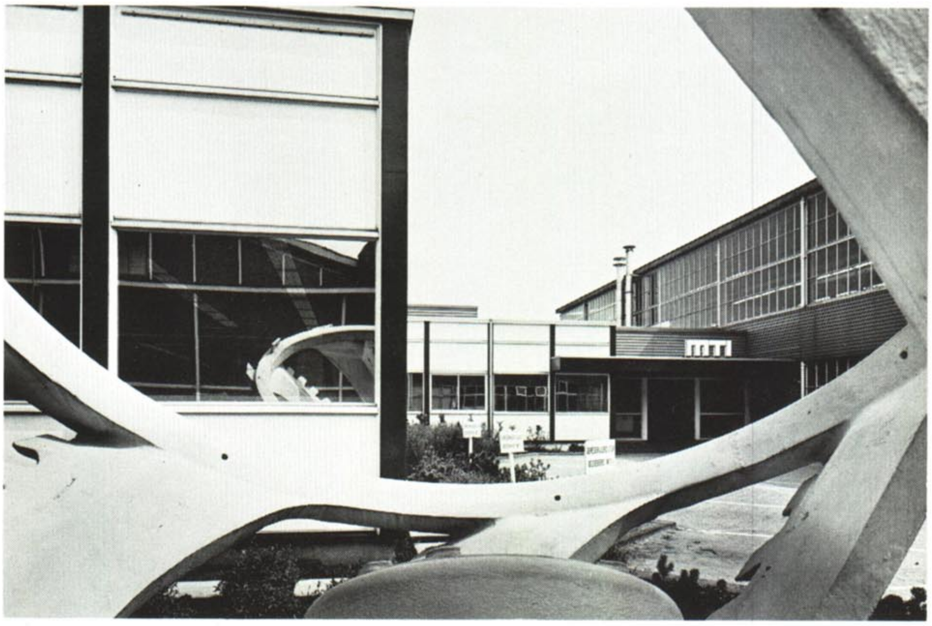
Centrifugal Dredgepumps 8

By
Ir. S. E. M. de Bree



This article is the eighth in a series written by Ir. S. E. M. de Bree, Head of the Mineral Technological Institute, the development laboratory of the Dredger Division of IHC Holland.
The earlier articles appeared in issues 77, 78, 80, 82, 84, 85 and 86.

Introduction

In previous articles in this series, we discussed a number of specific properties, details and characteristics of installations built around a single dredgepump.

The vast majority of modern large dredgers, whose function lies in the hydraulic transport of soil over considerable distances, are equipped with two or more pumps operating in series.

This article deals with some of the aspects of such installations.

Twin dredgepumps operating in series

Where two pumps are operated in series, it is of importance that they should be compatible. This implies that, in terms of flowrate, they are designed for the same working range; that from the flow point of view the passages and connexions are of similar dimensions; that the Q-H curves with respect to flowrate are of similar shape; and that, where diesel drive is employed, the positions of the nominal full-torque points of the prime movers are virtually identical.

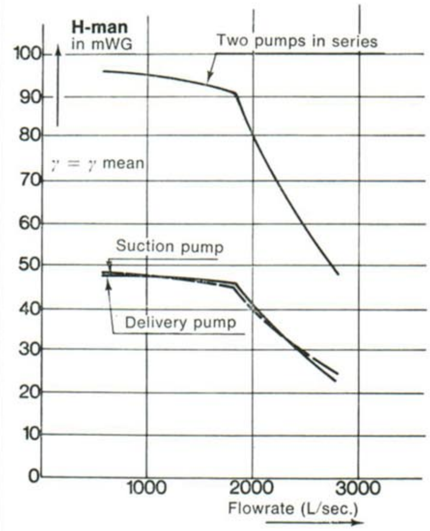


Fig. 1

Fig. 1 shows the characteristics, in the Q-H diagram, of two similar dredgepumps. If these two units are operated in series (in which situation the first is commonly referred to as the suction pump, and the second as the delivery pump), the pressure heads of the two pumps at equivalent flowrates must be summated in order to assess the installation as a whole. The new curve which results from this summation is the pump characteristic for the complete installation. This is also shown in Fig. 1.

To determine the working point of the twin-pump installation, it is necessary to establish the point of intersection of the new pump characteristic and the pipeline characteristic. In determining the pipeline characteristic, account must be taken of the static suction head, the total static delivery head and the resistance offered by all components of the pipeline, i.e. the suction pipe before the first pump, the delivery pipe linking the first and second pumps, and the delivery pipeline after the second pump.

Fig. 2 shows the curves for a single pump and for two pumps in series and also the pipeline resistance curves for a given layout at various delivery distances. The diagram is valid for the hydraulic transport of mixtures of water and normal sand, and is based on a mean specific gravity of 1.2 ton/m³. The points of intersection of a pipeline resistance characteristic with the two pump curves represent the respective working points of an installation with a single dredgepump (the suction pump) and of an installation with two pumps operating in series.

The diagram also shows the working range of the installations, the lower limit of which is formed by the flowrate coinciding with the critical velocity of the pumped mixture and the given delivery pipeline diameter, and the upper limit by the flowrate coinciding with the point of intersection of the curve representing the decisive vacuum of the suction pump and the suction pipe characteristic relating to the mixture being pumped.

It is clear from Fig. 2 that within the working range of the installation at the shortest delivery distances, i.e. less than 750 m or thereabouts, the working points relate only to the installation with the single pump.

At medium distances (750-1,100 m) working points for both single- and twin-pump installations are found.

However, the working point of the installation with two pumps in series coincides with a considerably higher flowrate than that of the installation with a single pump. This implies that where two pumps are operated in series – and provided the pit supply is adequate – the output will be significantly greater than with a single pump.

At large delivery distances (in excess of 1,100 m), the only points of intersection to occur relate to the installation with two pumps in series.

The outputs of these installations, as a function of the delivery distance, are set out in Fig. 3. The output graph for the single pump is divided into the three zones referred to earlier (see issue 84). The first sharp bend corresponds to a delivery distance of 200 m or thereabouts. This is the distance at which the vacuum on the suction side of the pump is equal to the decisive vacuum of the pump. The second bend corresponds to a delivery distance of approximately 1,100 m. This is the distance at which, with the pipeline concerned, the critical mixture velocity is reached. It is possible to exceed this distance with the single pump, but only at the expense of output, since it is necessary to reduce the specific gravity of the mixture (by admitting more water at the suction inlet) in order to avoid a sub-critical situation.

The output graph for the two pumps in series can similarly be divided into three parts. The first bend coincides with a delivery distance of about 900 m. Again, the output is limited by the decisive vacuum of the suction pump. The second bend coincides with a delivery distance of 2,500 m or thereabouts. Here, output is limited by the flowrate which, at the total available pressure head of the two pumps in series, corresponds to the critical velocity of the mixture when pumped through the pipeline in question. A greater delivery distance can be achieved only if the specific gravity of the mixture is reduced by the admission of more water at the suction inlet.

The graphs reproduced in Fig. 3 show that, at delivery distances greater than 200 m or so, two pumps in series can produce a higher output than a single pump.

At distances between 200 and 900 metres, both types of pumping installation are working at the upper limit of their working range. This is determined by the decisive vacuum of the suction pump. In this situation, the speed of both pumps should be reduced (in the case of diesel drive, by throttling down the engines) until the installation is again within its working range.

This has the effect of moving the working point to a lower flowrate within the working range. Because the curve representing the decisive vacuum is usually seen to rise as the flowrate decreases, the decisive vacuum at the new working point will be greater. An added advantage, notably on relatively fast-running pumps, is that reducing the speed is often followed by an increase of the decisive vacuum.

A better solution to the problem of working below the decisive vacuum lies in lowering the position of the suction pump; alternatively, the pump may be placed deeper under the waterline, or a submerged pump placed before the vessel's own suction pump. In the last-named situation, the submerged pump would become the suction pump, the vessel's suction pump the

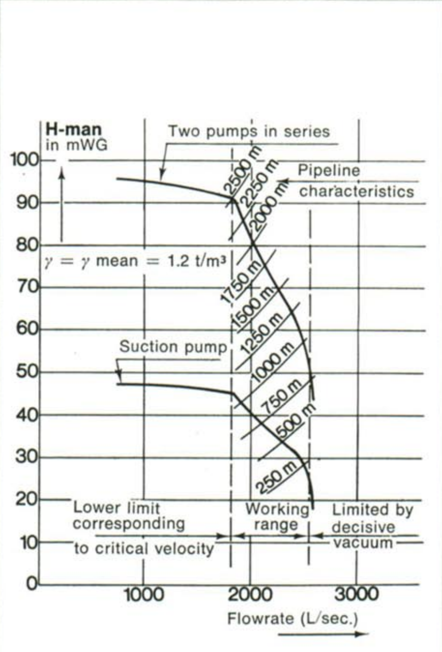


Fig. 2

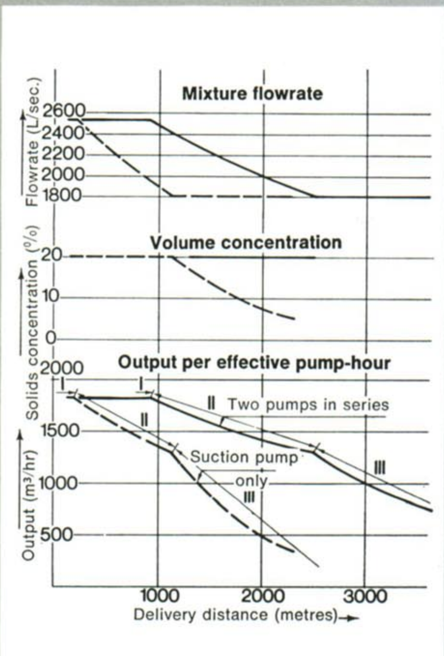


Fig. 3

first delivery pump, and the vessel's delivery pump the second delivery pump. The three units then operate in series.

In practice, when delivering into a relatively short pipeline such as corresponds to the initial portion of the output curve in Fig. 3, when the decisive vacuum limits the specific gravity of the mixture, one would seldom choose a system with three pumps in series. In this situation, a second delivery pump is not required. The best solution in most cases lies in replacing the suction pump by one situated much deeper under the waterline, the more since it is hardly ever feasible to lower the position of the existing suction pump in the dredger, which, if the design is correct, will have been placed as low as possible!

The position of the working points of the two pumps can be ascertained from the working point of the installation as a whole, which is the point at which the pump characteristic arrived at by summation intersects the pipeline resistance characteristic (see Fig. 4). The flowrate of the mixture pumped through the installation can be deduced from the position of the working point of the complete installation, and so it follows that, with the aid of the individual characteristics of the suction and delivery pumps, the working points of the pumps at this flowrate can be determined.

In calculating the total capacity available in the installation employing two pumps in series, the pressure heads of the two pumps should be added together, and not the nominal outputs of the prime movers. This is shown clearly in Figs. 5 and 6.

Fig. 5 contains the characteristics of an installation employing two differing pumps operating in series. Their nominal full-torque points do not coincide with the same flowrate. The joint characteristic of the two pumps, obtained by summating their individual characteristics, is also shown in the diagram.

The point of intersection of the pipeline resistance curve and the joint characteristic of the pumps is the working point of the installation; this is indicated by the letter A. At the flowrate corresponding to this working point, the engine which drives the suction pump is overloaded, and the pump does not reach operating speed (see working point B). The nominal full-torque point for the suction pump (point C) is seen to coincide with an appreciably lower flowrate.

In contrast, the power unit driving the delivery pump is less than fully loaded (see point D). The nominal full-torque point of this pump (point E) coincides with a much higher flowrate. Neither the suction pump nor the delivery pump is thus utilizing fully the driving power available to it.

Fig. 6 similarly shows the characteristics of an installation employing two pumps operating in series. These, however, are so dimensioned that their nominal full-torque points virtually coincide with one and the same flowrate. The pipeline resistance curve intersects the summated characteristic of the two pumps at point A.

Both pump engines operate in the full fuel flow range, therefore their full output is not absorbed. Only when a dredgepump installation such as is represented in Fig. 6 delivers into a pipeline with a resistance curve which intersects the pump curve for the two pumps in series (working point B) at or immediately adjacent to the full-torque point will the rated output of the prime movers be fully utilized. At the flowrate corresponding to working point B, both the suction pump and the delivery pump will be almost on full load (working points E and D respectively) and be absorbing virtually all the power available to drive them.

In this exceptional situation, the whole of the power available is used to good purpose.

Passages of dredgepumps

The size of the passage(s) of the pump(s) after the suction pump is of major importance for the trouble-free operation of the installation as a whole. As a general rule, anything which can pass through the suction pump must be able to pass through the delivery pump(s) with room to spare.

In choosing the dimensions of the impeller, attention must be given to the minimum distance between successive blades on the one hand, and the width of the impeller on the other.

The reference to the distance between successive blades is illustrated in Fig. 7. In the volute itself, the space between the cutwater and the periphery of the impeller is a decisive factor (see Fig. 7).

Where the dredged material contains a large amount of debris, the space within the impeller and the distance between the impeller and the casing

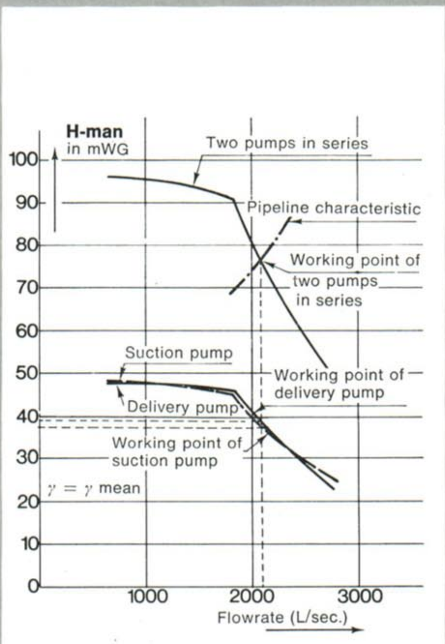


Fig. 4

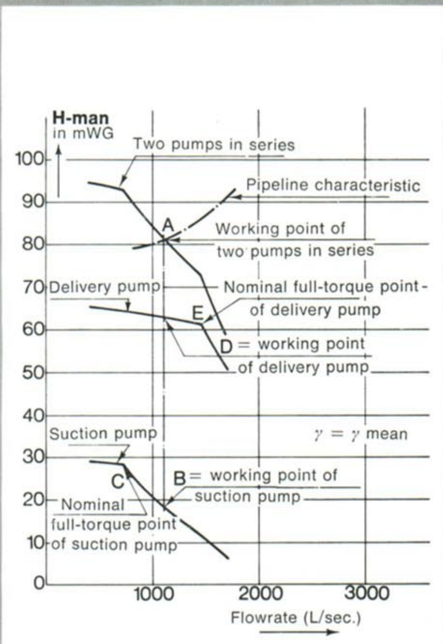


Fig. 5

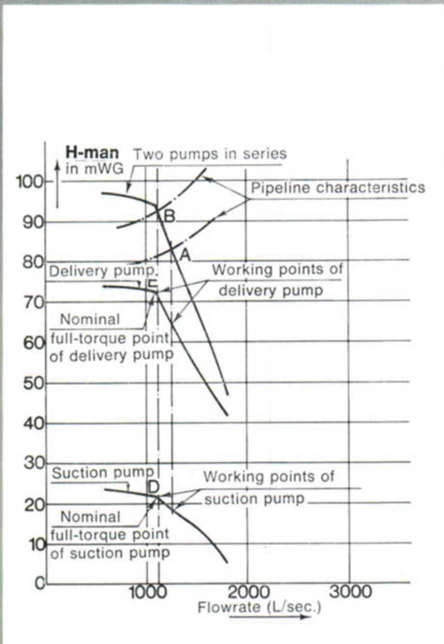


Fig. 6

of the delivery pump must be considerably larger. This is necessary in order to allow angular and oblong objects which have managed to get through the first pump to pass through the second without causing blockage.

Arrangement of twin dredgepumps in series

Where two pumps operating in series are installed in a dredger, the choice of installation seldom presents difficulty. In many instances virtually identical installations are chosen, in which case the characteristics of the two pumps are practically identical. This implies that the positions of their nominal full-torque points virtually coincide and that these can be chosen optimally with respect to the maximum attainable solids concentration in the type of soil regarded as decisive at the design stage.

In many cases where two identical dredgepumps are installed in a vessel which delivers spoil over a large distance (e.g. a cutter suction dredger or a suction reclamation dredger, delivering into a pipeline), the pipeline layout incorporates means whereby either pump can be employed for suction or delivery purposes; in other words, their original roles can be reversed. Moreover, it is not uncommon to find that such installations – in which, of course, it is feasible to operate with only one pump – are so arranged that either pump can be put "on line" by a simple manipulation. This greatly assists major overhaul schedules and facilitates running repairs, particularly on high-wear operations.

On trailing dredgers with twin suction pipes and two identical dredging installations which contribute independently to the filling of the hopper, it is often feasible to connect the pumps in series for the purpose of evacuating the hopper and transporting its contents to the shore via a pipeline.

The characteristics of an installation of this type are given in Fig. 8. This contains the characteristics of the two pumps as independent suction pumps (for the port and starboard suction pipes respectively) and the joint characteristic of the two when operating in series (for shore discharge).

The working range of each pump during trailing-suction dredging is determined by the points of intersection of the pipeline characteristics and the pump characteristics relating to each of the installations. During hopper filling, changes (shifts) occur in the pipeline characteristics: as the ship settles deeper in the water, the static pressure head with respect to the centreline of the pump and the geodetic pressure head (the static head calculated from the waterline), become smaller. The pipeline characteristics applying to an empty hopper (commencement of suction process) and to a full hopper (completion of suction process), between which the characteristic shifts during the suction process, are shown in Fig. 8.

The working range of the two dredgepumps operating in series to empty the hopper and transport the mixture to the shore is also shown in the Q-H diagram in Fig. 8. On the left-hand side, the range is limited by the flowrate corresponding to the critical velocity of the pumped mixture and the given delivery pipe diameter; on the right-hand side by the flowrate corresponding to the point of intersection of the curve representing the decisive vacuum of the pump and the suction pipe characteristic pertaining to the pumped mixture. Pipeline characteristics for a number of delivery distances are given in Fig. 8.

There are numerous installations in which the leading dredgepump serves as a suction pump. As this is required to achieve a high decisive vacuum, but provide only a low delivery pressure, it differs from the other, delivery, pump in the installation, being small and squat, and having a smaller impeller diameter and a relatively low running speed. In the majority of installations of this type, the suction pump is of the submerged type.

With dredging depths steadily increasing, and mixtures of higher specific gravity being transported, the use of such pumps is growing. In most installations of this type, the suction pump is situated:

a) in the vessel, in the lowest possible position and as close as possible to the point of entry of the suction pipe; this is accordingly situated as deep as possible in the hull;

b) on a ladder, beneath the waterline. Hitherto, most large pumps were driven, via a long shaft, by electric motors situated above the waterline; now, however, direct drive employing a submerged motor is feasible.

The smaller pumps were mostly driven, via reduction gearing, by a

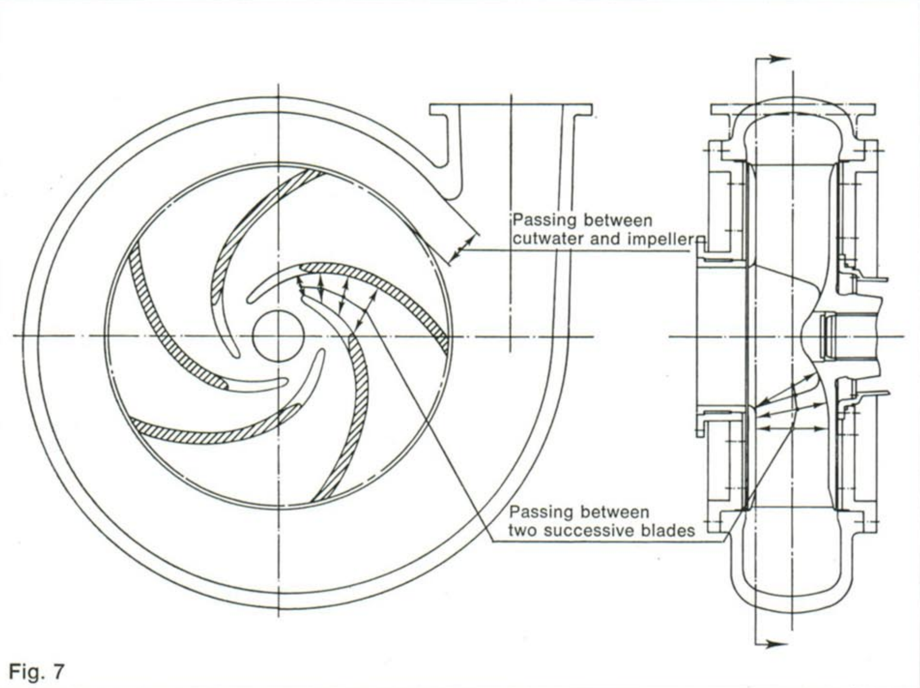


Fig. 7

hydraulic motor. Using this drive principle, IHC Holland has developed a series of compact, standardized, underwater pump units.

These consist of a dredgepump, reduction gearing and hydraulic motors, and are designed for mounting on a ladder.

There have been many special constructions relating to these types of drive. They include submerged systems having a small, watertight power plant mounted on the ladder.

c) in the suction pipe.
IHC Holland has developed an integral unit, consisting of a pump and an electric motor, for incorporation in the suction pipe. It is designed for underwater operation. In the majority of applications, however, this will not serve purely as a suction pump, but will transport the mixture directly to the hopper or into barges.

In almost all cases, these smaller suction pumps will be equipped with an impeller with a small number of blades (normally 3). This is in order to achieve the passage necessary to match the larger pumps which will operate in series with them. An added advantage of a 3-bladed impeller lies in the considerably steeper pump curve at constant speed in the Q-H diagram (see Fig. 9).

In the interests of simplicity, physical size, trouble-free operation and cost, the choice of drive system for the suction pump usually falls on a non-synchronous electric motor fed directly from a high-capacity supply, an "electric shaft" (non-synchronous electric motors and generators driven by a diesel engine), or a "hydraulic shaft" (hydraulic motors driven by hydraulic pumps with a fixed delivery).

With the first of these systems, a constant-speed drive is obtained, and accordingly the pump curve in the Q-H diagram is a constant-speed curve. A "hydraulic shaft" powered by an electric motor fed from a high-capacity supply is virtually a constant-speed drive, albeit a significant loss occurs in the "shaft" – the hydraulic system.

With either an electric or hydraulic "shaft" powered by a diesel engine, the pump characteristic in the Q-H diagram is a diesel-drive characteristic, which is adversely shifted and influenced by the efficiency of the electric or hydraulic "shaft" which serves as a bridge for the transmission of power from the prime mover to the dredgepump.

The dredging process

In order to achieve a uniform dredging process, fluctuations in the mixture velocity at the suction inlet must be prevented as far as possible. Such fluctuations occur as a result of variations in the soil dredged or pit supply, and of more or less complete blockage of the inlet by, for example, the collapse of vertical faces or by the suction mouth being pressed too hard against them.

Within the suction pipe, the velocity fluctuations interfere with the composition of the mixture, sometimes causing vacuum bubbles. The collapse of large faces or the complete blockage of the suction inlet can even cause the pump to choke.

Unfilled areas (vacuum) in the pipe result in acceleration and deceleration of portions of the mixture, causing them to collide with each other and with the walls of the pipes and components, possibly producing water hammer. If the pump chokes, the resumption of the mixture flow will be accompanied by severe water hammer, which can cause immense damage to the pump and other parts of the system. It is even known to have resulted in the sinking of dredgers.

Installation with two pumps in series, but without safety or special control system

If no safety system or special controls are to be incorporated in a pumping installation to prevent the considerable fluctuations in velocity and the resulting water hammer, care must be taken to ensure that the suction pump is so dimensioned that it will under all circumstances offer the mixture to the second (delivery) pump in the vessel. This means that the delivery pump may not perform a suction function, and because of this there will be less vacua in the mixture in the pipeline and thus considerably less risk of water hammer occurring.

This, however, is conditional upon the suction pump being capable of elevating the mixture to a given, fixed height above the waterline in the same way as a hopper-filling pump. This, of course, implies that, proceeding from a fairly static design depth for the suction pump, the aim in selecting the best pump characteristic from the various alternatives, will be to arrive

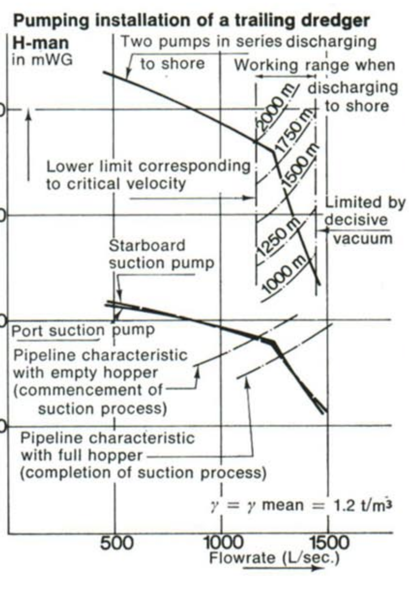


Fig. 8

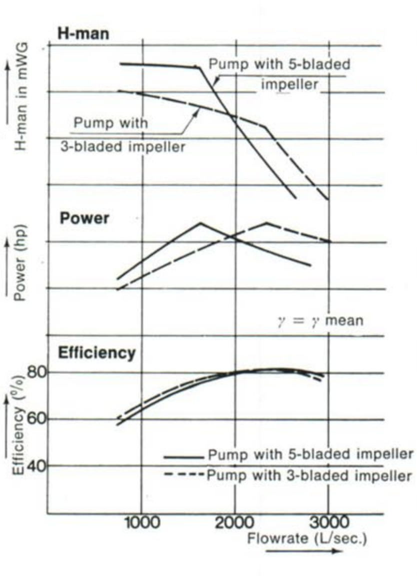


Fig. 9

at a constant-speed curve at which, with a constant mixture s.g., the drop in the pressure head as the flowrate increases will be limited throughout the working range. At the same time, a stable pumping situation will be desired in view of the risk of sedimentation when working in the vicinity of the critical velocity. As a pump with a 3- or 4-bladed impeller running at a constant speed possesses a more stable curve – i.e. which rises more steeply as the flowrate diminishes – than a pump with a 5-bladed impeller, it is preferable to choose a 3- or 4-bladed impeller for the suction pump.

An installation of this type which lacks a safety system or special controls for the prevention of choking of the pumps and the occurrence of water hammer possesses the drawback that it must be operated with extreme care if these evils are to be avoided. If they do occur in spite of caution, there is a grave risk of damage, leading to repair costs and downtime; at best, a large number of operations must be carried out in order to get the dredger working properly again.

If, for example, the suction pipe of a reclamation dredger is buried by the collapse of a face, but escapes damage, the vessel must be moved astern and an attempt made to raise the pipe. If this succeeds, the area of the collapse must be cleared, after which a fresh approach must be made. All this costs time and output, and widens the gap between actual production and the level of which the dredger is capable.

Another disadvantage lies in the high power required to ensure that the mixture is at all times offered to the first delivery pump. In the suction pump installation, the effects of this are reflected in a larger drive and transmission system, heavier supporting structures, gantries, etc. The power required to drive the suction pump must be further increased to offset the losses occurring in the "shaft".

A point in favour of this arrangement is that it eliminates the problem of suction properties of the second pump in the dredger.

The inlet can now be designed with a view to maximum size of the passage, rather than suction properties, which implies that the extension of the impeller blades into the inlet can be left undone. The bore of the second pump can therefore be of maximum dimensions. In many instances the blades are so constructed that in the bore they are parallel to the shaft. In practice it is common for the areas of blade which project into the bore to be burned away to enlarge the passage. Because the suction properties are now secondary, the hub nose can be further recessed, thereby considerably reducing the risk of debris, such as wires, vehicle tyres, etc., becoming trapped in the pump (see Fig. 10). With these measures, the requirements stated earlier with respect to the size of the passage of the second dredgepump can be simply met.

Installation with two pumps in series and having safety or special control system

The presence of a system for preventing water hammer (a vacuum relief valve), or special controls which afford a more continuous process, results in more uniform and higher output, and precludes irregularities in the process which can lead to water hammer.

Among these special controls is an automatic booster controller, the function of which is to regulate the speed of the delivery pump in accordance with the pressure existing at the inlet of this pump; and a system of small valves which are introduced into the suction pipe and the pipe linking the suction and delivery pumps, and which can be opened to admit water, thereby affording local control of the mixture.

Where a safety system or process control devices are fitted, the delivery pump may perform a suction function. Where means for process control exist, a substantially higher level of vacuum is permissible than in installations with only a vacuum relief valve. It is no longer necessary for the suction pump unit to transport the mixture to the delivery pump.

A further advantage of these controls is that they enable the power requirement of the leading pump to be substantially reduced, since its role is now principally one of suction; this saving, in turn, allows the pump, its supporting structure, the lifting gear, etc. to be smaller and lighter. Of the power saved, a portion – but only a portion – must be diverted to the delivery pump in order to obtain the same output curve for the installation as a whole. Because it can now perform a suction role, the second pump can without difficulty take over part

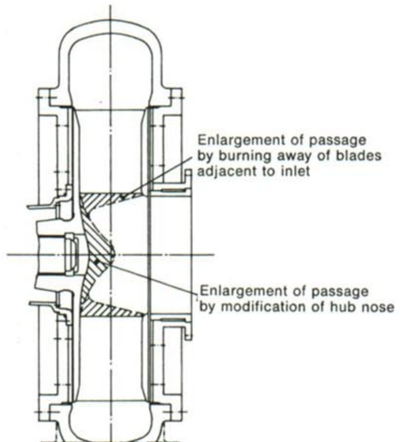


Fig. 10

of the role of the submerged pump, i.e. the transport of the mixture between the two pumps.

As stated, less than the whole of the power saved needs to be diverted to the delivery pump in the vessel. This applies particularly to installations embodying an electric or hydraulic "shaft" to transmit power to the pump, since the reduction in the requirement of the suction pump is accompanied by a reduction in the losses incurred in the "shaft".

In designing the installation as a whole, it must be borne in mind that the delivery pump need not produce so much suction as to exceed its cavitation limit (its decisive vacuum).

If it did, this would impose limits on the solids concentration in the mixture. This situation arises from an insufficiency of power for the suction pump where this is positioned relatively far below the delivery pump. Then, the suction pump has to deliver more in order to make good the deficiency in suction of the delivery pump.

Three or more pumps in series in a dredger

An installation in which three or more dredgepumps are installed in series is subject to the same rules for calculation and the same considerations as apply to a two-pump system, but to a greater extent. Here, too, the characteristics of the various pumps must be matched; the overall pump characteristic must be calculated by summating the individual characteristics in order to arrive at the output and working points; the passages of the two delivery pumps must be larger than that of the suction pump; and adequate measures must be adopted to prevent the (greater) risk of water hammer in the installation.

The characteristics of three pumps operating in series are reproduced in Fig. 11. In this the installation is assumed to comprise a ladder-mounted, submerged suction pump on a cutter suction dredger or a suction reclamation dredger, and two identical delivery pumps on board the vessel.

When three, or even more, dredgepumps are operated in series in a single installation, the terminal pressure, i.e. the pressure at which the mixture leaves the final pump, can be very high. The dredgers built more recently for delivery into a pipeline, especially those of the cutter suction type, reveal a trend towards higher and higher terminal pressures.

The chief advantage of multi-pump installations is that everything is contained in one item of plant and that, by switching one or more pumps in or out of circuit, a large range of delivery distances can be covered.

There are, however, drawbacks associated with the resulting, high, terminal pressures. These include:

a) The very heavy structures required to absorb the increased stresses in the pipeline and, frequently increased wear.

   The need for additional strength is reflected throughout the dredger in the delivery pumps, dredging valves, turning glands, the construction of the initial pontoon carrying the floating pipeline, the delivery pipeline itself, and even in the support structures and the repair tools carried on board. Ancillary equipment such as work barges also frequently have to be larger.

   In addition, foundries and builders of dredging plant are called upon to produce increasingly large and heavy units, which means that they can no longer use standard patterns (many of which are of optimum form), or standardized assemblies and components. Furthermore, a number of cases have shown that existing structures, when subjected to the higher pressures, do not meet the requirements for economic life. A typical example is to be found in the shaft seals of a pump designed for terminal pressures in excess of the range of 11-14 kgf/cm²; the Simmerring seals previously employed for shaft sealing are incapable of withstanding such pressures, so a new type must be used. This makes the installation considerably more costly.

b) The difficulty of handling the heavier dredger components.

   Components must be regularly inspected for wear (e.g. wall thicknesses measured) and where necessary repaired. Dismantling, repair and reassembly of the considerably heavier parts, particularly on larger installations, are more difficult, therefore repair costs will be appreciably higher. Naturally, the added difficulty can largely be offset by means of special gear for hoisting and other purposes, but this, too, adds to the cost of the installation.

c) Premature unserviceability of components.

   It is clear from what has been said that the heavier components, when they reach the end of their useful life, will have a greater residual wall thickness. This, however, by no means implies that they can automatically be transferred to another dredgepump installation, since the connexions and dimensions will often differ as a result of the greater wall thickness and/or other constructional factors.

In the light of the foregoing, it is open to doubt whether the advantages of installing three or more pumps in series in a dredger, with a consequently high terminal pressure, outweigh the disadvantages. The alternative is to employ one or more booster stations at intervals along the pipeline. This approach will be discussed in detail in the following article in this series.

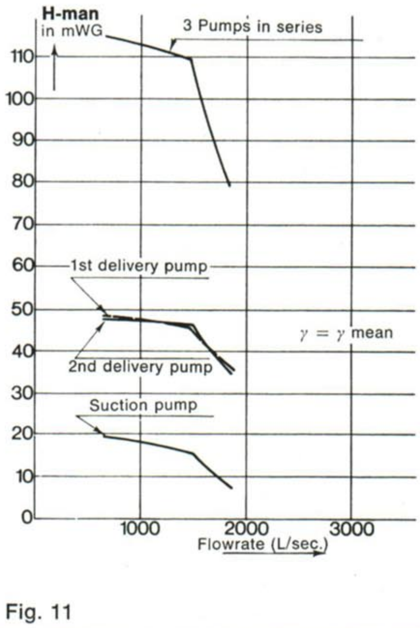


Fig. 11