the decay processes of organic material, gas can exist in the form of bubbles. Besides it is possible too that this gas is dissolved in the pore-water. When dredging silt, the gas-bubbles will grow when moving upwards caused by the pressure drop in the suction-pipe. ($p \cdot V = \text{constant}$) Regarding physics this situation is almost equal to the forming of vapor bubbles in water during a pressure drop, however than it is called cavitation. Because cavitation decreases the performance of the dredge pump, this will also be the case with gas bubbles. The advantage with gas bubbles is that it happens in the pipeline system before the pump. This creates the possibility to take away a part of the gas bubbles before they implode in the pump. For this reasons a de-gassing installation is mounted in the pipeline just before the pump. A well-designed de-gassings installation does not or hardly decrease the performance of the pump. Two systems are used: a de-gassings installation with accumulator (Figure 2.121) or a de-gassings installation with a gas-extractor tank (Figure 2.122).

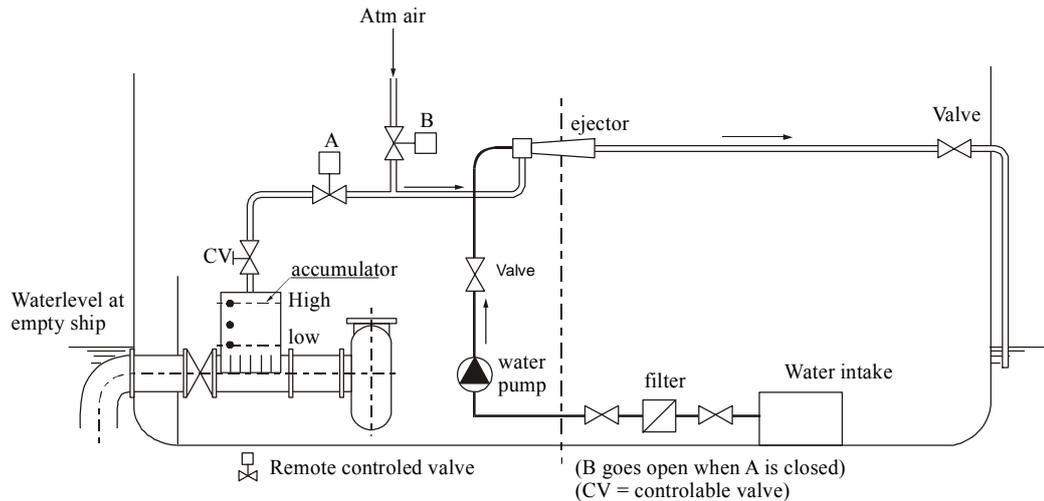


Figure 2-121 Degassing installation with accumulator

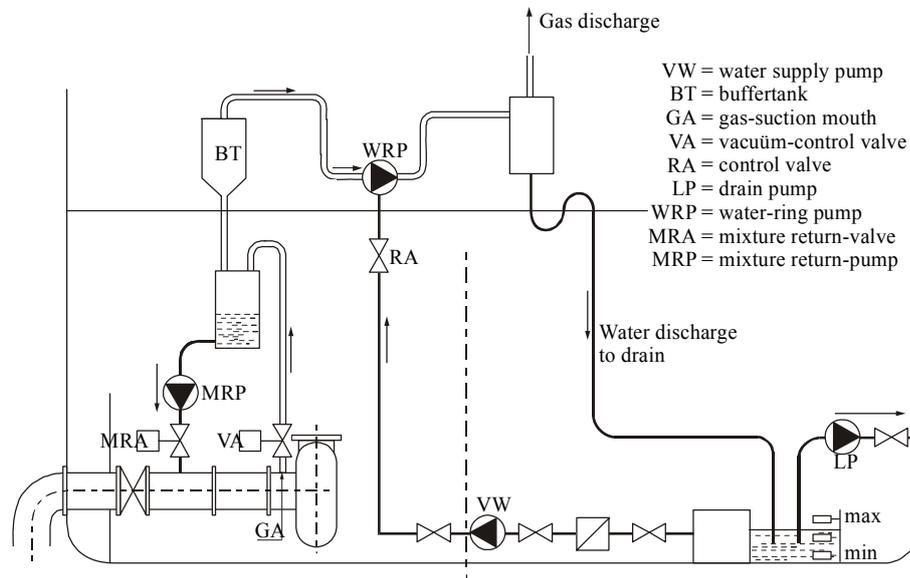


Figure 2-122 Degassing installation with gas-extractor tank

2.5.1.1.2 Excavating in cohesive soil types

In cohesive soil types, like **very soft rocks**, **clay** and to a less extend in **silt**, the cutting dominates the excavating process. In the dragheads blades, chisels or teeth are mounted (Figure 2.123). A well-shaped design is important to prevent clogging. Besides this improves the mixture forming too.

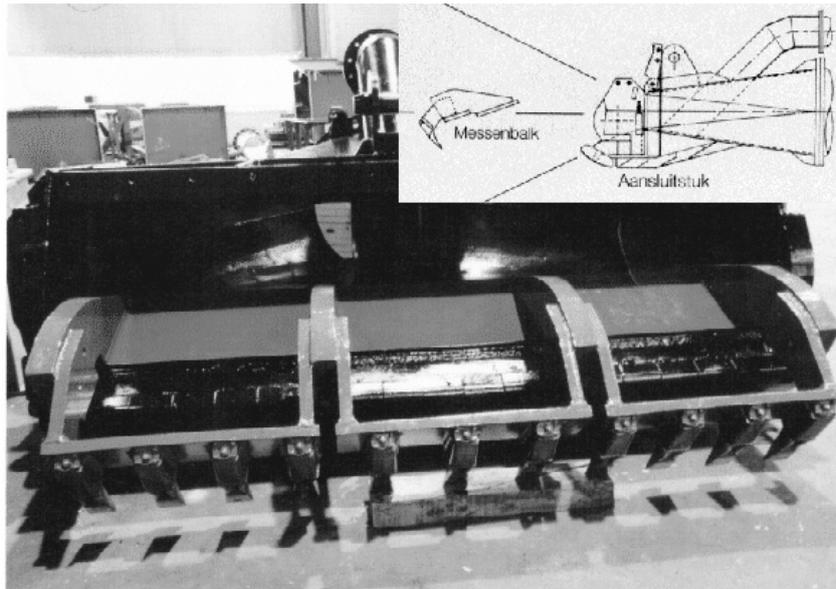


Figure 2-123 Modular draghead with a “teeth beam”

The linear cutting theories for rock cutting and undrained clay cutting apply here. In this case the cutting forces for the applied trail-velocities are only slightly speed-dependent. Besides the cutting forces increase linear with the depth. This means that the specific energy is almost constant for this cutting process. The pressure difference over the draghead plays not or hardly a role for the cutting forces. To keep the blades pushed into the soil the pressure difference over the draghead is usually insufficient and the visor has to be fixed to the helmet. The cutting depth is adjusted either by placing a stopper on the helmet related to the dredging depth or by hydraulic cylinders. As described in chapter 2.2.5.1 these cutting forces has to be provided by the propulsion.

For the calculation of the cutting forces for design purposes it is the custom to use the specific energy concept. The specific energy E_s is the energy needed to cut one m^3 . In formula:

$$E_s = \frac{N_s}{P_s}$$

E_s	=	Specific energy	[J]
N_s	=	cutting power	[W]
P_s	=	cutting production	[m^3/s]

For the force applies:

$$P_s = v \cdot d \cdot b$$

and for the power:

$$N_s = v \cdot F_s$$

$$F_s = \frac{E_s \cdot v \cdot d \cdot B}{v} = E_s \cdot d \cdot B$$

with:

v	=	drag velocity	[m/s]
F_s	=	cutting force	[N]
d	=	cutting depth	[m]
B	=	draghead width	[m]

The specific energy of different soil types is known within the dredging companies, but can be calculated also the linear cutting theories.

From the available thrust of the propellers the maximal available pulling force can be determined. For the calculation of the excavation production of the draghead, however, the average available force must be used. This depends among other things on the variation in the

cutting depth. $\frac{F_{piek}}{F_{average}}$ is usually between 1.25 and 1.5 and sometimes even 2.

The production is totally determined by the cutting process and is independent of the pump flow rate, if it does not interfere too much with the mixture forming.

2.5.1.1.3 Excavating in non-cohesive soil types

In non-cohesive soils, like sand and gravel, the excavation process within the draghead is physically complicated. If no jets are used to excavate the soil, the working of the draghead is totally based upon the erosion by the flow underneath the rims of the draghead generated by the dredgepump. The pressure difference over the draghead generated by this flow causes a groundwater flow underneath the draghead (Figure 2.124 and 2.125).

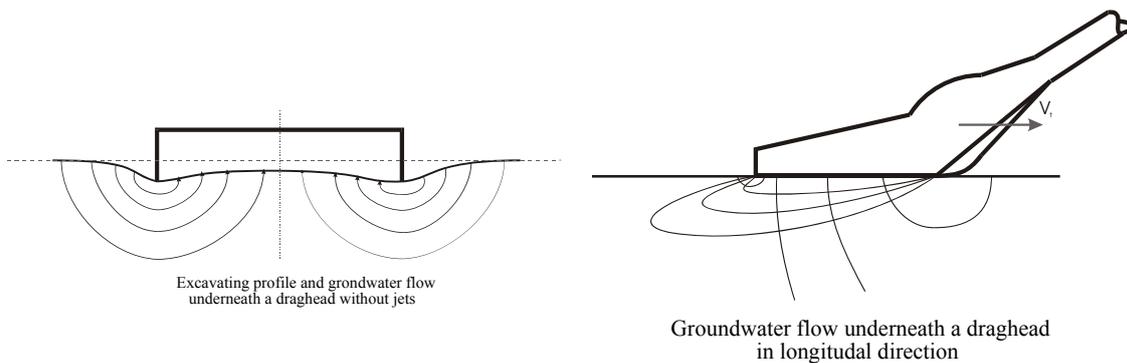


Figure 2-124

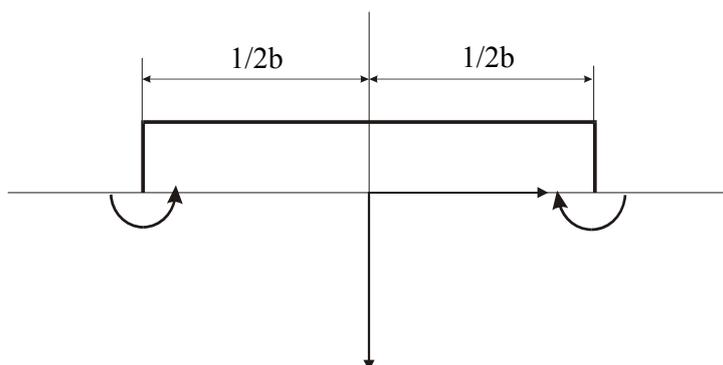


Figure 2-125

For the 2-D stationary situation this groundwater potentials can be describe according to :

$$\phi = \frac{H}{\pi} \left[\arctan \left(\frac{x + \frac{1}{2}b}{y} \right) - \arctan \left(\frac{x - \frac{1}{2}b}{y} \right) \right]$$

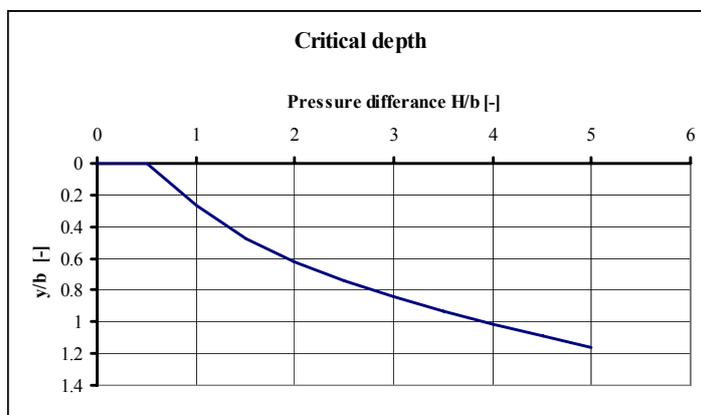
The vertical groundwater flow under the draghead generated by this pressure difference causes a decrease of the effective stress in the sand. The critical hydraulic gradient for moving the particles follows from the equilibrium of the flow force with submerged weight of the particles. This leads to the equation:

$$\frac{d\phi}{dy} = -\frac{H}{\pi} \left[\frac{x + \frac{1}{2}b}{y^2 + \left(x + \frac{1}{2}b\right)^2} - \frac{x - \frac{1}{2}b}{y^2 + \left(x - \frac{1}{2}b\right)^2} \right] = -\frac{100-n}{100} \frac{\rho_p - \rho_w}{\rho_w} \approx 1$$

$$\text{For } x=0 \Rightarrow \frac{d\phi}{dy} = \frac{H}{\pi} \left[\frac{b}{y^2 + \frac{1}{4}b^2} \right] > 1 \Rightarrow \frac{Hb}{\pi} > y^2 + \frac{1}{4}b^2 \Rightarrow \frac{y}{b} < \sqrt{\frac{H}{b\pi} - \frac{1}{4}}$$

$$\text{For } y=0 \Rightarrow \frac{d\phi}{dy} = \frac{H}{\pi} \frac{b}{x^2 - \frac{1}{4}b^2} > 1 \Rightarrow \frac{Hb}{\pi} > x^2 - \frac{1}{4}b^2 \Rightarrow \frac{x}{b} < \sqrt{\frac{H}{b\pi} + \frac{1}{4}}$$

For $y=0$ this condition is always fulfilled. The term $(100-n)/100$ is the ratio sand particles over the total volume. For $Y=0$ the condition is always fulfilled because X/b is always smaller than or equal to $1/2$



Critical depth for $X=0$ is shown in the Figure next and shows relatively very high critical depth!

Figure 2-126

However, by the erosive action of the water entraining into the draghead, the grains want to move from each other (dilatancy) and a pore pressure drop, which increases the effective stresses of the grains. Which process is dominant depends on a number of factors. The question is if the ground water flow is able to keep up with the increase of pore volume of the sand. If that is not the case than a further decrease of the water pressures arises, with a decreased erosion process as a result.

The ratio between the mixture flow rate Q_m and the erosion flow rate Q_e as function of the C_{vd} is:

$$Q_{mixture} = Q_{erosion} + Q_{pores} + Q_{sand} \Rightarrow 1 = \frac{Q_{erosion}}{Q_{mixture}} + \frac{Q_{pores}}{Q_{mixture}} + \frac{Q_{sand}}{Q_{mixture}}$$

$$1 = \frac{Q_{erosion}}{Q_{mixture}} + \frac{n_0}{1-n_0} \frac{Q_{sand}}{Q_{mixture}} + \frac{Q_{sand}}{Q_{mixture}} \Rightarrow 1 = \frac{Q_{erosion}}{Q_{mixture}} + \frac{n_0}{1-n_0} C_{vd} + C_{vd}$$

$$\text{or } \frac{Q_{erosion}}{Q_{mixture}} = 1 - \frac{C_{vd}}{1-n_0}$$

With:

$$Q_{mixture} = \text{The mixture or suction pump flow rate.} \quad [m^3/s]$$

$$Q_{erosion} = \text{the erosion flow rate, sucked from underneath the rims of the draghead} \quad [m^3/s]$$

Q_{sand}	= The sand flow rate.	$[\text{m}^3/\text{s}]$
Q_{pores}	= The flow rate of the pore water present in the sand.	$[\text{m}^3/\text{s}]$
C_{vd}	= transport concentration	$[-]$
n_0	= pore ratio	$[-]$

This volume balance is shown in Figure 2.127. From a physical point of view, the concentration will increase as well when the erosion or crack velocity underneath the draghead increases (erosion line in the Figure 2.127) when Q_{mixture} remains constant). From experience it is known that for a certain type of draghead without jets, the concentration C_{vd} is only slightly dependent on the mixture flow rate, which points out that the quotient $\frac{Q_{\text{erosion}}}{Q_{\text{mixture}}}$ remains almost

constant. As a rule of thumb for the average erosion depth can be written: $d = \alpha \frac{k^{0.3}}{v_t^{0.9}}$.

In this k is the water permeability of the sand and v_t the trail speed of the draghead, both in m/s. The factor α is dependent on the dimensions of the draghead.

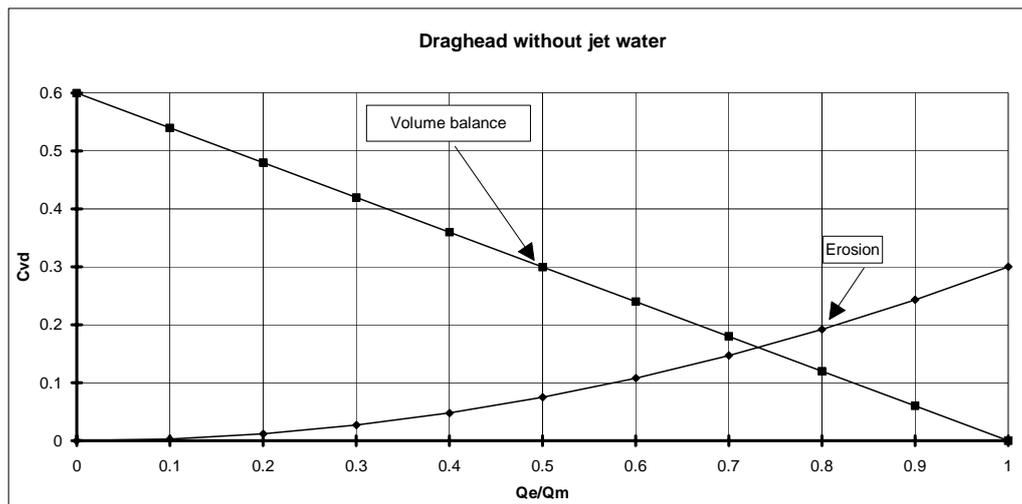


Figure 2-127

With increasing width of the draghead the average depth will decrease, looking to the erosion process around the draghead. Unfortunately there is yet insufficient knowledge of this process to determine an optimum width of the draghead. The maximum concentration C_{vd} for the dragheads without jets remains limited to 15 % in loose sand. In a lot of cases however C_{vd} is smaller than 10 %.

If jets are used to excavate the sand, this decreases the erosion flow rate, because the volume balance should be fulfilled:

$$Q_{\text{mixture}} = Q_{\text{erosion}} + Q_{\text{jet}} + Q_{\text{sand}} + Q_{\text{pores}} \quad [\text{m}^3/\text{s}]$$

With:

$Q_{mixture}$	= the mixture or suction pump flow rate.	$[m^3/s]$
$Q_{erosion}$	= the erosion flow rate, sucked from underneath the rims of the draghead	$[m^3/s]$
Q_{jet}	= the jet pump flow rate.	$[m^3/s]$
Q_{sand}	= the sand flow rate.	$[m^3/s]$
Q_{pores}	= the flow rate of the pore water present in the sand.	$[m^3/s]$
C_{vd}	= transport concentration	$[-]$
n	= pore ratio	$[-]$

Furthermore:

$$\frac{Q_{sand}}{Q_{mixture}} = C_{vd} \quad (\text{transport concentration}) \text{ and:}$$

$$Q_{pores} = \frac{n}{1-n} Q_{sand}$$

With: n = pore ratio $[-]$

From the above mentioned continuity condition now follows:

$$1 - \frac{C_{vd}}{1-n} = \frac{Q_{erosion}}{Q_{mixture}} + \frac{Q_{jet}}{Q_{mixture}}$$

This is a bundle of lines under 45° in a $\frac{Q_{erosion}}{Q_{mixture}}, \frac{Q_{jet}}{Q_{mixture}}$ diagram for constant values of $\frac{C_{vd}}{1-n}$ (Figure 2.128).

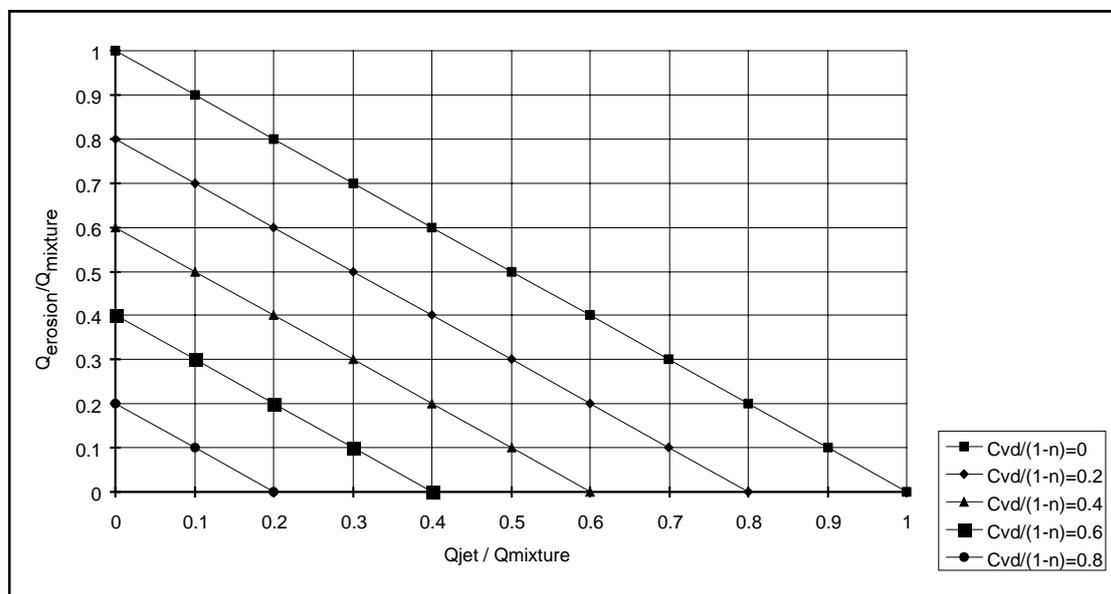
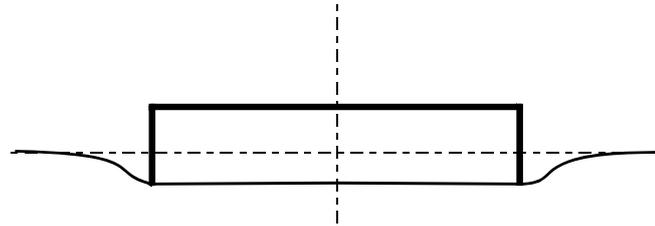


Figure 2-128 Relation between capacities to fulfill the volume balance in the draghead

This picture shows that high concentration or mixture densities can be reached only for low values of $\frac{Q_{erosion}}{Q_{mixture}}$ and $\frac{Q_{jet}}{Q_{mixture}}$

In case of a large jetpump capacity the erosion flowrate can get negative value resulting in spillage behind the draghead.

With jets well divided over the width of the draghead an erosion-profile can be reached with an almost constant depth over the full width



Erosion profile for a draghead with a well designed jetsystem

Figure 2-129

As said earlier, a reasonable assumption is that the jet- production is linear with the total momentum flux of the jet system and independent of the trail speed. The momentum $I = \rho_w Qu$.

$$M_{\text{sand}} = \alpha \cdot I = \alpha \rho_w \cdot Qu = \alpha \rho_w \cdot Q \sqrt{\frac{2p_{\text{jet}}}{\rho_w}}$$

- I = Momentum in N
- M_{sand} = Eroded sand mass in kg/s per jet
- p_{jet} = Jet pressure at the nozzle in Pa
- Q = Jet capacity in m^3/s
- U = Jet velocity at the nozzle in m/s
- α = Coefficient depending on the particle size, jet pressure, jet capacity and trailspeed.
A reasonable assumption for alpha is $\alpha=0.1$
- ρ_w = Water density in kg/m^3 .

When the nozzle are divided well over the width of the draghead the mass M should fulfill the relation:

$$\sum_{\text{all jets}} M_{\text{sand}} = B \cdot d \cdot v_{\text{trail}} \frac{\rho_{\text{situ}} - \rho_{\text{water}}}{\rho_{\text{particle}} - \rho_{\text{water}}} \rho_{\text{particle}}$$

- B = Width draghead in m.
- D = Eroded layer thickness in m
- v_{trail} = Trailspeed in m/s
- ρ_{situ} = Density soil in situ kg/m^3
- ρ_{particle} = Particle density in kg/m^3

When the trailspeed is said to 1.5 m/s, which equals 3 knots and the product B.d can be calculated.

In general the effective of the jet decreases somewhat with increasing pressure at constant momentum. This means that low pressure- high capacity jets are more effective than high pressure-low capacity jets. They use more specific energy too. On the other hand however, much jetwater dilutes the mixture density (Figure 2.128). So the designer has to search for the optimum solution between cost (power) and production

Jet-water is used for loosening the soil within the dragheads, as well as to assist the process during discharging the load, either by dumping or by pumping ashore. The flow rate of the water pump is between 20 to 30 % of the sand pump flow rate and the pressure is usually between 5 and 15 bar.

The required pressure can be calculated using the same basic formula's as mention in the forgoing chapter.

$$M_{\text{sand}} = C_{\text{vd}} Q_m \rho_{\text{sand}} = \alpha \rho_w Q_{\text{jet}} \sqrt{\frac{2p}{\rho_w}}$$

$$p = \frac{1}{2} \left[\frac{\rho_{\text{sand}} C_{\text{vd}} Q_m}{\sqrt{\rho_w} \alpha Q_{\text{jet}}} \right]^2$$

The results are given in fig 2-130

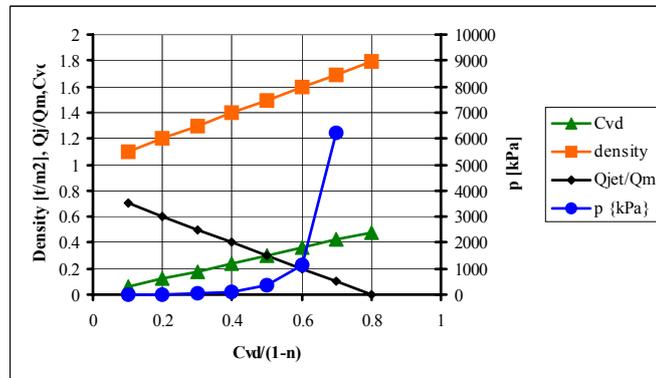


Figure 2-130

The breaking up of the coherence of the soil, which is done in the draghead either by the erosion or by jets, can also be done by the gravity under certain circumstances. When the sand layer has sufficient thickness a narrow path is deepened to full depth as quickly as possible. Next the trailing suction hopper dredger keeps on dredging at the base of the embankment. By the breaches process the embankment will slowly move perpendicular to the trail direction (Figure 2.131). Besides the breach causes the sand to be looser packed at the bottom of the embankment. Also mixing of various materials takes place.

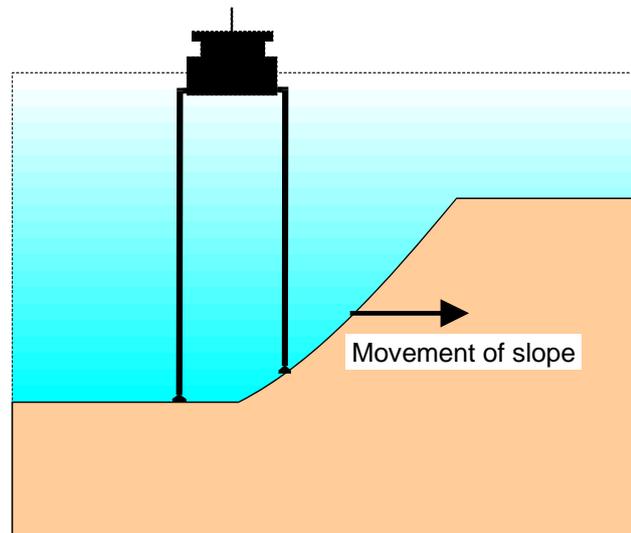


Figure 2-131

The disadvantage of this method is, of course, that the material has to be obtained at greater depth. If the "horizontal" or "vertical" method is preferred depends therefore on the grain distribution of the various layers, the suction depth and how far the pump of the trailing suction hopper dredger is below the waterlevel.

The dredging **soft rock** by trailing suction hopper dredgers is only done in exceptional cases. In fact only in those cases where the operating hours of a cutter suction dredger are so limited by the weather conditions that it is not profitable or where the amounts to be dredged are so limited that the mobilization of a cutter suction dredger is not profitable.

Dredging rock with a trailing suction hopper dredger is not just done. The dredger has to be equipped for that. This means that the dragheads, the suction pipes and hull attachments able to resist the forces that during the ripping of rock.

2.5.1.2 The transport of the slurry

In the course Dredging Processes II (Wb 3414) the pumping of sand-water mixtures will be discussed extensively, so that only specific cases will be discussed here with regard to the transport and deposition in the hopper of the dredged material.

If the trailing suction hopper dredger is limited for its dredging depth to a dredging depth of 30 m than one fixed pump-speed is sufficient. If the ship has to dredge over a deeper range of depth or equipped with an additional submerged pump, than the question rises whether the flow rate variations are not too high between the suction in shallow waters and at the maximum dredging depth. The maximum suction depth determines the highest pump speed, if the pump is sufficiently under water. If this pump-speed is fixed than the flow rate when dredging in shallow water will significantly larger than dredging at the maximum depth. Since overflow losses increase linear with the flow rate it must be considered if it is economical to equip the dredgepump with a speed control to keep the flow rate constant at different depth.

Furthermore the pump will have to be optimized for either the dredging operations or pumping ashore, depending on the total expected time of operations under these modes.

When no submerged pump is fitted, it might better to pursuit for straight a piping system in the suction line, even if lead to an extra elbow in the discharge line.

2.5.1.3 The loading

In order to obtain the highest possible fill rate during the loading the hopper with nonsettling slurries, the **poor mixture** (mixture with a too little density) van be pumped straight overboard. An automated valve controller can easily do this. However, with the increase of environmental requirements this is banned nowadays.

For settling mixtures like pieces of clay, sand and gravel, a part will settle and a part will leave the hopper through the overflow. A rule of thumb sometimes followed is that all with a $d_{50} < 75 \mu\text{m}$ flows overboard.

A measure for the quality of the settling process is the relative cumulative overflow loss. This is defined as the ratio between the total amount of solids that leave the hopper through the overflow and the total amount of solids pumped in the hopper. This relative cumulative overflow loss is, except for the material properties as grain size, the grain distribution, shape and specific mass, also dependent on the loading conditions like the flow rate, concentration, turbulence intensity, temperature and the hopper geometry.

These overflow losses are, like mentioned above, largely dependent on the parameter

$\frac{\left(\frac{s}{Q}\right)}{B \cdot L} = \frac{s}{s_0}$ and less of $\frac{\left(\frac{s}{Q}\right)}{B \cdot H} = \frac{s}{v_0}$ (see reader: Dredging Processes I (Wb3413)). The term $\frac{Q}{B \cdot L}$ is called the surface load.

In these:

Q	=	the total in-going mixture flow rate	[m ³ /s]
L	=	the length of the hopper	[m]
B	=	the width of the hopper	[m]
H	=	the settling height in the hopper	[m]
s	=	the settling velocity	[m/s]
s ₀	=	Surface load	[m/s]
v ₀	=	Horizontal velocity	[m/s]

The first parameter is the ratio between the time the particle needs to settle and the time that it stays in the hopper. The second parameter is the ratio between the horizontal velocity in the well and the settle velocity of the particle and is a measure for the degree of turbulence in the hopper.

The overflow losses as function of the earlier mentioned terms: $\frac{\left(\frac{s}{Q}\right)}{B \cdot L} = \frac{s}{s_0}$ and

$\frac{\left(\frac{s}{Q}\right)}{B \cdot H} = \frac{s}{v_0}$ are “reasonably” approximated by the theory of **Camp**, although the sedimentation process in the hopper is quite different as assumed by Camp. For a real understanding of the sedimentation process the reader is referred to the thesis of Dr.Ir. C. van Rhee .

In Figure 2.132 the settled part (**removal ratio**), so $R_t = (1 - \text{overflow losses})$, is shown as function of these two parameters.

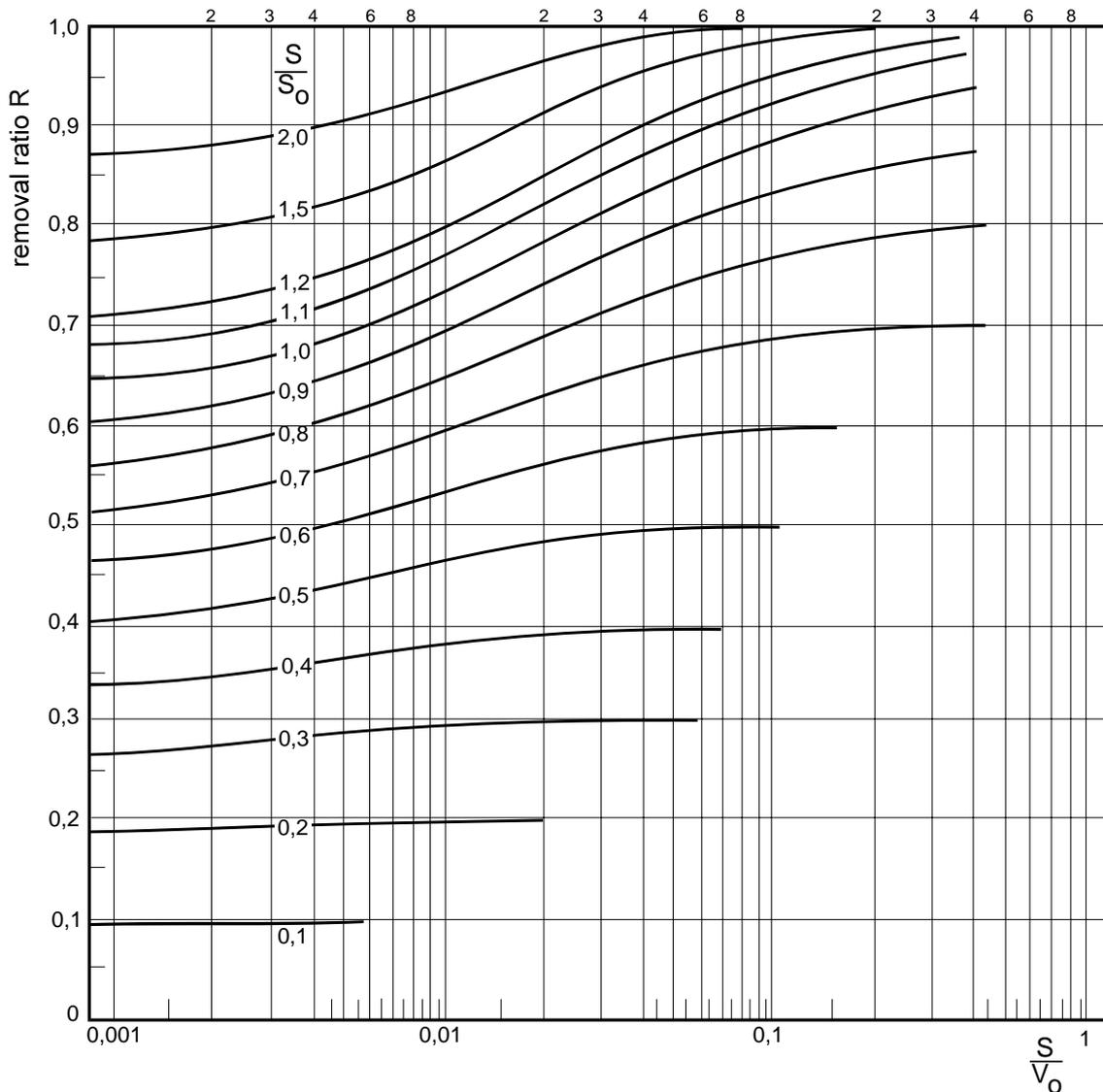


Figure 2-132 Camps diagram

By calculating the settling process in a number of steps the relative cumulative overflow losses can be determined as function of time or load rate. From the theory of Camp can be derived that the influence of the bed height is marginally. This implies that during the loading process the overflow losses are almost constant. Although in practice loading curves are almost straight. The overflow rate is not.

2.5.1.3.1 Loading curve

Dependent on the way of payment, in cubic meters or in Tons Dry Solids (TDS), the contractor will like to know the development of the volume in m^3 or of the TDS in the hopper during loading. To do this it is necessary to measure the volume of the total load (sand and water). Acoustic silo indicators usually do this. The weight of the (useful) load is measured by determining the development of the draught as function of the time (chapter 2.2.2.1). From the volume and the weight of the useful load the volume in m^3 or the TDS can be determined if the volume weight γ_z of the sand and the specific weight ρ_k of the sand and the water ρ_w are known.

The loading curve can be divided in three phases:

1. Before the overflow is reached:

$$V_{load}(t) = Q_i t$$

$$G_{load}(t) = V_{load}(t) \gamma_i = Q_i \gamma_i t$$

$$V_{sand}(t) = Q_i \frac{\gamma_i - \gamma_w}{\gamma_z - \gamma_w} t$$

$$G_{sand}(t) = V_{sand}(t) \frac{\gamma_z - \gamma_w}{\gamma_k - \gamma_w} \gamma_k t = Q_i \frac{\gamma_i - \gamma_w}{\gamma_z - \gamma_w} \cdot \frac{\gamma_z - \gamma_w}{\gamma_k - \gamma_w} \gamma_k t = Q_i \frac{\gamma_i - \gamma_w}{\gamma_k - \gamma_w} \gamma_k t$$

In this:

G_{load} and V_{load} , the weight and the volume of the total load, so sand and water. V_{sand} the sand volume (including the pores) in the hopper and G_{sand} the weight of the sand (excluding the pore water), so TDS.

Q_i and Q_u are the in- and out-going flow rate. γ_i , γ_k , γ_z and γ_w are the volume weights ($\gamma = \rho g$) of the mixture, the sand grains, the sand volume with the pores and the water.

In this it is silently assumed that the hopper is totally empty before the start of the suction. If this is not the case than volume must be increased with the value V_0 and the weight with G_0 .

2. When the overflow is reached t_{ov} , but the ship is not yet on its dredge mark, the hopper volume remains constant (constant volume loading).

$$Q_i = Q_u$$

$$G_i = Q_i \gamma_i \text{ and } G_u = Q_u \gamma_u \text{ with } \gamma_i > \gamma_u > \gamma_w$$

and therefore:

$$V_{load}(t) = V_{hopper}(t) = \text{constant} = V_{load}(t_{ov})$$

$$G_{load}(t) = G_{ov} + Q_i (\gamma_i - \gamma_u) (t - t_{ov})$$

$$V_{sand}(t) = V_{sand}^{ov} + Q_i \frac{(\gamma_i - \gamma_u)}{(\gamma_z - \gamma_w)} (t - t_{ov})$$

$$G_{sand}(t) = G_{sand}^{ov} + Q_i \frac{(\gamma_i - \gamma_u)}{(\gamma_k - \gamma_w)} (t - t_{ov})$$

V_{sand}^{ov} and G_{sand}^{ov} are the volume of the sand and the weight of the grains at the moment the overflow is reached.

3. The overflow is reached and the ship is on the dredge mark.

In this case the weight of the total load (water and sand) remains constant (constant tonnage loading).

$$G_i = G_u = Q_i \gamma_i = Q_u \gamma_u \text{ and therefore } Q_u = Q_i \frac{\gamma_i}{\gamma_u}$$

$$V_{load}(t) = V_{load}^{mark} - Q_u(t - t_{mark}) = V_{load}^{mark} - Q_i \frac{\gamma_i}{\gamma_u}(t - t_{mark})$$

$$G_{load}(t) = G_{mark} = \text{constant}$$

$$V_{sand}(t) = V_{sand}^{mark} + Q_i \frac{\gamma_i - \gamma_u}{\gamma_z - \gamma_w}(t - t_{mark})$$

$$G_{sand}(t) = G_{sand}^{mark} + Q_i \frac{\gamma_i - \gamma_u}{\gamma_k - \gamma_w} \gamma_k (t - t_{mark})$$

V_{sand}^{mark} and G_{sand}^{mark} are the volume of the sand with pores and the weight of the sand grains (TDS) on the moment the hopper reaches the valid dredge mark.

The total load curve is now known in mass and volume if Q_i , γ_i , γ_u , γ_k , γ_h and γ_w are known. γ_u can be determined from the overflow losses and γ_v depends on the type of soil.

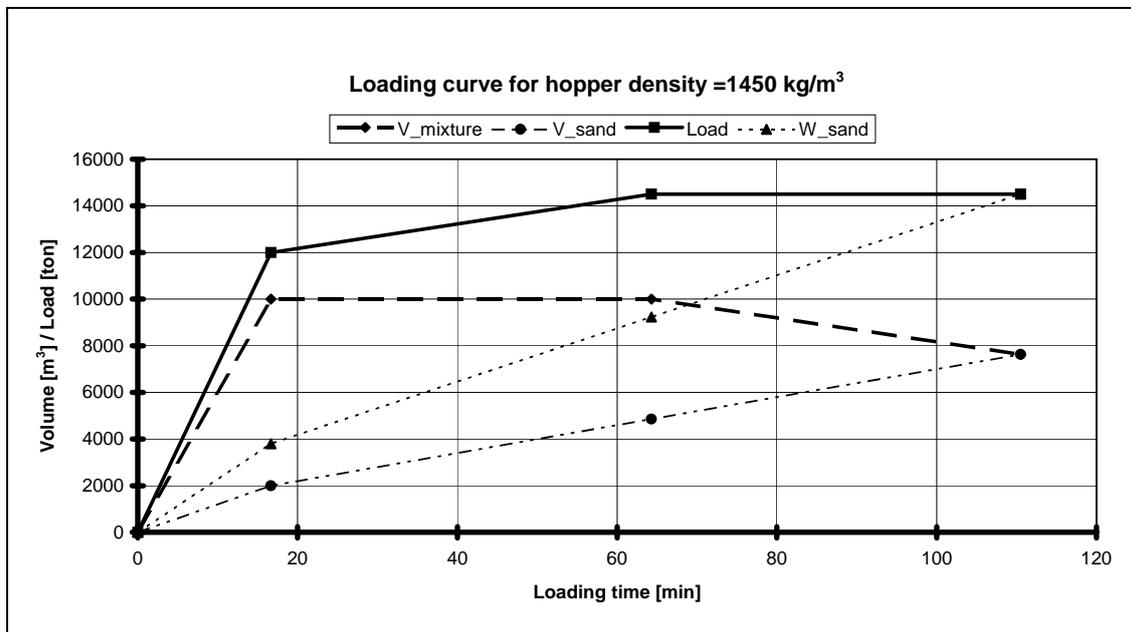


Figure 2-133

For pure **constant volume hoppers** the weight of the load is proportional to the draught of the ship. This increases in time, though the mixture-volume in the hopper remains constant.

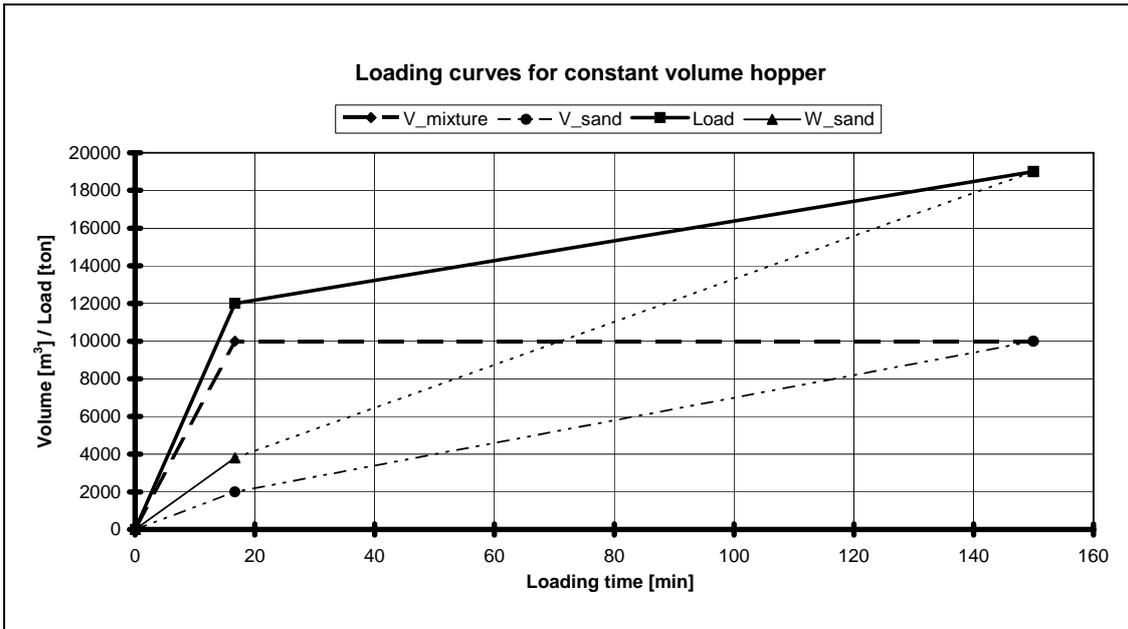


Figure 2-134

This does not account for the pure constant tonnage hoppers. Then the draught remains constant after reaching the overflow (Figure 2.135).

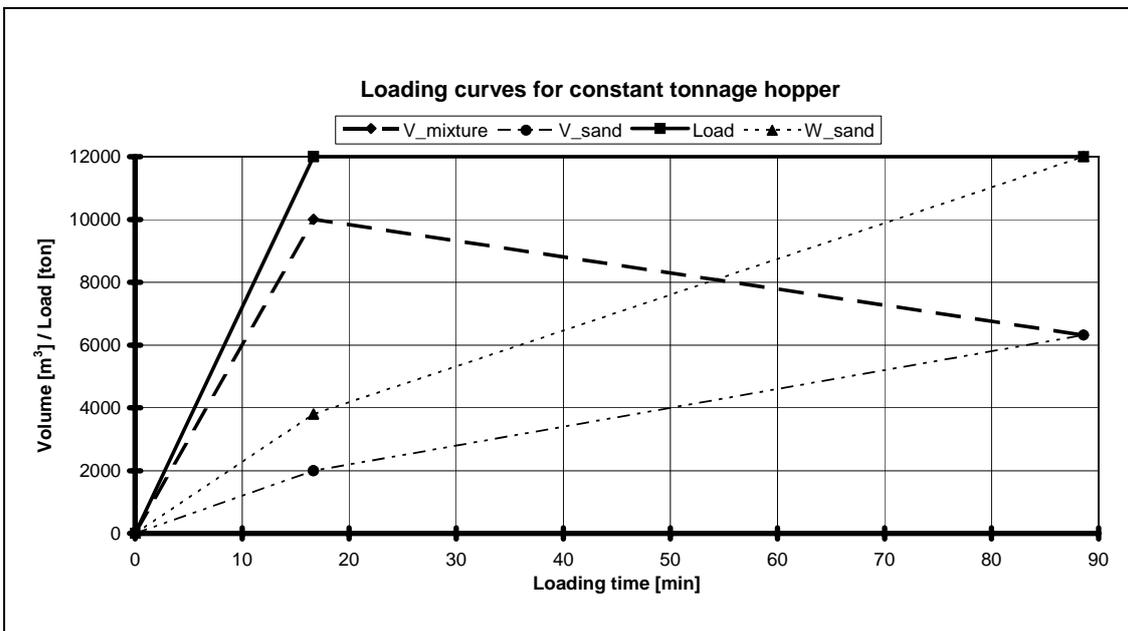


Figure 2-135

To calculate the weight of the load extra data is needed: the volume of the mixture and the volume-weight (or density) of the sand in the hopper. The first quantity is measured with silo indicators and the second by probing on several trips the volume of the sand.

Now the determination of the load during the dredging process is done as follows:

- Before the start of the dredging the displacement and the weight of the water in the hopper is determined. The displacement by measuring the draught of the vessel and the water-volume by the silo indicators.

$$\text{displacement empty ship} = \frac{\text{displacement}}{\text{volume water in hopper}} \cdot \rho_{\text{water}} \cdot g$$

- During dredging the fore and aft draught of the ship is measured continuously and so the displacement as well as the mixture volume by means of silo indicators.
- By subtracting the start values from the momentary values of the displacement and the mixture volume, the weight of the dry load (TDS) can be determined with the following formula.

$$\text{TDS} = \frac{\frac{G_{\text{load}}}{V_{\text{load}}} - \gamma_w}{\gamma_k - \gamma_w} \gamma_k V_{\text{load}}$$

$$\frac{G_{\text{load}}}{V_{\text{load}}} = \gamma_{\text{load}} \text{ is the volume weight of the mixture in the hopper.}$$

Though the load nowadays usually is expressed in TDS, it does not imply that payment is also dependent on the amount of TDS. This can be:

1. ton dry solid (TDS)
2. m³ in the hopper (means of transport)
3. m³ in the excavation

The mutual relation between these quantities is:

TDS with volume load in the hopper:

$$V_{\text{load}} = \frac{\text{TDS}}{\gamma_{\text{grains}}} \left(\frac{\gamma_{\text{grains}} - \gamma_{\text{water}}}{\gamma_{\text{load}} - \gamma_{\text{water}}} \right)$$

Therefore the conversion factor of TDS to m³:

$$f_v = \frac{V_{\text{load}}}{\text{TDS}} = \frac{1}{\gamma_{\text{grains}}} \cdot \left(\frac{\gamma_{\text{grains}} - \gamma_{\text{water}}}{\gamma_{\text{load}} - \gamma_{\text{water}}} \right)$$

And for m³ to TDS:

$$f_{\text{TDS}} = \frac{\text{TDS}}{V_{\text{load}}} = \gamma_{\text{grains}} \left(\frac{\gamma_{\text{load}} - \gamma_{\text{water}}}{\gamma_{\text{grains}} - \gamma_{\text{water}}} \right)$$

Shown in Figure 2.136.

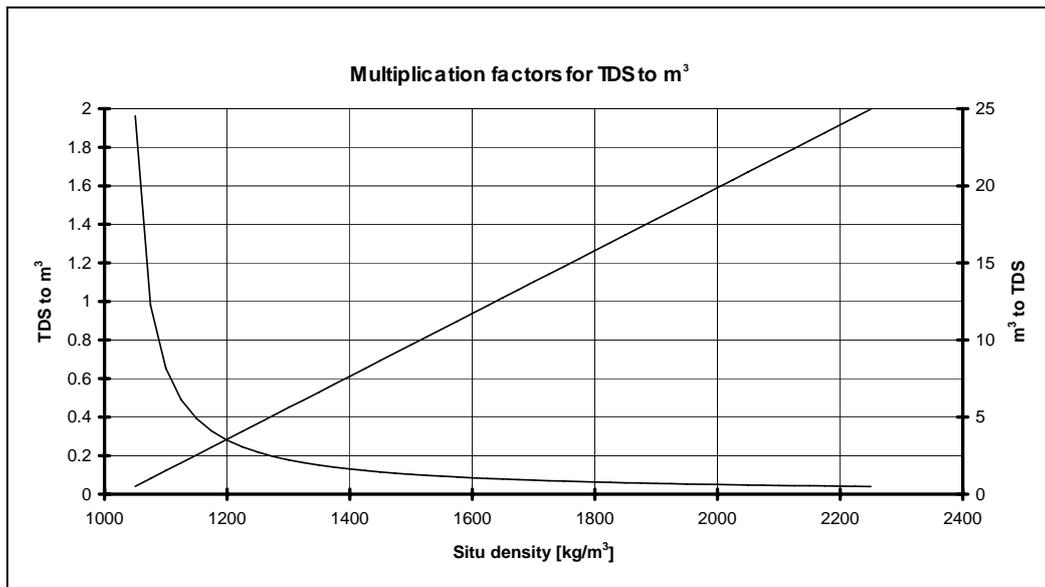


Figure 2-136

An aspect that also takes place during loading is the change in the volume weight of the dredged material, the **bulking**, which can be positive, so more, as well as negative, so less. The production unit in the dredging industry is the cubic meter per time unit. Unfortunately this is not an unambiguous unit. A m^3 in excavation appears to be a "different" m^3 after settlement in the well. Because sand grains in the hopper are usually stacked looser than in situ. The volume weight in the hopper is lower than the **situ volume weight**. Also, as a result of overflow losses, more fine sand particles will flow overboard than coarse particles. If these particles are located in a matrix of coarser particles than the volume weight will decrease even if the stacking of the matrix remains the same. If this phenomena happens in the dredged material can be simply shown by comparing the sand curve with the Füller-distribution (Figure 2.137).

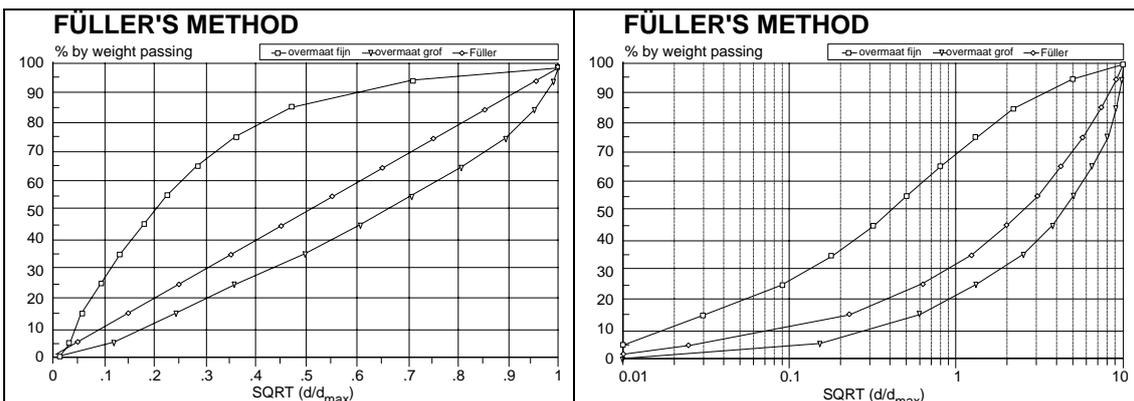


Figure 2-137

In a Füller-distribution the cumulative grain distribution, given as function of $\sqrt{\frac{d}{d_{\max}}}$, is a pure straight line. Such a distribution appears to give a maximum volume weight, which implies that the pores are constantly filled with the smaller particles. If the gradient of the smaller particles is above the Füller-distribution than there is a surplus of fine material and the above mentioned phenomenon would not show. If the gradient of the fine material is below the Füller-

distribution than the fine material is embedded in the coarser material and the phenomenon shows.

The volume weight in the hopper is usually lower than in situ. Dependent on the grain distribution, a situ m^3 takes the same or more space in the hopper, caused by the increase of the ratio, which are filled with water. So the water takes this larger volume.

Example:

Assume the in situ density of the sand ρ_1 and the density in the hopper ρ_2 . The specific weight of the sand is ρ_k and of the water ρ_w . The cumulative overflow losses are ov and according the Füller distribution there is a surplus of fine material. If the situ volume is V_1 , then the volume in the hopper with in-situ density $(1-ov) V_1$. The weight of solids of this volume must be equal to the solid weight of the volume V_2 .

Weight of the volume V_1 for ρ_1 :

$$G_1 = (1-ov) \cdot V_1 \cdot \frac{\gamma_1 - \gamma_w}{\gamma_k - \gamma_w} \cdot \gamma_k \quad \text{with } \gamma = \rho g$$

Weight of the volume V_2 :

$$G_2 = V_2 \cdot \frac{\gamma_2 - \gamma_w}{\gamma_k - \gamma_w} \cdot \gamma_k$$

Since $G_1 = G_2$:

$$\frac{V_2}{V_1} = (1-ov) \cdot \frac{\gamma_1 - \gamma_w}{\gamma_2 - \gamma_w} = (1-ov) \cdot \frac{\rho_1 - \rho_w}{\rho_2 - \rho_w}$$

Example:

$$\begin{aligned} \rho_1 &= 2000 \text{ kg/m}^3 \\ \rho_2 &= 1900 \text{ kg/m}^3 \\ \rho_{\text{water}} &= 1020 \text{ kg/m}^3 \\ ov &= 10 \% \end{aligned}$$

$$\rightarrow \frac{V_1}{V_2} = (1-ov) \cdot \frac{2000-1020}{1900-1020} = 0.9 \cdot 1.11 = 1.0$$

So the volume in the hopper occupies the same space as the in the excavation. It has been silently assumed that the overflow losses do not flow back into the winning area. If that is the case than the term $(1-ov)$ is discarded and the delivery becomes 11 %.

If the fine sand particles are situated in a matrix of coarser particles than, for a similar stack of the coarser particles, $G_2 = 0.9 G_1$ with $V_1 = V_2$. This leads to:

$$(1-ov) \cdot V_1 \cdot \frac{\gamma_1 - \gamma_w}{\gamma_k - \gamma_w} \cdot \gamma_k = V_2 \cdot \frac{\gamma_2 - \gamma_w}{\gamma_k - \gamma_w} \cdot \gamma_k \Rightarrow (1-ov) \cdot (\rho_1 - \rho_w) = \rho_2 - \rho_w$$

This gives in the example:

$$\rho_2 = 0.9 \cdot \rho_1 + 0.1 \cdot \rho_w = 1800 + 102 = 1902 \text{ kg/m}^3$$

If all overflow losses remain in the winning area than this still holds but as a result the original layer will be covered with 10% fine material at the end of the work.

When sucking very loose sand the bulking can be smaller than 1. The bulking is then called negative. When dredging firm clay the bulking in the hopper is substantial, as is proven in the following example:

Assume the situ density of the clay as 2000 kg/m^3 . After cutting the pore percentage of the clay fragments is 40 %. The volume weight is then $\rho_2 = 0.6 \cdot 2000 + 0.4 \cdot 1020 = 1608 \text{ kg/m}^3$. And the bulking then will be:

$$\frac{V_1}{V_2} = \frac{2000 - 1020}{1608 - 1020} = 1.67$$

This can be seen directly as the new volume is only 60 % of the original.

During pumping ashore to a reclamation area, usually a negative bulking takes place, since the volume weight of the dump material is often higher than the volume weight of the material in the hopper and losses can occur at the reclamation.

2.5.2 Sailing from and to the discharging area

It will be clear that the sailing speed determined during the sea trials, for an empty as well as for a fully loaded ship, cannot be used as the average speed during the lifespan of the trailing suction hopper dredger. Between the dry dock periods the hull of the ship becomes overgrown with barnacles and seaweed and the propulsion engines and propellers are subjected to wear. This leads to a 5 to 10 percent lower average or operational speed in deep water than the sea trial speed. In general the trailing suction hopper dredger sails in seaways with a depth which gives the ship extra resistance. The trailing suction hopper dredger "feels" the bottom. The influence of the less deep seaway on the operational velocity is calculated with Lackenby's formula (Figure 2.138).

$$v_{\text{shallow}} = v_{\text{deep}} \left\{ 1 - \left[0.1242 \left(\frac{A}{(d+D)^2} - 0.05 \right) + 1 - \sqrt{\frac{c - \frac{1}{c}}{c + \frac{1}{c}}} \right] \right\}$$

with:

$$c = e^{g(d+D) \left(\frac{4}{v_{\text{deep}}} \right)}$$

in this:

d	=	keel clearance	[m]
D	=	draught of the ship	[m]
A	=	wet cross-section of mid ship	[m ²]

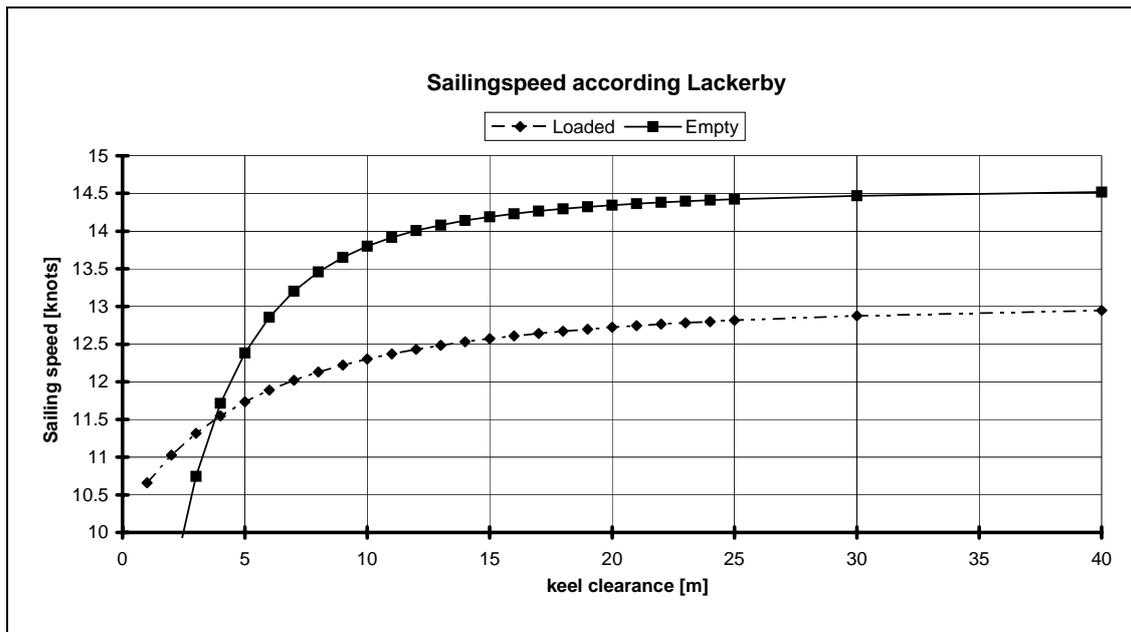


Figure 2-138

The sail time can now be determined with:

$$T_{vh} = \sum_{n+1}^N \frac{S_n}{(v_{vol})_n} \quad \text{with draught full}$$

$$T_{vt} = \sum_{n+1}^N \frac{S_n}{(v_{leeg})_n} \quad \text{with draught empty}$$

Another facet that has to be accounted for, are the sail-limitations in certain areas like harbors and narrow fairways. Furthermore the fairway has always to be checked for sufficient depth. In case of doubt it might even be wise to carry out a hydrographic survey

2.5.3 The discharge

As described in the chapter Technical Construction the trailing suction hopper dredger may be able to discharge its load in two ways, either by direct dumping or by means of the self-emptying installation by rainbowing or pumping to the shore.

If the load can be dumped directly it has to be known if the depth of the dump area is always sufficient to sail with opened doors or valves, even with extremely low water. The increasing lack of dump areas it happens regularly that the depth of the dump is limited. In such a case it is advised to make a dump plan to use the dump as efficient as possible.

For land reclamation works for which the first layer of the sand body can be dumped directly, a dump plan has to be made too, in order to dump directly as much material as possible, so that less material needs to be pumped ashore.

The discharge of the load through the bottom doors or valves usually costs little time. For free flowing soils this is done within several minutes. The discharge time increases when the material becomes finer and more cohesive. For plastic clays this can increase to half an hour. For such a material it has to be checked that no load, the **rest load**, remains in the hopper. There is a possibility that this rest load increases with the number of trips. It appears that the longer the clay remains in the hopper the more difficult it is to flush it out.

Discharge through the hopper self-emptying installation is done to:

- pump the load, through pressure piping to the shore.
- to heighten, for example, submerged dumps that are too shallow to dump; the so-called **rainbowing** (Figure 2.139).
- to accurately fill submerged dumps or to cover pipelines with the use of **pipe dumping**.

After the pumps are started and the water comes out of the pipe the discharge of the load is started on the side of the hopper that is the furthest away from the pump. This assures that the pump is always as deep under water as possible. Because the material in the hopper is in general pretty loose packed, the process looks a lot like the process of a stationary suction dredger. The sand breaches to the opening of the suction pipe.



Figure 2-139 Rainbowing

If the hopper is not equipped with an installation that improves the breaching by means of water-jets, than, as a rule of thumb, the discharge time is equal to the suction time. If the hopper is equipped with water-jets to fluidize or loosen the load, than the discharge time can be shortened considerably.

The discharge process through the hopper self-emptying installation behaves clearly like an S-curve. The discharge process is started usually slowly, because a quick start often leads to a blocked suction pipe. After that there is for 75 to 80 % of the time an almost constant high production. At the end of the unloading process the decreases almost linear zero (Figure 2.140).

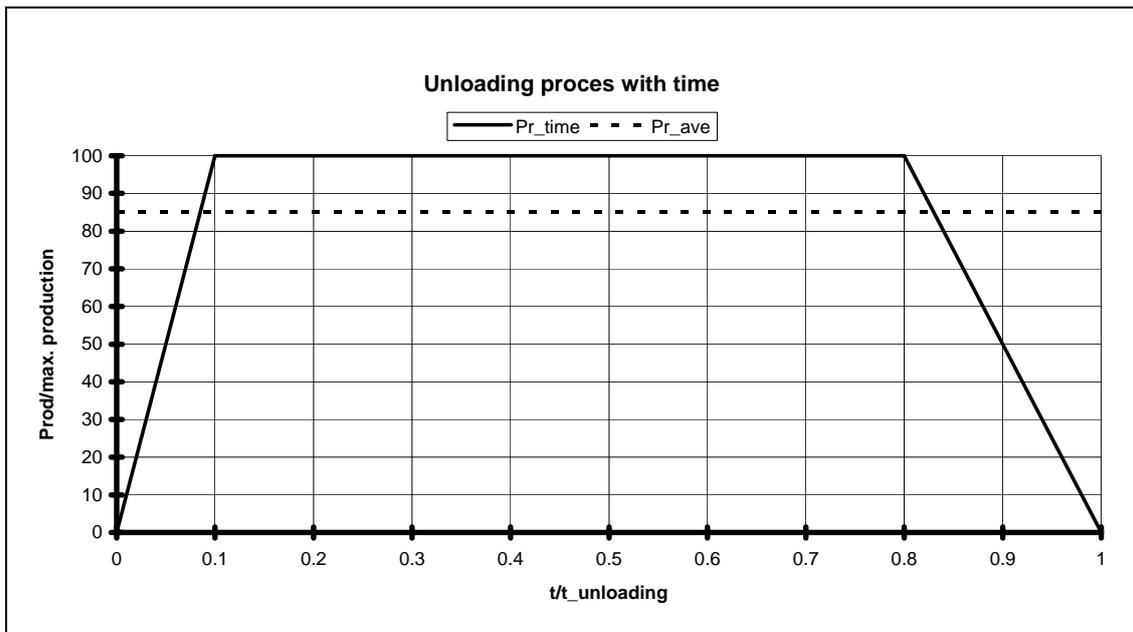


Figure 2-140 Production of the unloading process

In almost all self-emptying installation a rest load remains of around 5 %. By the fluidization process the rest loads of rocks and dirt accumulate, so that regularly the rest-load needs to be dumped on a dump.

2.5.4 The cycle production

The cycle consisting of: loading, sailing to, discharging, sailing back can be optimized simply.

The cycle production is defined as:

$$P_{cycle} = \frac{L(t)}{t_{suction} + t_{sail} + t_{discharge}}$$

If $t_{suction}$ and $t_{discharge}$ are considered constant than this production is optimal when the following condition is met:

$$\frac{dP_{cycle}}{dt_{suction}} = 0$$

This is the tangent to loading curve $L(t)$ that also crosses the negative y-axis in the point $t_{sailing} + t_{discharge}$ (Figure 2.141).

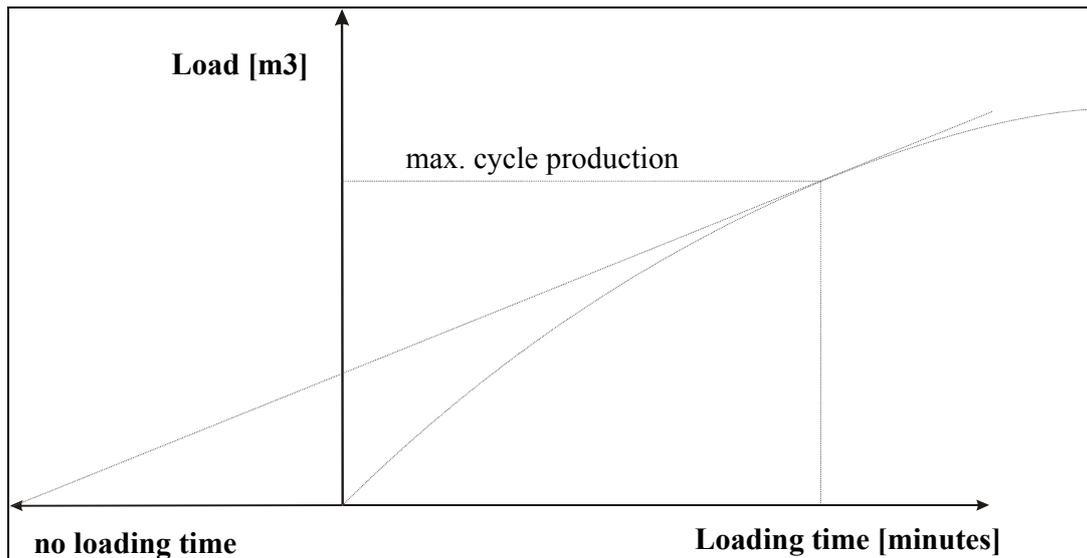


Figure 2-141 Optimal cycle production

This loading process can be made visible on board of the dredgers to determine the optimal load. However it should be noticed that the overflow losses increase sufficient at the end of the loading process to determine the optimal point.

2.5.5 The instrumentation

To support the **dredge master** instruments are available. Modern trailing suction hopper dredgers are equipped with suction pipe position indicators both in the longitudinal as in the transverse direction. Not only the position in relation to the bottom is indicated but also the position of the suction pipe and the draghead in relation to the ship and sometimes even the soil. Furthermore the dredge master has a direct view on the swell-compensators to judge if the dragheads are on the bottom. If this is not the case than indicators are necessary. For the suction process there are besides the vacuum and pressure indicators, also velocity and concentration indicators. With the aid of these instruments the suction chief will optimize the suction process by trial and error.

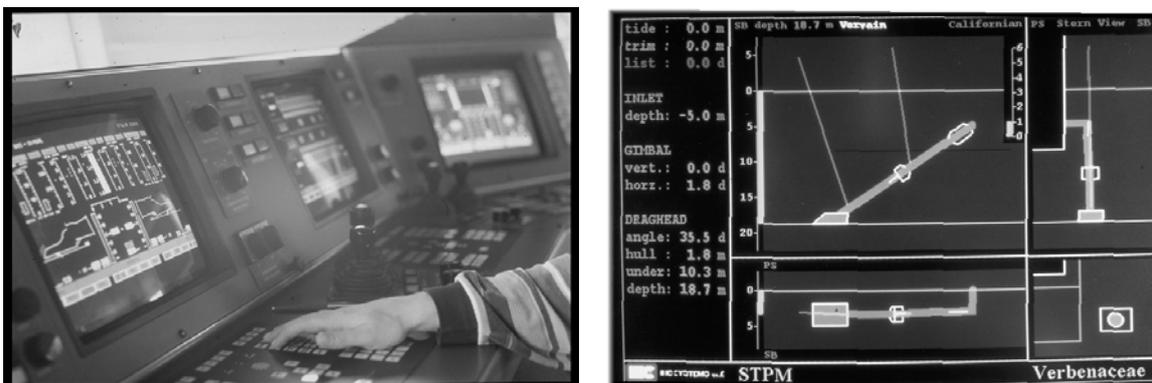


Figure 2-142 Instrumentation panels

2.6 Special designs of trailing suction hopper dredgers

2.6.1 The gravel suction dredger

Trailing suction hopper dredgers that collect aggregates for the concrete industry and road construction differ in several aspects from the "standard" trailing suction hopper dredger. These differences usually arise from economical considerations. Items that are of less use are left out, while others are added.



Figure 2-143 Gravel dredger Charlemagne

These include:

- **The maneuverability.** A lot of gravel suction dredgers are built to collect aggregates at sea. These are relative wide concessions where accurate dredging is of no or small concern. Furthermore there are long transportation distances. Therefore the requirements for the maneuverability are less strict than for the trailing suction hopper dredger that has to dredge frequently in busy fairways or ports. For this reason the gravel suction hopper dredger is equipped with only one screw.
- **The longer dredge cycle.** The longer sail distances mean that the suction time is only a small percentage of the total cycle time. Therefore it is much more economical to equip the ship with only one suction pipe and one dredge pump.
- Since the quality of the material determines the price, these ships are equipped with a **creening installation**. The "bad" material can then be put overboard. Of course it is also possible to load all the material (called **all-in** or **tout-venant**).
- A **discharge installation** with which it is possible to unload "dry" in every arbitrary port. Seldom a gravel suction hopper dredger has bottom doors or valves.



Figure 2-144 Screening installation

Since the concessions are increasingly further away from the land and therefore in deeper waters, submerged pumps on the suction pipe are also used on modern gravel suction hopper dredgers.

The discharge systems are of the drag system, clamshell or excavation wheel (Figure 2.114) principle that delivers the material from the hopper to a silo from which the material is distributed further via a conveyor belt. The way of operation does not differ much from the "classical" trailing suction hopper dredger. Instead of pumping the material straight into the hopper, it is now pumped into the screening installation, where it is separated into the required class(es). When sailing to the discharge area the drain installation is turned on to bring the load as dry as possible ashore.

Trailing suction hopper dredger for inland waters provides also sand and gravel to the concrete industry as well as sand for reclamation purposes. They do also maintenance dredging in river harbours Their design is much simpler than ordinary trailer suction hopper dredgers (Figure 2.145).



Figure 2-145 Trailing suction hopper dredger for inland waters

2.6.2 The stationary suction hopper dredger

The stationary suction hopper dredger is the predecessor of the trailing suction hopper dredger. In the most well known design the stationary suction hopper dredger has a hopper and behind it the pump room with one dredge pump. The suction pipe is directed however forward. Stationary suction hopper dredgers are single-screw ships. The propulsion engine directly drives the dredge pump.

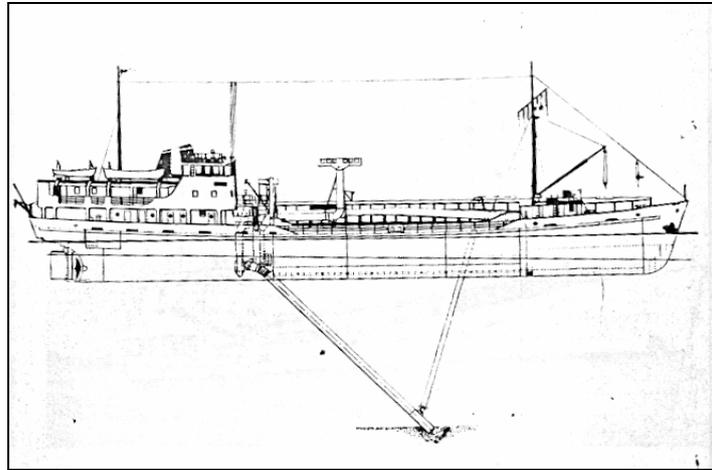


Figure 2-146 Stationary suction hopper dredger

The method of operation differs significantly from the trailing suction hopper dredger and is in principle equal to the suction dredger.

When dredging the vessel anchored in its borrow area. The amount of anchors needed depends strongly on the operational circumstances, like current and wind velocity, current and wind direction and shipping. If the circumstances are well than one or two front anchors are sufficient. If the dredging takes place in a tidal area where the current change direction depending on the tide, than also one or two aft anchors are placed. A second anchor is needed if the ship must be hauled frequently.

As with suction dredgers the stationary hopper dredger is used in free running sand. Dependent on the breach height the ship is slowly hauled in the direction of the suction direction. The loading of the hopper is similar to the process of the trailing suction hopper dredger.

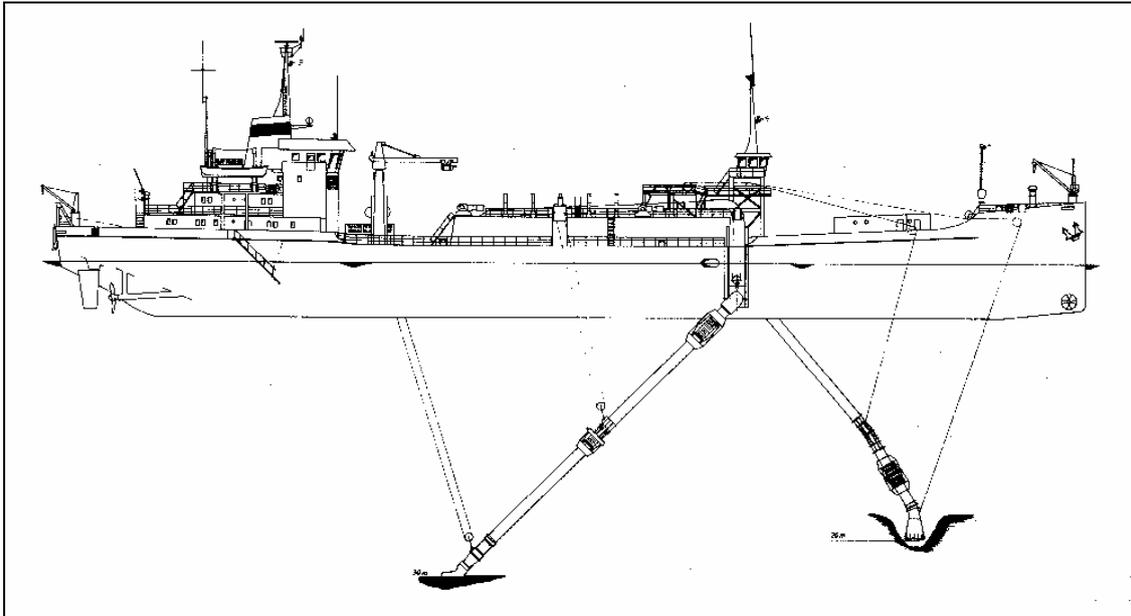


Figure 2-147 Trailing suction hopper dredger for stationary dredging

Sometimes the trailing suction hopper dredger is used as a “stationary dredgers” for certain works. To do this the dragheads are removed and if not already present an aft anchor is mounted. When arriving at the winning area first the aft anchor is placed. Dependent on the weather conditions the front anchor is also placed. Since the pipes put backwards the trailing suction hopper dredger works itself while dredging backwards. There are also trailing suction hopper dredgers that have the possibility to bring their suction pipe forward and are then able to work on the bow anchor (Figure 2.147). With well-breaching sand trailing suction hopper dredgers can also suck profiles with the drag suction method. The embankment must then be at all times more gentle than the suction pipes of the trailing suction hopper dredger. The trailing suction hopper dredger forces its way into the embankment with a velocity of 0.25 to 0.5 knots. The main advantage of this method is that no anchors are needed which gives more freedom of movement and a quicker leave in case of an emergency.

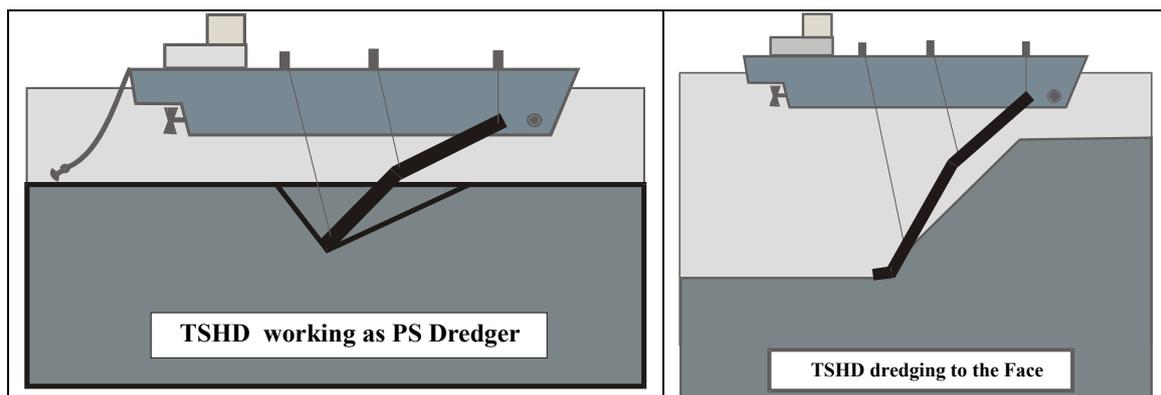


Figure 2-148 Trailer suction hopper dredger working in a plain suction mode

2.6.3 Boom dredgers

The boom dredger (Figure 2.149) is a special design of the trailing suction hopper dredger.



Figure 2-149 Boom dredger

It is equipped with a 50 to 60 meter long construction, **the boom**, that makes it possible to pump the dredged material immediately sideways back (**side casting**). This method of dredging is used in silt rich fairways, where it is cheaper to spray the material to the side, a hundred meters from the bank of the fairway instead of bringing it to a dump far away. Approach channels at the lake of Maricaibo in Venezuela are dredged in this manner

2.7 Literature

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